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AN EXPERIMENTAL EVALUATION OF THE
NON-NEWTONIAN SCALING EFFECTS IN A
ROTODYNAMIC LEFT VENTRICULAR ASSIST DEVICE

A Dissertation

Presented in Partial Fulfillment of the Requirements for the Degree Doctor of Philosophy in the Graduate School of The Ohio State University

By

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*****

The Ohio State University
1999

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ABSTRACT

Significant work has been done in the last 10 years to advance the technology of long-term mechanical circulatory assistance, particularly the left ventricular assist device (LVAD). Traditionally, rotary LVADs have been developed using conventional fluid dynamic design methods and Newtonian scaling laws, since non-Newtonian effects were previously assumed to be of second-order importance.

To evaluate centrifugal pump performance scaling and flow patterns in a non-Newtonian fluid, the Large-Scale Rotor Testbed (LSRT) at The Ohio State University was developed to test two 9X-scale blood pump impellers in a baseline volute housing of the Innovative Ventricular Assist System (IVAS) designed by the Cleveland Clinic Foundation. Non-Newtonian fluids yielded pump performance deficits of first-order importance, or up to 11% of the Newtonian performance. Thus, the non-Newtonian effects were of the same magnitude as substantial variations in the impeller geometry.

Moreover, the dimensionless pressure- and flow-coefficients showed that the non-Newtonian performance deviated from the similarity laws at critical
Reynolds numbers that were 2.4-2.7 times higher than the Newtonian value of 71,000. Above the critical Reynolds number, the non-Newtonian fluids followed a similarity behavior, but it was different from the Newtonian case. The deviation increased with the magnitude of shear-thinning behavior as measured by the Weissenberg number.

Shear-thinning xanthan gum solutions were used as non-Newtonian test fluids in concentrations from 0 to 1,200 ppm. Fluid samples were characterized in a Couette rheometer to determine viscosity behavior, biological degradation, and shear-induced polymer chain breakdown. The solutions proved to be stable and useful for a duration of up to two weeks of routine LSRT testing.

Because the LSRT pump operates in a low-specific speed, low-flow regime, flow visualizations revealed a strong adverse pressure gradient and a prominent inverse Ekman layer in the inlet region. The flow patterns along the upper housing were dependent on the circumferential position, but not the pump flow conditions. The rotor disk surface flow patterns were only dependent on the pump flow conditions.
I will give you a new heart and put a new spirit within you;
I will take the heart of stone out of your flesh and give you a heart of flesh.

— Ezekiel 36:26 (New King James Translation)
ACKNOWLEDGMENTS

First and foremost, I wish to thank my adviser and mentor Gerald Gregorek, who commissioned and enabled me to do this research. I hope to carry on his teaching vision and thus produce many “sons” of my own. My other committee members provided valuable assistance and direction: John Lee, with his years of seasoned wisdom and words of faith, and Richard Bodonyi, for his assistance in analytical and numerical methods. A special thanks goes to Bill Smith, whose knowledge, vision, and grace supported me in accomplishing this work. I look forward to continuing my association with him. Kurt Koelling served on my general exam committee; I am grateful for his input and for the cooperation of CAPCE in the rheological measurements of our test fluids.

This work was made possible by funding from the National Institutes of Health, specifically NIH prime contract N01-HV-58159. I would especially like to acknowledge the staff of the Cleveland Clinic Foundation’s Biomedical Engineering Department, who made the subcontract to OSU possible. Dave Horvath in particular provided IVAS performance data for scale comparisons and the CAD files necessary for machining the LSRT components. Len Golding managed the overall IVAS contract.
The staff of the AARL was very helpful in the allocation of laboratory space and in the fabrication of test hardware. Tom Veit provided guidance, tools, after-hours machine shop access, and a true friendship throughout the fabrication process. Mike Grunden of the Physics Department's machine shop did the CNC milling of all Plexiglas® LSRT housing components.

I owe a special thanks to the undergraduate crew who helped build the OSU Hydrodynamics Laboratory from the floor up: Matt Cole, Barry Craft, David Jungeberg, Erik Foster, and Matt Schweitzer. Their combined designing, welding, machining, sanding, programming, debugging, testing, and internet surfing are sincerely appreciated.

I would also like to thank my indefatigable loyal partner, wife, and friend, Jill. Her boundless love and tireless prayers have sustained me day and night. A wife of noble character is her husband’s crown, and I am bountifully blessed to have the privilege to love and serve mine for a lifetime.

Above all others, I would like to thank my savior, Jesus Christ. My awesome creator has designed and gifted me with the abilities to complete this project for His glory and good pleasure. Along the way, He has inspired, guided, energized, strengthened, and comforted me in my efforts. I count all my accomplishments as loss compared to the surpassing greatness of personally knowing Jesus Christ my Lord.
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<td>AARL</td>
<td>The Aeronautical and Astronautical Research Laboratory</td>
</tr>
<tr>
<td>CCF</td>
<td>The Cleveland Clinic Foundation</td>
</tr>
<tr>
<td>CMC</td>
<td>Carboxymethylcellulose</td>
</tr>
<tr>
<td>GPM</td>
<td>Flow rate [gallons per minute]</td>
</tr>
<tr>
<td>IVAS</td>
<td>Innovative Ventricular Assist System</td>
</tr>
<tr>
<td>LPM</td>
<td>Flow rate [liters per minute]</td>
</tr>
<tr>
<td>LSRT</td>
<td>The Large-Scale Rotor Testbed</td>
</tr>
<tr>
<td>LVAD</td>
<td>Left Ventricular Assist Device</td>
</tr>
<tr>
<td>OSU</td>
<td>The Ohio State University</td>
</tr>
<tr>
<td>PAA</td>
<td>Polyacrylamide</td>
</tr>
<tr>
<td>PPM</td>
<td>Weight concentration [parts per million]</td>
</tr>
<tr>
<td>RPM</td>
<td>Impeller speed [revolutions per minute]</td>
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Arabic Symbols

<table>
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<th>Symbol</th>
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<tr>
<td>C</td>
<td>Solution concentration [weight/weight]</td>
</tr>
<tr>
<td>$c_{ew}$</td>
<td>Impeller endwall (upper housing) clearance [mm]</td>
</tr>
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D Impeller diameter [cm]
g Gravitational acceleration [m/sec^2]
gsp Specific gravity [dimensionless]
H Head pressure [m]
K Newtonian threshold constant [seconds]
n Slope of power-law region of the viscosity curve [dimensionless]
PIN Pump inlet pressure [mm mercury]
POUT Pump discharge pressure [mm mercury]
pT Total or pitot pressure [mm mercury]
Q Pump volumetric flow rate [liters per minute]
R Impeller radius [cm]
ReD Reynolds number based on rotor diameter [dimensionless]
u Local velocity component in the streamwise direction [cm/sec]
U Freestream (or reference) velocity [m/sec]
v Local velocity transverse to the streamwise direction [cm/sec]
v Flow rate [m^3/sec]
We Weissenberg number based on RPM [dimensionless]

Greek Symbols

Δp Pump pressure rise, pOUT−PIN [mm mercury]
φ Pump flow coefficient [dimensionless]
\[ \lambda \quad \text{Elastic fluid relaxation time constant [sec]} \]
\[ \gamma \quad \text{Strain [dimensionless]} \]
\[ \dot{\gamma} \quad \text{Strain (or shear) rate, } \frac{du}{dy} \text{ for a pure shear flow [sec}^{-1}] \]
\[ \eta \quad \text{Coefficient of dynamic viscosity [centipoise or mPa-sec]} \]
\[ \eta_0 \quad \text{Zero-shear viscosity coefficient [centipoise or mPa-sec]} \]
\[ \eta_\infty \quad \text{Infinite-shear viscosity coefficient [centipoise or mPa-sec]} \]
\[ \eta_s \quad \text{Newtonian solvent viscosity [centipoise or mPa-sec]} \]
\[ \nu_{sp} \quad \text{Specific speed [dimensionless]} \]
\[ \theta \quad \text{Circumferential position [deg]} \]
\[ \rho \quad \text{Density [kg/m}^3]\]
\[ \tau \quad \text{Shear stress [Pa]} \]
\[ \tau_0 \quad \text{Reference shear stress [taken here as 0.1 Pa]} \]
\[ \omega \quad \text{Rotational velocity [rad/sec]} \]
\[ \psi \quad \text{Pump pressure coefficient [dimensionless]} \]
CHAPTER 1

INTRODUCTION

It is estimated that congestive heart failure currently affects 4 million people in the United States, leading to 50,000 deaths annually (Marcus 1998, 65). Although some patients may endure years of a stabilized stage of the disease, the prognoses for most are unpredictable and the indicators crude (Massin 1996, 40). When the end stages of the disease become apparent and medical palliation is no longer effective, heart transplantation is the only permanent option.

Heart transplants provide new life for 2,300 people per year, but almost 4,000 are eligible for such procedures under current criteria. As many as 25% of end-stage heart disease patients die while on the wait-list for a donor heart (Marcus 1998, 65). However, even transplantation does not remove all risks. At the Texas Heart Institute, transplant procedures since 1982 have revealed a mortality rate of 23% after 1 year and 40% after 5 years (Cooley 1996, 9). These statistics are borne of several perioperative complications, including acute right ventricular failure, primary donor heart failure, and bleeding complications. Postoperatively, allograft rejection and immunosuppression complications may
become a factor (Frazier and Macris 1996, 201-206). In short, performing a heart transplant is considered an extraordinary means to treat heart failure.

The engineering solution to this dilemma seems straight-forward: Develop a mechanism to pump the blood in lieu of the ailing heart. A tremendous amount of work has been accomplished on this front; many devices have been tested and a few certified by the FDA. One approach has been more of a trial-and-error methodology whereby a device is designed, fabricated, and implanted in an animal, whose demise hopefully provides enough information for another design iteration. Another philosophy is to thoroughly research the underlying phenomena in an attempt to completely understand the problem and generate an engineering process firmly founded on "first principles."

Consequently, the ultimate goal of the present body of work reflects the latter philosophy in support of the design and testing of a left ventricular assist device developed by The Cleveland Clinic Foundation (CCF). The rotodynamic Innovative Ventriclelular Assist System (IVAS) is the subject of Reynolds scaling experiments to evaluate the effects of a non-Newtonian fluid on blood pump design. Results are presented for two 9X-scale impeller models of the centrifugal IVAS pump in the Large-Scale Rotor Testbed (LSRT) at The Ohio State University.
1.1 THE HUMAN HEART

To thoroughly understand the feasibility and impact of mechanical circulatory assistance, the structure and function of the heart must be grasped. Therefore, a brief compendium of the physiology and disease of the heart will be documented here.

1.1.1 Healthy Heart Function

Shown in Figure 1.1, the heart is a complex double pump beating 80,000-100,000 times and pumping 7,900 liters (2,100 gallons) per day (O’Connor 1991, 38). The superior vena cava and the inferior vena cava discharge into the right atrium of the heart. The venous blood, having been depleted of oxygen, flows into the right ventricle where it is pumped to the lungs for oxygenation. From there, it flows into the left atrium and left ventricle and is discharged into the ascending aorta. The aorta supplies the coronary arteries, brachiocephalic trunk, carotid and subclavian arteries, and the descending aorta and is responsible for distributing blood to the entire body.

Because considerably greater pressures are required to circulate arterial blood throughout the body than venous blood through the lungs, the left ventricle is the most critical component of the four chambers of the heart. Consequently, it is the component that fails most frequently and often most catastrophically (Institute of Medicine 1991, 15).
1.1.2 Congestive Heart Failure

Congestive heart failure is the basic degradation of the heart's pumping effectiveness. It is typically a gradual process of deteriorating performance and may occur in either ventricle or both (biventricular failure). Surgically treatable heart disease can be separated into four categories: valvular disorders, diseases of cardiac innervation (e.g., arrhythmias), muscle disorders, and cardiac malformation (congenital heart defects). While the former two have clearly-defined treatments in artificial heart valve implantation and cardiac pacemaker application, the latter two do not. The muscle disorders (including coronary artery disease, myocardial infarction, and cardiomyopathy) and irreparable
heart malformations can become fatal if extraordinary measures are not taken in the form of heart transplantation or extensive mechanical circulatory support (Unger 1984, 3).

1.2 MECHANICAL CIRCULATORY SUPPORT

Circulatory assistance via mechanical means is not a new concept. Since 1953, cardiopulmonary bypass (heart-lung) machines have been used to maintain total-body perfusion during open-heart surgery. And since 1967, the intra-aortic balloon pump has assisted patients with impaired, but reversible, cardiac output. However, the 1970s and 1980s saw a surge in diverse research efforts by a handful of institutions to concentrate on replacing the heart with an implantable mechanical device. The 1990s have seen a focus on ventricular assist device development. Cooley (1996, 9) estimates that each year, 60,000 patients with end-stage heart failure might benefit from mechanical circulatory support. This number could grow to as high as 70,000 patients annually by the year 2010 (Institute of Medicine 1991, 75). At the time of this writing, 20-year-old Bill Merriman is being kept alive with an external blood pump at The Ohio State University Medical Center. He is awaiting a heart transplant to alleviate idiopathic cardiomyopathy (Baird 1999, 1B-2B).
1.2.1 The Artificial Heart

The early goal to develop the "artificial heart" quickly converged on the permanent mechanical replacement of the cardiac function and evolved into the so-called Total Artificial Heart (TAH) initiative. Several institutions were involved in this national ambition, fueled by funding from the NIH, and many innovative designs emerged; one concept by Thermo Cardiosystems even involved a compact Plutonium-238 power source (Kolff 1976, 52-62 and Stipp 1996).

Figure 1.2 Early artificial heart: the Jarvik-7
[reproduced from Smithsonian Institution 1996]
The watershed for artificial heart development came in 1982 when the Jarvik-7, shown in Figure 1.2, was implanted in Barney Clark and sustained his life another 112 days. This proved the feasibility of implantable circulatory support and stimulated the competition in alternatives to transplantation.

Unfortunately, many TAH designs are bulky and exhibit complications associated with hemorrhage, thromboembolism, infection, and stroke; these issues must be addressed to make the TAH a viable alternative (Cooley 1996, 11). Therefore, these devices are no longer counted on as the panacea to heart replacement, but they have served a useful role in advancing the technology.

One of the short-term applications of TAH devices has been as a bridge to transplantation. Between 1988 and 1993, the TAH implantation rate decreased sharply from 75 to less than 10 per year, but the statistics show a favorable survival rate if the TAH is used temporarily to bide time to locate a suitable heart donor. By 1995, a total of 12 different TAH device designs were implanted in 265 patients worldwide. This led to 173 eventual heart transplants with 115 survivors, or 66% (Pantalos 1995, 7-8).

1.2.2 The Left Ventricular Assist Device

As a result, the push to engineer a total heart replacement has been redirected in pursuit of more reliable, nearer-term, implantable ventricular assist devices. Throughout the 1990s the National Heart, Lung, and Blood Institute (NHLBI) has spearheaded the movement to develop a left ventricular assist
device (LVAD) based on the potential for providing bridge-to-transplant and bridge-to-recovery (also known as "support-to-weaning") circulatory support. This effort has culminated in FDA approval in October 1998 of two portable devices powered by a battery pack worn around the waist. The HeartMate™ by Thermo Cardiosystems Inc. and the Novacor™ LVAS by Baxter Healthcare Corporation allow for untethered patient activity. These two devices have already helped many patients with irreversible heart failure who were likely to die within 24 to 48 hours without the LVAD option (Food and Drug Administration 1999).

These two LVADs are pulsatile in nature, while other devices involve pulsatile, rotodynamic (axial and centrifugal), and nutating design concepts. Since 1987, various forms of ventricular assistance have been used in almost 200 cases each year; about 50 cases per year are bridge-to-transplant situations (Pantalos 1995, 6). These numbers are expected to increase dramatically in the next few years in the wake of the FDA approval of the HeartMate™ and the Novacor™ devices.

Ventricular assistance is also useful as a bridge to recovery. Studies have shown that if permitted a period of reduced activity, a weakened heart may actually recover and the damage may be partially reversed. In such cases, a heart transplant may be totally unnecessary where an LVAD is implemented for a duration of several weeks; Dasse et al. report marked improvements in certain
patients after an average of 79 days of ventricular assistance (1995, 15). And more recently, the American Heart Association reported that of 111 patients who had LVAD support while awaiting a transplant, five could be weaned from the device without the need for transplantation (AHA 1998). In this study, LVAD support was maintained for an average duration of 100 days.

As remarkable as these statistics may seem, the far-reaching impact of LVAD systems becomes incontrovertible when the savings in medical costs are considered. An LVAD device like the Cleveland Clinic’s Innovative Ventricular Assist System (refer to Figure 1.3) could cost about one-fourth the $235,000 heart transplant expense (Parrino and Stone 1997, 11).

Figure 1.3 A prototype of the CCF IVAS ventricular assist pump [reproduced from Parrino and Stone 1997, 10]
1.2.3 The Innovative Ventricular Assist System

The subject of study for the present work is an LVAD being developed by the Cleveland Clinic Foundation. This plum-sized device is known as the Innovative Ventricular Assist System (IVAS). IVAS takes advantage of a high-speed centrifugal impeller (2,500-3,500 RPM) and a unique blood-lubricated journal bearing to pump 5-10 liters per minute at an input power of just over 6 Watts. An implanted backup battery can provide a few hours of operation, but primary power comes from the transcutaneous energy transfer system (TETS). Figure 1.4 shows the implantation of the IVAS heart pump and other

![Diagram of IVAS heart pump and other components]

Figure 1.4 Implantation of the CCF Innovative Ventricular Assist System [reproduced from Parrino and Stone 1997, 11]
components. The device is situated to draw blood out of the apex of the left ventricle and inject it back into the ascending aorta.

The primary advantages of IVAS are its size and power consumption. Due to its dimensions and mass, the LVAD system can be implanted in all but the smallest of children. This pump has yet to be used on a human subject, but it is currently undergoing extensive animal testing. The IVAS assembly drawing is shown in Figure 1.5, and the interior view, in Figure 1.6.

The innovative impeller assembly, shown in Figure 1.5 and Figure 1.6, pumps the blood without undue hemolysis. This is accomplished through a double-impeller design; the primary impeller does the cardiac pumping and the secondary impeller draws a fraction of the blood through the journal bearing for

Figure 1.5  Assembly view of the CCF IVAS heart pump
lubrication purposes. Both impellers are mounted to the annular rotor assembly.

The motor is based on an "inverse" design. Traditional motor designs place the rotor on the inside and the stator on the outside. The IVAS motor, by contrast, locates the permanent DC magnets in the outer impeller assembly (rotor) but the motor windings are found in the stationary hub (stator). In this manner, the lubricating blood is free to travel axially through the journal bearing in the rotor/stator clearance gap. Radial magnetic forces keep the cylindrical rotor assembly properly aligned on the semi-elliptical stator housing.
1.3 LARGE-SCALE PUMP MODELING

To effect pump pressure measurements and flow visualization studies of the IVAS pump configuration, the Large-Scale Rotor Testbed (LSRT) was developed at The Ohio State University’s Aeronautical and Astronautical Research Laboratory (AARL). Figure 1.7 shows the LSRT installation, which is a 9X scale model of the IVAS "baseline" volute housing (CCF assembly number 3374). This facility permitted highly accurate and repeatable measurements to be made using two impeller geometries, CCF part numbers 3386 and 3452. Results from the LSRT facility form the basis of the current body of work.

Figure 1.7 The OSU Large-Scale Rotor Testbed facility
CHAPTER 2

BACKGROUND

Dynamic similarity is an important concept in fluid flow modeling. The term "scaling effect" or "scaling law" seems to imply a universal applicability of the phenomenon in question, such that results based on one set of controlled parameters can be utilized in another case involving a different set of related parameters. In this way, similarity laws are extremely useful for design purposes because they supplant the need for a unique solution every time a new device is designed; the results from one geometry can be scaled to provide reliable performance predictions for another. And, since the main focus of the present work is to explore the dynamic scaling effects of a centrifugal LVAD's pumping a non-Newtonian working fluid, the applicability of primarily Newtonian similarity laws must be verified before confidence can be placed in the results.

There are different ways scaling laws can be implemented, and the applicability depends on the type of flow field and the properties of the fluid. Certain regimes of pump operation create flow fields that violate the similarity
conditions. Varying fluid properties, such as strain-dependent viscosity, can hinder the application of Newtonian scaling laws. Therefore, this chapter will address the issues of dynamic scaling and similarity, the fundamentals of centrifugal pump design and performance, the nature of non-Newtonian fluid mechanics, and the specific characteristics of blood flows.

2.1 DYNAMIC SCALING AND SIMILARITY

Dynamic scale is the most fundamental classification of a flow field, and similarity is a means of matching the dynamic scale. Dynamic scaling is equally important in computational and experimental flow modeling. Two flows obeying the same dynamic scaling are known as similar and will have congruent streamline patterns (angles, gradients), matching force coefficients (lift, drag, wall friction, moment), and equivalent flow qualities (turbulence, swirl, cavitation).

In computational work, length and time scales may be manipulated to enable a dimensionless family of similar solutions that holds true for an infinite combination of parameters. Reynolds scaling, which is commonly used to achieve dynamically equivalent experimental flow fields, is another application; it relies on maintaining the same balance between momentum and viscous influences. In wind tunnel testing, the model may be one or two orders of magnitude smaller than the full-scale aircraft, yet the results are directly
comparable in the best case scenario. Therefore, differing size, speed, and fluid viscosity affect the changes in dynamic scale.*

In order for flow fields to be similar (or dynamically similar), they must exhibit two criteria. First, the geometric scales must be proportional. This affinity requirement is pertinent to the shape of an immersed body or the contour of a wetted flow passage. Second, the streamline patterns must be proportional. To complement the latter requirement, flow field initial conditions or other far-field boundary conditions must also be similarly maintained.

* One interesting point about the current pump testing is that the model is an order of magnitude larger than the true-scale article. And, the large-scale experiments are conducted using different fluids than were used in the true-scale pump.

\textit{Similarity Parameters}

Generally established by dimensional analysis, the parameters of dynamic similarity for hydrodynamics arise out of force balances. If the ratio of forces everywhere is congruent between two flows, the corresponding streamlines will be proportional. To develop the order-of-magnitude analysis for the similarity parameters, two simple steady flow cases are considered with the following influential forces:

i. Friction, inertia, and pressure

ii. Inertia, gravity, and pressure

For both of these cases, the pressure can be determined from the first two (force balance), so there are only two critical independent forces.
Similarity of the first case will require a balance between inertia and friction. The local velocity, \(u\), is proportional to the freestream (reference) velocity, \(U\), and the derivative \(\partial u/\partial x\) is proportional to \(U\) and inversely proportional to the body (reference) length, \(L\). Therefore, the inertial force per unit volume (proportional to the density \(\rho\)) scales as:

\[
\rho \frac{u}{L} \frac{\partial u}{\partial x} \approx \rho \frac{U}{L} \frac{U}{L} = \rho U^2
\]

In the same manner, the frictional force per unit volume scales as:

\[
\eta \frac{\partial^2 u}{\partial y^2} \approx \eta \frac{U}{L^2}
\]

This is proportional to the constant coefficient of viscosity (\(\eta\) will be used for the viscosity throughout this document, rather than the traditional \(\mu\)). Then, the ratio of these two competing forces can be written as:

\[
\frac{\text{Inertia}}{\text{Friction}} = \frac{\rho U^2 L}{\eta U L^2} = \frac{\rho U L}{\eta}
\]

Thus, the Reynolds number is a similarity or scaling parameter representing the ratio of inertial to frictional forces in a flow field:

\[
\text{Re} = \frac{\rho U L}{\eta}
\]

For turbomachinery, the Reynolds number is often expressed using \(\omega D\) as the representative velocity (twice the tip speed) and the diameter \(D\) as the representative length scale:
\[
Re = \frac{\rho \omega \cdot D^2}{\eta}
\]

Following the same analysis, the Froude number is a similarity parameter representing the root of the ratio of inertial to gravitational forces per unit mass in a flow field:

\[
\frac{\text{Inertia}}{\text{Gravity}} \approx \frac{\rho U^2 L}{\rho g g L} \approx \frac{U^2}{gL}
\]

\[
Fr = \frac{U}{\sqrt{gL}}
\]

Thus the Froude number is used where the fluid free surface plays a role in the dynamic behavior, such as wave action or hydraulics. This parameter is not convenient in the present study but has been presented to complete the background discussion on similarity parameters.

### 2.2 CENTRIFUGAL PUMP ANALYSIS

Centrifugal pump flow patterns are diverse and complex. Flow through the blade passages shows a small degree of smooth, streamlined behavior, and flow through the volute diffuser shows high levels of turbulence and vorticity. And, despite having been used for more than 1,000 years, centrifugal impellers produce significant secondary flow fields at the discharge plane that are still not thoroughly understood. As a result, 1D methods are still used to achieve preliminary conceptual designs and have proven fairly reliable. Much of the CCF's design of the IVAS pump involved information from trade studies using
a modified version of the Pump-A code at NASA's Glenn Research Center at Lewis Field.

The IVAS pump operates in a unique regime of low specific speed and comparatively low flow rate. Therefore, the issue of secondary flows, which result in a pump efficiency of less than 20%, is a salient feature affecting IVAS performance. This section details those issues of centrifugal pump performance that are germane to the present experimental results and interpretation.

2.2.1 Performance Characteristics

Centrifugal pump functionality is accomplished by imparting angular momentum to the fluid. This is a form of forced vortex and is the mechanism for elevating the energy of the fluid. However, because the pressure increases with radius through the impeller, an adverse gradient is created which can create problems, especially for low-specific-speed, low-flow regimes. In order to understand this specific regime, ideal pump analysis will be presented in conjunction with the issues of prerotation and inlet recirculation, which inhibit the ideal performance of a radial impeller.

2.2.1.1 The Ideal Radial Pump

A generic pump can be modeled after a "black box," in which input power is used to change the fluid energy between inlet and outlet planes. This model is assumed for the following analysis. The inlet flow is referenced as station 1 and is assumed to be aligned with the axial direction, and the discharge
flow is referenced as station 2 and is assumed to be aligned along the outward radial direction. In addition, the ideal radial impeller will have two critical dimensions. The inner, or eye, diameter is designated as $D_1$ and represents the circle on which the blade leading edges lie. The outer, or tip, diameter is designated as $D_2$ and represents the circle on which the blade trailing edges lie. This latter dimension is the maximum diameter of the rotor.

This ideal analysis will cite three reference velocities: $W$, $u$, and $V$. The components $W_1$ and $W_2$ indicate the velocities with respect to the rotating reference frame, or the velocities through the blade passage. The components $u_1$ and $u_2$ indicate the velocities resulting from pure rotation, or the tangential velocities:

$$u_1 = \frac{1}{2} \omega D_1 \quad \text{and} \quad u_2 = \frac{1}{2} \omega D_2$$

Therefore, the absolute velocities with respect to the fixed laboratory reference frame are $V_1$ and $V_2$; these are just the vector sums of $W$ and $u$. Figure 2.1 shows the ideal geometrical relationship of these velocities for inlet and outlet planes.

Figure 2.1  Ideal velocity diagrams for a centrifugal impeller  
[reproduced from Turton 1994, 3]
Two other derived velocities are discussed here. The so-called whirl (or swirl) velocities are designated \( V_{u1} \) and \( V_{u2} \) and indicate the degree of rotation in the flow. The radial components of the absolute velocities are denoted as \( V_{r1} \) and \( V_{r2} \). Blade inlet and outlet angles, relative to the tangent line, are \( \beta_1 \) and \( \beta_2 \).

Required rotor torque is the net increase in angular momentum. As the fluid travels through the impeller, momentum is exchanged between the rotating blades and the fluid. This transport is a function of the impeller geometry and the swirl velocities:

\[
T = \dot{m} \left[ V_{u2} \frac{D_2}{2} - V_{u1} \frac{D_1}{2} \right]
\]

Since the input power is the product of the torque and angular velocity (\( \omega T \)), the energy addition per unit mass results in Euler's equation for an impeller:

\[
gH_E = \frac{\Delta P_{\text{impeller}}}{\dot{Q}} = u_2 V_{u2} - u_1 V_{u1}
\]

This equation assumes zero losses and zero change in hydraulic grade. \( H_E \) is the traditional measurement of pump head in units of meters of water and corresponds to the net pressure increase through the impeller, \( \Delta p \). The conversion is:

\[
gH_E = \frac{\Delta p}{\dot{Q}}
\]

Ideally, the inlet flow will have a zero swirl velocity (also known as preswirl) such that \( V_{u1} = 0 \). Therefore, the velocity triangle on the left of Figure 2.1
will be a right triangle, and the fluid will enter the impeller with a purely radial component (no tangential component). Euler’s equation simplifies to:

$$gH_e = u_2 V_{u2} = u_2 (u_2 - V_{R2} \cot \beta_2) = u_2^2 - \frac{Q \cdot u_2 \cot \beta_2}{D_2 b}$$

Here, the radial velocity is replaced by the combination of net flow rate, Q, and the height of the impeller discharge passage, b. At zero flow (shut valve condition), the pump head rise is proportional to $u_2^2$ and thus $\omega^2$. At positive flow rates, the ideal pump head varies linearly with Q as shown in Figure 2.2.

The real performance for an acute outlet angle as used in the IVAS impeller is a result of flow losses. Note that the overall pump head will be higher than the curve shown, which is representative for the impeller only. Further pressure rise is accomplished through efficient flow deceleration in the volute and diffuser.

The representation of pump head versus flow rate is known as the pump curve.

![Figure 2.2 Ideal specific energy rise for a centrifugal impeller](reproduced from Turton 1994, 5)
2.2.1.2 Prerotation Corrections

Often, it is desirable to correct data for zero prerotation (prespirl) by using data containing a measured degree of preswirl. It is also convenient to shift data that has zero preswirl in order to predict the results for a specified degree of preswirl. This procedure is summarized.

At zero preswirl, the impeller head is equal to \( u_2 \cdot V_{u2} \). Thus, the non-ideal case can be written as a relation of the ideal head rise:

\[
gH_e = gH_{e0} - u_1 V_{u1}
\]

Now, the inlet preswirl velocity can be replaced by a fraction of the impeller velocity using \( f \), which varies between 0 and 1. The equation yields:

\[
V_{u1} = f \cdot u_1 \text{ so that } gH_e = gH_{e0} - f \cdot u_1^2
\]

Thus, the corrected pump pressure rise for non-zero preswirl involves the specific gravity, \( g_{sp} \), the impeller speed, and the diameter. It is written as:

\[
\Delta p_{corr} = \Delta p_0 - f \cdot u_2^2 = \Delta p_0 - C_g \cdot g_{sp} \cdot f \cdot \text{RPM}^2 \cdot D^2
\]

The constant \( C_g \) is defined for common units used (D in cm) and is a function of the impeller geometry. It is based on the large-scale dimensions of the LSRT:

\[
C_g = \begin{cases} 
1.7653 \times 10^7 & \text{(impeller #3386)} \\
2.8455 \times 10^7 & \text{(impeller #3452)}
\end{cases}
\]

2.2.1.3 Inlet Recirculation

Inlet recirculation can result from the strong adverse pressure gradient through the impeller at low-specific-speed, low-flow regimes. Shown
schematically in Figure 2.3, inlet recirculation has been described by some as an inverted Ekman layer (Nakamura and Ding 1999). An Ekman layer is a low-momentum boundary layer flow that is exposed to an external rotating velocity field. It is often used to describe surface streamlines in planetary atmospheric boundary layers. The behavior of the housing surface flow depends on the clearance of the rotating impeller with respect to the stationary wall (endwall). If the blade endwall clearance is small, the Ekman layer is compressed and the surface streamlines will be heavily influenced by the strong tangential gradient produced by the passing rotor blades. A larger endwall clearance will result in a thicker Ekman layer. In this latter scenario, the surface streamlines will experience a greater influence by the adverse radial pressure gradient and may spiral inward toward the inlet. Impeller efficiency can be sharply reduced.

Figure 2.3  Recirculation in the inlet region of a centrifugal impeller [reproduced from Kittredge 1976, 2.173]
2.2.2 Pump Performance Parameters

There are four parameters that influence centrifugal pump performance: flow coefficient, pressure coefficient, Reynolds number, and specific speed. These four parameters are dimensionless groupings that result from applying the principle of dimensional analysis, as was done in Section 2.1, to develop the concept of dynamic similarity. Two of these, the flow and pressure coefficients, are non-dimensionalized forms of the pump flow rate and pump pressure rise. They are defined by the following equations:

\[ \varphi = \frac{\dot{V}}{\omega R^3} = C_1 \frac{Q}{(RPM)R^3} \]

\[ \psi = \frac{\Delta p}{\frac{1}{2} \rho \omega^2 R^2} = C_2 \frac{H}{g_sp(RPM)^2 R^2} \]

The third parameter is the Reynolds number. It is represented as:

\[ \text{Re} = \frac{\rho \omega \cdot D^2}{\eta} = C_5 \frac{g_sp(RPM)D^2}{\eta} \]

These coefficients are useful for plotting the similarity behavior of a pump, as demonstrated by Figure 2.4. The conventional curves shown on the left are dimensional plots of impeller head versus flow rate that reveal separate curves for each rotational speed. On the right, the dimensionless data collapse onto a common curve of pressure coefficient versus flow coefficient. This is the universal behavior of the similarity law for a centrifugal pump.
Based on the similarity relationships, scaling laws can be defined so that model pumps can be used to predict true-scale performance. Thus, multiple designs can be derived from a single set of performance data by applying the scaling laws. Scaling must be applied with care, however, because there are several factors such as viscous losses that can affect the results. The most significant difference comes with a shift of the peak efficiency. As a pump is scaled down, the point of best efficiency shifts to a lower relative flow rate (Kittredge 1976, 2.136; Turton 1994, 8). This error, though small, can cause over-predicted efficiency when using a larger-scale model. More importantly, the error can cause under-predicted efficiency when using a smaller-scale model.

If the flow and pressure coefficients are referenced to the point of best efficiency, the resulting similarity parameter is the specific speed, also called the characteristic number or the non-dimensional speed number. The specific speed

Figure 2.4  Construction of similarity law from pump curves
encompasses overall dynamic similarity and can be written as a function of both
the flow and pressure parameters:

$$v_{sp} = \frac{\rho^{12}}{\psi^{3/4}} = \omega \sqrt{V} \cdot \left( \frac{\rho}{\Delta p} \right)^{3/4} = C_4 \cdot \text{RPM} \sqrt{Q} \cdot \left( \frac{g_{sp}}{H} \right)^{3/4}$$

Therefore, the flow and pressure coefficients are both a function of the specific
speed, which can singularly define the dynamic scale. The largest contribution
to pump power losses comes from phenomena that are functions of the specific
speed. These include disk friction losses, leakage losses, and casing hydraulic
losses, and can constitute 60-90% of total pump losses. Thus, the specific speed
also affects the point of best efficiency. Kittredge (1976, 2.134) provides a
breakdown of the total pump losses at peak efficiency and demonstrates that
minimum losses occur in the range:

$$0.73 < v_{sp} < 1.10.$$

The constants for each of these similarity parameters are given such that
the coefficients yield dimensionless forms when pump flow rate Q is in LPM,
impeller dimensions R and D are in cm, pump head H is in mmHg, and fluid
viscosity \( \eta \) is in cP. These are the common engineering units used throughout
this document. The specific gravity \( g_{sp} \) is itself dimensionless. Thus:

$$C_1 = 1.5915 \times 10^2$$

$$C_2 = 1.2158 \times 10^5$$

$$C_4 = 1.9376 \times 10^{-3}$$
2.3 NON-NEWTONIAN FLUID MECHANICS

An elastic solid generally follows the relationship of Hooke’s law, which establishes the proportionality between shear stress $\tau$ and strain $\gamma$. A Hookean solid will have a constant modulus of elasticity $G$ and thus a linear relationship:

$$\tau = G\gamma$$

This is an idealization of elastic solid deformation.

On the other hand, Newton’s law of friction describes the idealization of inelastic fluid behavior as assumed in classical fluid mechanics. For a Newtonian fluid, the stress is proportional to the deformation rate rather than the deformation:

$$\tau = \eta \dot{\gamma}$$

Here, the constant of proportionality is the coefficient of viscosity $\eta$. The strain rate is represented by $\dot{\gamma}$.

The study of non-Newtonian fluids, which are viscoelastic materials that lie somewhere between Hookean solids (elastic) and Newtonian fluids (inelastic), is the basis for the field of rheology. As a result, the non-Newtonian shear viscosity may be a highly nonlinear function of shear stress, strain rate, or both. Thus, in general:

$$\tau = \eta(\tau, \dot{\gamma})\dot{\gamma}$$
The relationships of shear stress to strain rate of some types of viscoelastic materials are shown in Figure 2.5. Note that these plots are representative for the specified fluids. The two constant viscosity fluids are the idealized Bingham plastic and the Newtonian fluid; the chief difference is the presence of a yield stress. A Bingham plastic behaves as a Hookean (or neo-Hookean) solid when the stress is below its yield point (resisting flow), yet flows as a Newtonian fluid when the stress is above it. The materials exhibiting variable viscosity are the dilatant and pseudoplastic fluids. These are also known as shear-thickening or shear-thinning fluids, respectively, based on how the shear stress, and thus the viscosity, changes with increasing strain rates.

![Figure 2.5](image_url)  

**Figure 2.5** Rheological behavior of some viscous materials
Having established the basic definition of a non-Newtonian fluid, the remainder of Section 2.3 will address the importance of conducting non-Newtonian fluid/flow studies, some important non-linear viscoelastic considerations, the use of shear viscosity modeling, and some results of non-Newtonian boundary layer analysis.

2.3.1 The Importance of Conducting Non-Newtonian Studies

Blood is a non-Newtonian fluid with a prominent pseudoplastic region below strain rates of 100 sec\(^{-1}\) when exposed to a pure shear flow field (refer to Figure 2.12 on page 54). At higher strain rates, blood is intrinsically Newtonian in response to shearing (Mann and Tarbell 1990, 711). By using the Poiseuille flow analysis, and assuming common arterial dimensions and flow rates, the average circulatory strain rates are greater than 100 sec\(^{-1}\). Therefore, the non-Newtonian character of blood has been traditionally ignored for many studies. This assumption is also common in the study of blood handling devices such as blood pumps, heart valves, and oxygenators, where researchers routinely report calculations performed with a constant viscosity assumption or present flow studies carried out with "blood analogs," such as glycerin/water, that are Newtonian in behavior (Massiello et al. 1994; Fontaine et al. 1996; and Z. Wu et al. 1996).

However, flow fields such as those around a hydrodynamic heart valve leaflet or through a rotodynamic LVAD pump reveal a greater complexity than
pulsatile tube flows. In the former, flow is complicated by stagnation (high shear rates), attached boundary layer development (moderate shear rates), and areas of separation and recirculation (low shear rates). Because the traditional topics of concern of hemolysis (rupture of erythrocytes) and thrombosis (clot formation) are linked to the mechanisms of stagnation and separation, further study into the impact of non-Newtonian fluid mechanics is warranted. In other words, some of the bothersome fluid-dynamic issues affecting blood handling devices occur at lower shear rates, where non-Newtonian effects may also occur. Despite these facts, it is still common to assume non-Newtonian effects to be of second-order importance. This section will outline some of the biomedical and industrial applications of non-Newtonian fluid mechanics studies.

2.3.1.1 Biomedical Studies

Non-Newtonian flow fields have been studied both analytically and experimentally. Fung (1993, 87-91) discusses simple tube flow, showing how a non-Newtonian viscosity blunts the velocity profile and increases wall shear stress. Mann et al. (1987) experimentally studied a ventricular assist device and showed that a non-Newtonian solution yielded different results than a water/glycerol solution. Yet Mann and Tarbell (1990) studied flows in rigid curved and straight artery models and found that the significance of the viscosity characteristic depended on the test condition. Pohl et al. (1996) studied the Bjork-Shiley valve with Newtonian and non-Newtonian fluids and found
that the viscosity affected some performance characteristics but not others. It is
not obvious that the same relationships would exist for other valves. Friedman
(1993) and Dutta (1996) have incorporated non-Newtonian viscosity into the
study of arterial flow. Many other similar biomedical studies presented in the
literature have been motivated by the design of blood-handling devices.

2.3.1.2 Other Related Studies

The bioengineering community is not alone in making an effort to better
understand the subtleties of non-Newtonian fluid mechanics. The
Commonwealth Scientific and Industrial Research Organisation (CSIRO) of
Australia has been using state-of-the-art flow visualization and analysis
methods to study the non-Newtonian effects on slurries in mixing agitators (La
Fontaine 1996). Similar studies focused on pipe flows of non-Newtonian slurries
at Southwest Research Institute (Park et al. 1989). The Indian Institute of
Technology made generalized correlations for predicting the friction loss in pipe
bends for a pseudoplastic fluid (Das et al. 1991). Imperial College in London has
made measurements on non-Newtonian polymer solutions to support the
recycling of polyvinyl chloride (Savvas et al. 1994). CSIRO has investigated the
fluid dynamics of a C4 airfoil in a viscoplastic shear-thinning fluid (Sheridan et
al. 1996). The University of Pittsburgh performed laser Doppler velocimetry
(LDV) measurements on a corrugated channel, suggesting a significance to
polymer processing, contaminant seepage, and oil recovery in addition to the flow of biological fluids (Rajagopal 1993).

2.3.2 Nonlinear Viscoelastic Effects

A viscoelastic fluid is exhibited in one or more of the following measured properties: shear thinning, shear thickening, time-dependence, and normal stresses in shearing. The first two behaviors are usually strongly linked, as are the last two. However, there is a weak connection between the first two and the last two, and it is possible for a fluid to be nearly inelastic, but have a strong shear-thinning characteristic. It is also possible for a fluid to be elastic, but have a constant viscosity; these are known as Boger fluids. Based on the relevancy to blood-like behavior, only the first and last fluid property and related phenomena will be discussed in the following pages.

2.3.2.1 Shear Thinning

The non-constant shear viscosity is the most salient characteristic of a non-Newtonian fluid, and the most common type of viscosity variation is a shear-thinning behavior. In many shear-thinning cases, the viscosity may decrease an order of magnitude or more. Figure 2.6 shows the nature of this phenomenon for a xanthan gum solution with an overall viscosity decrease of about 200:1 in a pure shear (i.e., Couette) flow field.

Two asymptotic regions are apparent in the viscosity curve, one at low shear rates (below $3 \times 10^1$) and one at high shear rates (above $3 \times 10^3$). These are
Known as the lower and upper Newtonian plateaus, respectively, because as zero or infinite shear rates are approached, the fluid viscosity behaves in a Newtonian fashion. In such specific flow regimes, even a strong shear-thinning fluid may respond exactly as a Newtonian one. The quantities $\eta_0$ and $\eta_\infty$ refer to the viscosity at these Newtonian plateaus, respectively, and are also known as the zero-shear and infinite-shear viscosities.

![Graph](image)

**Figure 2.6** Non-Newtonian properties of a 0.2% xanthan gum solution [generated using data from Escudier et al. 1999, 202]

Shear thinning is most prominent in macromolecular fluids (polymer solutions and a few naturally occurring fluids such as blood). With these types of fluids, higher rates of shearing will tend to align the molecules with the flow direction and untangle the polymer chains. This effect is more pronounced for
larger-molecular-weight polymers. Thus at higher shear rates, polymer solutions flow at proportionately lower stresses. This very property has been exploited to reduce the drag and pumping power for transporting fluids through pipelines (Escudier et al. 1999; Chang and Darby 1983).

2.3.2.2 Normal Stress Differences

One result of the elasticity of a fluid is the presence of normal stress differences. Normal stresses can affect extrudate die swell and hole pressure error. The notation here will be referenced to a simple shear flow with $x$ designating the flow direction, $y$ designating the transverse direction (perpendicular to the planes of constant-shear), and $z$ designating the neutral direction. By this notation, the shear stress would be represented as $\tau_{xy}$. The normal stress differences are now defined as:

$$N_1 = \tau_{xx} - \tau_{yy}$$

$$N_2 = \tau_{yy} - \tau_{zz}$$

The first difference, always a positive value, is larger and more influential than the second difference. Isotropic materials will usually have the ratio $(N_1/N_2)$ in the range $-20$ to $-3$ (cited in Macosko 1994, 138). The second difference is rarely important in hydrodynamic calculations and is often ignored because of its magnitude relative to $N_1$. In addition, reliable experimental measurements of $N_2$ have proven to be extremely difficult to obtain.
The dashed curve in Figure 2.6 shows one example of the increasing normal stress produced by the fluid, which can often exceed the shear stress of the fluid. Escudier et al. (1999, 202) corroborated the power-law behavior (with respect to stress) of dilute polymer solutions, such that for this fluid the normal stress difference is:

\[ N_1 = 0.97 \tau^{1.47} \]

As a point of comparison, the curve shows a viscosity of 3.4 cP at 10,000 sec\(^{-1}\). This results in a maximum shear stress of 33.6 Pa and a normal stress difference of 170.3 Pa.

2.3.2.3 Hole Pressure Error

The tension along streamlines created by the normal stress differences in a viscoelastic fluid may give rise to measurable pressure deviations at free-surface streamlines. This is particularly significant for measurements made through wall pressure taps using recessed transducers, which are susceptible to the so-called hole pressure errors. As shown in Figure 2.7, tension along a streamline near the wall will tend to lift the fluid out of the hole and decrease the indicated sensor pressure. This error is not present in Newtonian or inelastic non-Newtonian fluids.
The true pressure at the wall is written as the sum of the measured pressure, $p_m$, and the hole pressure error, $p_h$:

$$P = p_m + p_h$$

However, the hole pressure error is itself a function of the normal stress differences:

$$3mp_h = N_1 - N_2 \approx N_1$$

The value of $m$ comes from the slope of the log-log plot of $p_h$ vs. $\tau_w$ and is approximately constant. A constant slope $m$ means that the hole pressure error is proportional to the first normal stress difference, so that:

$$m = \frac{d(\ln p_h)}{d(\ln \tau_w)} \approx \frac{d(\ln N_1)}{d(\ln \tau_w)}$$
Using the values for the dashed curve in Figure 2.6, the value for m is 1.47, based on the power law equation for $N_\lambda$, and the expression for the hole pressure error is:

$$p_h = 0.22 \cdot \tau^{1.47}$$

At a strain rate of 10,000 sec$^{-1}$, the hole pressure error is 38.6 Pa, less than 0.04% of standard atmospheric pressure. Thus, such solutions exhibit weakly elastic behavior and produce negligible hole pressure errors.

2.3.2.4 The Weissenberg Number

One way to gauge a fluid's degree of elasticity, which is not necessarily related to its degree of non-Newtonian character, is by the Weissenberg number or the Deborah number. In general, the Deborah number is defined as the ratio of a material's characteristic relaxation time, $\lambda$, to the time of observation (or flow residence time), $t$. This ratio is important because it describes the relative importance of elasticity in a given flow regime. Therefore it can be defined as:

$$De = \frac{\lambda}{t}$$

For many flow situations, the time of observation is related to the inverse of the shear rate or the length-velocity ratio:

$$De = \lambda \gamma = \lambda \frac{V}{L}$$

Molecular relaxation is a viscoelastic effect that is negligible for Newtonian fluids and non-Newtonian inelastic fluids (for gases, $\lambda < 10^{-10}$). Thus, a
very low Deborah number would indicate a short relaxation time and a smaller influence of viscoelastic effects. Likewise, a higher Deborah number would warn of more prominent viscoelastic influences. A Deborah number of unity marks a fundamental change in the material's strain response, as will be demonstrated.

With a relaxation time on the order of 10 seconds, Silly Putty® may sometimes have a low Deborah number (De<<1) and sometimes a high Deborah number (De>>1). When allowed to sit for an hour or more, the putty will slowly spread out, flowing like a liquid over a large time scale on the order of 10^3 seconds (De=0.01). If stretched very quickly, on the order of 10^1 seconds, the putty will fissure like a solid material (De=100). Thus, the main importance of the Deborah number is to categorize a viscoelastic material's behavior as liquid- or solid-like based on the relaxation time and time scale, not solely on the material's state properties.

The Weissenberg number is often different in form but fundamentally the same as the Deborah number in describing a fluid's viscoelastic qualities. It is convenient to use the Weissenberg number when working with polymer solutions, where it is desired to represent the relaxation time with easily measured fluid properties. Dilute polymer solutions of the type used in this study can be modeled using the bead-spring kinematic model, in which case the relaxation time constant is proportional to the zero-shear viscosity and inversely proportional to the weight concentration of the solution (Macosko 1994, 496).
The following formula is thus a dimensional representation of the fluid's relaxation time in seconds:* 

\[ \lambda = \frac{\eta_0}{\tau \cdot C} = C_7 \frac{\eta_0}{\tau \cdot \text{PPM}} \]

\[ C_7 = 10^3 \]

If the 2,000 PPM xanthan gum solution of Figure 2.6 were considered, the reference stress would be 30 Pa and \( \lambda \) would be about 9.6 seconds.

The form for the Weissenberg number used here will be the same Rubart used (1992, 667) for axisymmetric flow in a cylindrical container with a spinning lid. Flow residence time is taken as the inverse of the rotational speed so that:

\[ \text{We} = \lambda \omega = \frac{\eta_0 \omega}{\tau \cdot C} \]

In this definition, the term \( \tau \) is the constant reference stress in Pa used to obtain the dimensionless shear rate for the viscosity master curve. The concentration is denoted as \( C \) in dimensionless weight fraction. In general, high Weissenberg values correspond with high impeller speeds and thus high dimensionless shear rates on the master curve (described in more detail in Section 2.3.3.2).

This form of the Weissenberg number is justified in that it is used to represent a rotating flow field, essentially a forced vortex, within an axisymmetric boundary. A centrifugal pump impeller induces a strong vortical

* NOTE: This does not actually indicate the fluid relaxation time, just a proportional representation of it for comparison purposes.
flow in an almost axisymmetric boundary. Even though the impeller discharges into an asymmetric volute, the pressure boundary condition is constant around its circumference (if the volute is properly designed). And, although there is outflow from a centrifugal impeller, the majority of the flow field is composed of eddies and recirculation zones. Nakamura and Ding (1999) showed this is especially true of the IVAS impeller designs.

Some computational results from Rubart’s problem are shown in Figure 2.8. These images show the halfplane streamlines resulting from the forced vortex inside the cylindrical container at a Reynolds number of 1,247. The left edge of each frame is the axis of symmetry (cylinder centerline) and the right edge is aligned with the wall. Thus, no-slip is enforced at the wall and the lower surface; the upper surface is rotating around the axis of symmetry. These results
suggest that the effect of heightened Weissenberg values is to compress the core flow such that the circulation streamlines become more localized near the wall. This effect would produce a stronger radial pressure gradient, which is revealed by the enhanced low-pressure "bubble" formation at the line of symmetry. Higher pressures at the periphery will correspond with lower pressures at the center, thus creating larger bubbles. Overall, these images suggest that a departure from inelastic fluid properties as gauged by the Weissenberg number has a progressive impact on the flow-field qualities.

2.3.3 Viscosity Fluid Models

It is desirable to analytically formulate the nonlinear fluid effects. In presenting some viscosity models for shear-thinning fluids, two aspects relating to polymer solutions will be presented. The condition of a dilute solution, which affects many other fluid assumptions, will be defined. This condition impacts the viscosity similarity behavior, which will follow. Then, power law and multiple-parameter models will be presented to effectuate shear-thinning fluid behavior.

The viscosity models will assume a pure-shear flow. Thus the full 3D generalized stress tensor will not be considered so that only the shear viscosity will be represented. For a fluid in pure shear, the strain rate is expressed as the velocity gradient normal to the flow direction, or \( \dot{\gamma} = \frac{\partial u}{\partial y} \).
2.3.3.1 Requirements for a Dilute Solution

As presented in Sections 2.3.2.2 and 2.3.2.3, the normal stress differences and thus the hole pressure error for dilute polymer solutions can be expressed as a power law in terms of shear stress. Although the power law for stresses is the most important, many of the relationships for polymer solution properties presented in this chapter are dependent on the fluid being a dilute solution. As long as the intrinsic viscosity varies linearly with the concentration C, the solution is considered dilute and the power law for stresses is an accurate assumption. The intrinsic viscosity is written:

\[ \frac{\eta_s - \eta_s}{\eta_s C} = \alpha C + \beta \]

The Newtonian viscosity of the solvent is denoted by \( \eta_s \), and \( \alpha \) and \( \beta \) are the constants for the linear relationship. As intermolecular forces and polymer chain entanglement become increasingly prominent, the fluid will depart from the linear relationship and will no longer be considered a dilute solution.

2.3.3.2 Viscosity Similarity

As long as the polymer solutions are considered dilute, the fluid will conform to a viscosity similarity law or master curve. Similarity relations that are applicable to dilute polymer solutions have been developed for both temperature and concentration. An example of a viscosity master curve for solution concentration is shown in Figure 2.9.
Temperature greatly affects the viscosity of a fluid. All liquids will decrease in viscosity if the temperature is increased (or the pressure decreased, but this effect is much less pronounced). Viscosity curves can be shifted to some reference temperature, designated by the subscript \( r \), by calculating a shift factor \( a_T \). The glass transition temperature of a polymer is commonly used as the standard reference. Shift factors are calculated at each temperature using the ratio of zero-shear viscosities:

\[
a_T(T) = \frac{\eta_0(T)}{\eta_0(T_r)}
\]

Then the shifted strain rate and viscosity can be calculated at the reference temperature:

\[
\dot{\gamma}(T_r) = \dot{\gamma} \cdot a_T \quad \text{and} \quad \eta(T_r) = \frac{\eta}{a_T}
\]

Using this method (Koelling 1996), fluid properties can be measured at a variation of temperatures to yield a standardized, dimensional viscosity curve.

Calculation of the solution concentration master curve involves scaling strain rate and viscosity in a dimensionless correlation. These normalized values are the points plotted in Figure 2.9. The normalized strain rate and viscosity are, respectively:

\[
\dot{\gamma} \cdot \frac{\eta_0 - \eta}{\tau \cdot C} \quad \text{and} \quad \frac{\eta - \eta_\infty}{\eta_0 - \eta_\infty}
\]

\( C \) is the concentration and \( \tau \) is the reference shear stress, a fixed value for all
concentrations. Some choices for the reference stress are the demarcation of the beginning of the pseudoplastic regime, the end of this regime, or the point at which the viscosity is the average of $\eta_0$ and $\eta_\infty$. Defining these demarcations can be challenging, so threshold values can be set at 2% below $\eta_0$ or 2% above $\eta_\infty$, for example. For dilute polymer solutions, the reference stress is independent of concentration.

![Normalized Viscosity vs. Normalized Shear Rate](reproduced from Rubart 1992, 666)

**Figure 2.9** Viscosity master curve for a shear-thinning solution

2.3.3.3 The Power Law Model

Although there are multiple analytic viscosity models, the power law model is classically used to enable exact solutions of the governing flow equations. This is the simplest non-Newtonian viscosity model that is often used
in the literature and is defined by a power law relationship between shear, \( \tau \), and strain rate, \( \dot{\gamma} \):

\[
\tau = m \dot{\gamma}^n \quad \text{and so} \quad \eta = m \dot{\gamma}^{n-1}
\]

Here, the slope of the power curve is denoted by \( n \) and the proportionality constant by \( m \). This model is excellent for flows exclusively in the pseudoplastic regime, where \( n < 1 \) would indicate a shear-thinning fluid. However, the model predicts an undefined viscosity at zero shear rate when \( n < 1 \). Thus, the power law assumption may be valid near the stagnation point (e.g., early in the boundary layer development where shear rates are moderate in magnitude) or in a constantly accelerating flow. But near the point of flow separation, where the wall shear vanishes as the flow streamlines leave the surface, the viscosity predictions are unacceptably large. Similarly, the viscosity predictions approach zero as the shear rate becomes large, and the modeled fluid takes on an inviscid behavior. Therefore, a power law fluid model is inadequate for a flow regime that involves low shear rates because the analytic value of the viscosity approaches infinity. On the other hand, flows involving high shear rates are similarly deficient because the viscosity approaches zero. The only situations that will produce good numerical agreement with a power law fluid model are accelerating flows with non-zero wall shear, such as developing pipe or channel flows. Similar problems arise for fluids with shear-thickening characteristics, that is, \( n > 1 \).
2.3.3.4 Multiple-Parameter Models

For most practical flow situations, the power law will give non-physical solutions because it is only accurate when used in strictly accelerating flow fields, an example being a fully developed arterial flow. Because of these very restrictive circumstances, more involved viscosity models are required. In rotary blood pump impellers, the relative velocity of blood to vane surface usually decreases significantly from inlet to outlet, and varying velocity magnitudes precipitate varying shear rates. Other viscosity models may seem more realistic, but they involve increasing complexity.

The following three models describe the non-Newtonian shear-thinning characteristic similar to that of the power law model, but also allow for the upper and lower Newtonian plateaus. All of these models will represent shear viscosity as a function of the strain rate, \( \dot{\gamma} \). The three-parameter Ellis model is the simplest. It is based on the zero-shear viscosity, \( \eta_0 \), a threshold constant, \( K \), and a power-law slope, \( a \). It is simple enough to allow analytic solution of some complex flow problems without the need for computational solutions:

\[
\frac{\eta_0}{\eta} = 1 + \left( \frac{\dot{\gamma}^2}{K} \right)^{a-1}
\]

The common Cross model takes a four-parameter form involving a power-law slope, \( n \), a relaxation factor, \( K \), and the upper and lower Newtonian viscosity values, \( \eta_0 \) and \( \eta_\infty \):
\[
\frac{\eta - \eta_\infty}{\eta_0 - \eta_\infty} = \frac{1}{1 + (K\gamma)^{-n}}
\]

The shear viscosity curve in Figure 2.6 has been plotted using a Cross model curvefit. The following parameters were used: \(\eta_0=578\ \text{cP}, \eta_\infty=2.76\ \text{cP}, K=1.3\ \text{sec},\) and \(n=0.276\). This model is used for curve fitting data throughout this study.

Another model, the five-parameter Yasuda model, is similar to the Cross model:

\[
\frac{\eta - \eta_\infty}{\eta_0 - \eta_\infty} = \left[1 + (K\gamma)^a\right]^{-\frac{1}{a}}
\]

When \(a=2\), the form reduces to the four-parameter Carreau model. The four-parameter models are the most common used for curve-fitting data or simple computational solutions.

Some other models that can simulate non-Newtonian behavior are the viscoplastic models that assume a yield stress. These curvefits, such as the Bingham or Casson models, model a discrete yield point and then Newtonian deformation or hybrid pseudoplastic behavior that closely resembles a power law fluid. Still other, more accurate models take the full 3D stress tensor into consideration, but they are beyond the scope of this discussion. Thus, it would be ideal if a simpler, analytic viscosity model were used (Ellis, Cross, or Yasuda), rather than more complex models which would require involved numerical solutions for each value of shear rate, pressure, and/or temperature.
2.3.4 Non-Newtonian Boundary Layer Analysis

To illustrate the effect of non-Newtonian fluids on boundary layer characteristics, Bodonyi (1997) numerically considered the viscous flow over a circular cylinder for two viscosity laws. Such a geometry models the flow near the leading edge of an airfoil or hydrofoil cross-section by simulating the stagnation point boundary layer development. The boundary layer equations were solved numerically using a Newton linearization technique. A more detailed treatment of the numerical method is presented in Appendix B.

To obtain the results for a cylinder, the following non-dimensional pressure distribution was imposed:

\[ p(x) = -\ln[2\sin(x)] \]

This distribution is a result from inviscid irrotational flow theory. The 2D boundary layer calculations were performed for a Newtonian fluid (having a constant viscosity) and a non-Newtonian hybrid fluid (modeling a shear-thinning fluid). These two fluids are represented by the normalized viscosity functions:

\[ \eta_\bullet = \alpha \]

\[ \eta_\bullet = 1 + \beta \left[ 1 + \epsilon \left( \frac{\partial u}{\partial y} \right)^2 \right]^\frac{n-1}{2} \]
In the two viscosity models, $\alpha$, $\beta$, $\epsilon$, and $n$ are suitably chosen constants to simulate the desired blood-like viscosity characterization ($\alpha=1.000$, $\beta=5.271$, $\epsilon=0.0246$, and $n=0.6825$).

The second equation is a form of the Carreau model having four dimensional parameters. In the dimensionless scaling employed for numerical methods, these constants can be represented as:

$$\eta^* = \frac{\eta}{\eta_\infty}$$

$$\beta = \frac{\eta_0}{\eta_\infty} - 1$$

$$\epsilon = \lambda^2$$

Thus, these parameters are consistent with the Carreau formulation as presented in Section 2.3.3.4 ($\eta_0=6.271$ cP, $\eta_\infty=1$ cP, $\lambda=0.1569$ sec, $n=0.6825$).

As shown in Figure 2.10 and Figure 2.11, non-Newtonian fluids can have a significant effect on the characteristics of the boundary layer flow. The dimensionless boundary layer thickness and the dimensionless skin friction (and thus the wall shear) are significantly increased over those given by the usual Newtonian fluid model. These figures show about a 250% increase over the Newtonian case, despite the fact that the incompressible flow equations have been scaled by the Reynolds Number.
Figure 2.10  A comparison of 2D boundary layer thickness

Figure 2.11  A comparison of 2D boundary layer skin friction
Thus, the use of a shear-thinning fluid in these calculations causes the Newtonian similarity law to fail. This is due to the disparate diffusion length scales manifest in the different fluids. It is expected that the magnitude of this discrepancy is a function of the Weissenberg number such that increasingly viscoelastic fluids will further depart from the Newtonian similarity behavior.

It should also be noted that the separation point, defined as the downstream point in Figure 2.11 where the wall shear and thus the skin friction vanish at the boundary, is not significantly altered. It appears to occur at a dimensionless surface distance of 1.86 for both fluids, which is approximately 107° from the stagnation point (a Blasius series solution would predict separation at 108.8°). In these figures, the dimensionless streamwise coordinate is scaled by the body length, in this case the diameter of the cylinder. The boundary layer thickness has been non-dimensionalized by the body length and the Reynolds number term, $\sqrt{Re}$. The skin friction has been non-dimensionalized by the velocity head pressure and $\sqrt{Re}$. This is the same scaling that results in a boundary layer similarity law for Newtonian fluids.

2.4 BLOOD FLUID DYNAMICS

Blood is a complex fluid that makes experimental measurement challenging. The major component of blood, plasma, is essentially an aqueous fluid that flows as a homogeneous Newtonian fluid. This suggests blood is classifiable as a solution. But whole blood is a cellular aggregation of
erythrocytes, leukocytes, and platelets, which would seem to best classify it as a solid suspension. On the other hand, the red blood cells are pliant and easily ruptured, therefore acting similarly to the behavior of many gel materials.

This section will address some of the unique aspects of blood flows, namely the characterization of its shear viscosity, blood fluid qualities, and blood substitutes for experimental flow studies.

2.4.1 Blood Characterization

Continuous attempts are being made to characterize blood's viscosity experimentally. It is generally accepted that, because leukocytes and platelets occupy a small volume fraction and plasma is Newtonian, the macroscopic rheology of blood is determined by the presence of erythrocytes (Cokelet 1987, 14.1). Studies have shown that the flow behavior of blood is strongly dependent on the hematocrit, or volume fraction of red blood cells (i.e. erythrocytes). Normal adult hematocrit is in the range of 25-50% solids.

A summary of human blood viscosity measurement as presented in the literature is shown in Figure 2.12 (Liepsch and Moravec 1984, 575; Thurston 1989, 519; citation in Macosko 1994, 86; and Banerjee et al. 1991, 103-104). It can be clearly seen that blood viscosity generally increases with hematocrit. The data represented by Banerjee et al. is itself a compilation of human blood viscometry for hematocrits from 33-45%. The figure shows how this trend bisects the span of data. It is interesting to note that the elastic and inelastic components of

![Blood Viscosity Characterization](image)

**Figure 2.12** Blood viscosity characterization

### 2.4.2 Blood Fluid Qualities

It is well known that blood is a non-Newtonian fluid. That is, blood viscosity is not a simple, constant ratio of shear stress to strain rate. This aspect of blood’s flow properties results primarily from the tendency of red cells to aggregate locally, reducing the fluidity in that region. Plasma, the non-cellular component of blood, closely resembles a Newtonian fluid of lower viscosity than blood (Fung 1993, 66-67). Whole blood has a marked shear-thinning characteristic with upper and lower asymptotic Newtonian limits (with some
caveats to be noted below). It is a subject of debate as to whether blood has a finite yield stress to initiate flow at low strain rates (Evans 1992), such that its behavior is like an ideal Bingham plastic with subsequent pseudoplastic deformation. Nevertheless, blood is widely accepted to be a weakly elastic fluid.

The anomalous flow behavior of blood most commonly recognized is the "Fahreus-Lindquist effect" occurring in a range of small-diameter flow passages (Cokelet 1987, 14.12; Fung 1993, 172-186). This has been attributed to a number of effects that relate to the heterogeneity of whole blood. A plasma layer against the surface causes the viscosity to be apparently lower in value when calculated from pressure differential and flow rate than would be measured in a viscometer. When the tube diameter falls to red cell dimensions, the apparent viscosity rises.

A potential difficulty in performing simulated hemodynamic flow studies is the generally used assumption that blood is a homogeneous fluid rather than a heterogeneous one. The cells and their aggregating/disaggregating behavior are not modeled due to complex constitutive equations and lack of a feasible experimental fluid. The "plasma skimming" behavior of whole blood is hypothesized to create a low viscosity, cell-free plasma zone immediately adjacent to surfaces. However, at normal hematocrits, the thickness of this cell-free layer is believed to be very thin compared to the boundary layer dimension (Fung 1993, 91-92) and therefore can be neglected for the purposes of this study.
Blood Usage in Experimental Flow Studies

While using small amounts of blood (< 1L) for viscosity characterization has proven effective, it is unrealistic to expect good results from using large quantities of whole blood (> 100L) for flow studies. There are three main aspects that make standardization of blood difficult and experimental repeatability almost impossible.

First, osmotic pressure dramatically affects the rheology of the red blood cells. Osmotic pressure is an indicator of the chemical potential of water in the plasma and thus affects the transport across the cell membranes. Water easily passes through the membranes and can promote rapid changes in the size and morphology of the erythrocytes, which in turn influences the rheology.

Second, several factors affect aggregation of the red blood cells. The macromolecules present in plasma, such as fibrinogen and globulins, actually cause erythrocyte aggregation, although this can be reversed with an adequate applied shear stress. The degree to which this occurs is a function of the ionic environment of the plasma (Cokeley 1987, 14.3).

Third, the life cycle of the red blood cell affects the composition of blood. The erythrocyte life span in the body is only about 120 days, and the cell goes through age-dependent changes. This results in varying red blood cell density. Also, metabolic depletion causes a decrease in cell volume that proceeds to the point of transforming it into a crenated shape less likely to promote transport across the cell membrane.
Therefore, reproducing the properties of a large volume of blood for controlled flow studies can be very difficult. This is especially true for blood that has been in storage and has become “outdated” with respect to its fluid properties. Moreover, blood immediately begins the coagulation process via the action of the platelets when exposed to air. Thus, any flow studies would necessitate adding anti-coagulants and blood thinners, which would further change the rheological characteristics.

2.4.3 Blood Analog Fluids

To circumvent the difficulties in working with large quantities of whole blood, “analog” fluids are used as a substitute. Many researchers have reported using a variety of polymer solutions that attempt to match blood density, viscosity, and elasticity. This section will summarize some of the fluids that are most applicable to large-scale fluid dynamics and flow visualization studies.

In an effort to match the viscosity of blood, Mann et al. (1987, 141) reported on using an 800 ppm polyacrylamide (PAA) solution of Separan® in water. This type of solution required 4% isopropanol to prevent viscosity degradation, but was still only useful for 4-5 days in flow experiments conducted in the Penn State Electric VAD. PAA solutions tend to be highly elastic, much more so than blood (see also Liepsch and Moravec 1984 and Moravec and Liepsch 1983).
Three fluids of low elasticity have been used to model blood-like viscosity. A 0.25% aqueous solution of sodium carboxymethylcellulose (CMC) has been used to study drag reduction in turbulent pipe flows (Escudier et al. 1999, 201). The viscosities obtained were in the same range as blood. Jeroen et al. (1998, 531) mixed a solution of 2.5% polyisobutylene in tetradecane for use in a 3D stagnation flow experiment. Although the zero-shear viscosity of 630 cP was a bit higher than blood, the shear-thinning character with very little elasticity was simulated. These solutions create problems with opacity and toxicity, however. Another citation describes how Manero (1987, 256) used a solution of 1.5% Carbopol® in ethylene glycol to create an inelastic blood analog fluid. This fluid, requiring a trace of triethylamine for viscosity control, was used to study the slow flow past a sphere. It closely matches blood viscosity, but also introduces the issue of toxicity. A more common low-elasticity fluid is an aqueous xanthan gum solution, and this will be described in the next section.

Xanthan Gum

Xanthan gum is itself an extracellular polysaccharide produced by a microorganism (Xanthomonas campestris). It is a high-molecular-weight (1.1-8.7x10⁶) derivative of cellulose and has found widespread use in the food, oil, and pharmaceutical industries. In solutions at moderate temperatures, xanthan gum conforms to a rigid rod-like configuration that aggregates into chains up to

Aqueous xanthan gum has often been used as a blood analog fluid, and it is ideal for flow experiments as reported in Thurston (1989), Mann and Tarbell (1990), and Brookshier and Tarbell (1993). Although the PAA solutions achieve a more accurate viscosity characterization, they comprise an elasticity that is far greater than that of blood. On the other hand, xanthan gum solutions are poorer representatives of true blood viscosity, but their weak elasticity is closer to blood's behavior (although still slightly higher). By adding a significant quantity of glycerin (40-50% by weight), the inelastic component of viscosity can be promoted with respect to the elastic component to get a true blood analog (Brookshier and Tarbell 1993, 109). Moreover, xanthan gum is resistant to viscosity degradation, is non-toxic, and produces nearly transparent solutions.
CHAPTER 3

EXPERIMENTAL ARRANGEMENT

The facilities used to generate the data in this report are described here. Two main laboratories were accessed: the OSU Hydrodynamics Laboratory and the OSU Rheological Measurement Laboratory.

Established in late 1995, the OSU Hydrodynamics Laboratory grew out of a seed grant to develop the 6x18-Inch Rheology Tunnel at AARL. This tunnel, patterned after common water tunnel designs, was used to study the external fluid dynamics of hydrofoil models that were simplified 2D representations of the IVAS impeller blades. In late 1997, the focus shifted to the scale modeling of the IVAS impeller and volute, and the Large-Scale Rotor Testbed was developed in the same laboratory space as the tunnel. This facility is known as the LSRT and, as the primary tool in this study, will be described in detail in Section 3.1.

The OSU Rheological Measurement Laboratory is managed by the Center for Advanced Polymer and Composite Engineering. Located within the Department of Chemical Engineering, the laboratory equipment comprises
various spectrometers as well as fluids and solids analyzers. The focus of Section 3.2 will be the rheological measurements performed and the equipment used.

### 3.1 THE LARGE-SCALE ROTOR TESTBED

The purpose of the LSRT is to allow fluid dynamics tests to be performed on a variety of different candidate centrifugal impeller designs. The volute and pump housing, made of polished Plexiglas®, is a 9X-scale (9.162:1) model of IVAS volute #3374. The layout of the LSRT flow loop facility as it was

![Layout of the LSRT installation](image)

**Figure 3.1** Layout of the LSRT installation
configured for the current study is shown in Figure 3.1.

Different rotor configurations are easily installed in the testbed, which was used to make precision pressure measurements and flow visualizations to deduce the performance of IVAS impellers #3386 and #3452. The impeller assembly involves a common rotor disk and centerbody (nose cone) with seven removable rotor blades; the outside diameter of the impeller is 29.043 cm. By scaling the geometry up by 9.162, the impeller speed is scaled down by almost 1/84 of the full-scale speed of 2,850 RPM. This manageable low speed, combined with the clear acrylic housing, allows for dye injection visualizations to be performed and photographed.

To fully describe this facility, the following subsections will outline the LSRT performance specifications, configuration and design, instrumentation, data acquisition, and flow quality assessment.

### 3.1.1 Specifications

The LSRT operating specifications are shown in Table 3.1. Both metric and English equivalents have been tabulated. Much higher flow rates (over 700 lpm) are attainable and can be measured in the facility, but these are out of the range of interest for the current study. Similar limitations are placed on the impeller speed range. The velocities and Reynolds numbers tabulated are based on the range of operability used in this study.
Flow Rate  
13.5 – 295 l/min (3.5 – 78 gal/min)

Discharge Gauge Pressure  
72 – 269 mmHg (1.4 – 5.2 psig)

Impeller Speed  
0 – 60 RPM

Pipe Velocity  
3.0 – 66 cm/sec (1.2 – 26 in/sec)

Reynolds Numbers (Water)  
Pipe  
3,000 – 65,100
Rotor  
0 – 265,000

Measurable Head Rise  
Up to 5.2 mmHg (0.1 psid)

<table>
<thead>
<tr>
<th><strong>Table 3.1</strong></th>
<th>LSRT performance envelope</th>
</tr>
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### 3.1.2 Facility Configuration and Design

The existing Rheology Tunnel flow loop was modified to accommodate the LSRT. The piping was configured to take advantage of existing components because the design philosophy was based on a dual-usage flow facility (i.e., utilizing both with a minimum required effort to convert the piping system).*

As a result, the facility uses the reservoir, circulation pump, and lower piping of the original tunnel plumbing (if shown in Figure 3.1, the tunnel test section would be situated on the side of the reservoir that is opposite the circulation pump). The mid-level pipe is 1.96 m above the floor and connects to the loop flow meter and the LSRT feed pipe; this is the tunnel's bypass line when a butterfly valve and elbow pipe are bolted to this flange. The upper pipe, which

---

* It should be noted that this loop configuration is not the optimum layout for a facility like the LSRT. Recommendations for a dedicated flow loop supplying only the LSRT are presented in chapter 6.
connects to the LSRT siphon line at a height of 2.90 m, is the main supply pipe for the tunnel when a 1.52-m connecting pipe section is bolted to this flange.

The rotor testbed itself is positioned next to the reservoir so that a relatively short pipe can discharge the fluid back into the reservoir. The housing is located 1.14 m above the floor.

3.1.2.1 System Flow Loop

The 10.16-cm (4-inch) schedule-80 PVC piping system that supplies the LSRT shares common components with the Rheology Tunnel, as described in the previous section. The flow loop as pictured in Figure 3.1 is traversed in the following sequence: reservoir tank, circulation pump, vertical pipe, turbine flow meter, flow conditioner, LSRT, discharge pipe, orifice plate flow restrictor, shutoff valve, and reservoir tank. The variable compliance chamber and siphon/relief tube are both encountered in parallel with the main flow line.

Circulation of the flow loop is provided by an Aurora centrifugal pump head close-coupled to a U.S. Electrical Motors® direct-drive motor. The pump stage is capable of providing up to 795 lpm (210 gpm) of flow at a discharge gauge pressure of 105 kPa (15 psig). The impeller diameter is 15.88 cm (6.25 in). The driver stage is a 2,200-W (3-hp) three-phase, 208-volt AC motor that operates at speeds up to 1,750 RPM. Speed control of the motor is accomplished with a Reliance Electric® 3,700-W (5-hp) variable frequency drive system. This
drive system was not operated above 58% power so that the circulation pump motor would not incur overdrive damage.

Branching off of the main line is a variable compliance chamber. The compliance chamber is a vertical column of trapped air. It acts as a shock absorber for the piping system by damping out pump pressure pulsations; this device works very effectively here as it does in home plumbing systems. A pressure valve at the top allows for varying the water/air ratio and thus the compliance in the tube.

Downstream of the compliance chamber and upstream of the turbine flow meter, the piping system branches off to the siphon/relief tube (upper pipe), which is the highest point in the system. This includes a check valve that allows vacuum break when the circulation pump is shut down. It also has a ball valve controlling a small bypass line to allow trapped air to escape during flow startup. The relief tube empties back into the reservoir tank.

Fluid traveling in the horizontal leg of the piping system passes through an inline flow meter. This device will be described in Section 3.1.3.

Downstream of the flow meter section, the pipe turns downward to feed the 31.75-cm (12.50-in) flow conditioner segment. The flow in the pipe must be conditioned to promote low levels of uniform turbulence prior to entrance to the LSRT inlet. Flow uniformity is accomplished through a series of inline removable and interchangeable flow restrictions and straighteners as seen in
Figure 3.2. Trial-and-error tests yielded a 2.54-cm (1-inch) thick section of honeycomb (6-mm cells) and two stainless steel screens (70% open area). These components were arranged in the following streamwise sequence: 2.54-cm spacer, honeycomb section, 4.13-cm spacer, first screen, 10.16-cm spacer, second screen, and a 12.07-cm spacer. This setup allows enough porosity for air bubbles to rise freely into the upper pipes during startup while providing enough flow-smoothing pressure loss during test runs. Velocity distributions at the LSRT inlet will be presented and discussed in Section 3.1.5. The LSRT housing will be described in detail in Section 3.1.2.2.

Figure 3.2  Inlet flow conditioning assembly

Downstream of the LSRT is a length of discharge pipe and an orifice-plate flow restrictor for backpressure control. The sharp-edge orifices (0.97-, 1.40-, 1.92-, 2.59-, and 3.18-cm diameters) give a continuous range of 13.5–295 lpm.
(3.5–78 gpm) through the LSRT. Orifice flow calibrations are documented in Appendix E. The orifice plates ensure that the backpressure is 72–269 torr (1.4–5.2 psig). Thus, the turbine flow meter remains fully wetted during operation. To prevent disturbances from the orifice plate propagating upstream into the LSRT, a pipe 12 diameters long separates the two components. Finally, the shutoff valve permits the flow loop to be filled with fluid and purged of air.

3.1.2.2 Pump/Volute Housing

A four-view diagram of the IVAS pump baseline volute/housing is shown in Figure 3.3. All interior surfaces of the LSRT, including the inlet, upper housing contour, volute, and discharge passage, are scaled to 916% of the true-scale pump contours shown. The outer shape of the LSRT housing does not reflect the actual pump geometry however. The outside is milled to a 54.61-cm (21.50-in) constant-diameter section and the lower housing is terminated just below the volute section, such that the total height of the assembled LSRT pump volute/housing is 14.61 cm (5.75 in). The split between upper and lower housings occurs at the volute midplane, and an O-ring seal prevents leakage. The discharge passage is located in a 5.08-cm (2.00-in) tangential extension of the housing. All components were CNC-machined in the OSU Department of Physics machine shop using a bed mill. The flow surfaces were wet-sanded with 600-grit sandpaper and polished to a glassy finish. See Figure 3.4 and Figure 3.5 for the upper housing and inlet detail and the entire LSRT assembly.
Figure 3.3  Geometry of the IVAS baseline volute/housing

Figure 3.4  LSRT upper volute/housing contours
3.1.2.3 Impeller Models

Unlike the true-scale impeller assembly, which involves an annular rotor with a stationary centerbody, the LSRT impeller assembly is simpler in construction. It has a common 29.479-cm (11.606-in) impeller disk with a rotating centerbody (rounded cone of 17.104-cm base diameter, 4.798-cm height). The seven blades are removable for trade studies between different designs, each blade being positioned and secured to the rotor disk with a 3.18-mm (0.125-in) dowel pin and a #10-24 stainless cap screw. Geometric data for
CCF impeller models #3386 and #3452 are shown in Table 3.2, and 2D blade characteristics are shown in Table 3.3.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Impeller #3386</th>
<th>Impeller #3452</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tip Diameter (cm)</td>
<td>29.043</td>
<td>29.043</td>
</tr>
<tr>
<td>Eye Diameter (cm)</td>
<td>8.502</td>
<td>10.802</td>
</tr>
<tr>
<td>ID/OD Ratio</td>
<td>0.293</td>
<td>0.372</td>
</tr>
<tr>
<td>Inlet Angle (deg)</td>
<td>45</td>
<td>25</td>
</tr>
<tr>
<td>Exit Angle (deg)</td>
<td>78</td>
<td>78</td>
</tr>
</tbody>
</table>

Table 3.2   LSRT impeller geometry parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Impeller #3386</th>
<th>Impeller #3452</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chord (cm)</td>
<td>11.18</td>
<td>10.95</td>
</tr>
<tr>
<td>Thickness (cm)</td>
<td>1.04</td>
<td>1.17</td>
</tr>
<tr>
<td>Percent Thickness</td>
<td>9.3</td>
<td>10.7</td>
</tr>
<tr>
<td>XCAM,max (cm)</td>
<td>9.35</td>
<td>4.29</td>
</tr>
<tr>
<td>Max. Camber (cm)</td>
<td>-0.28</td>
<td>+0.47</td>
</tr>
<tr>
<td>Percent Camber</td>
<td>-2.5</td>
<td>+4.3</td>
</tr>
</tbody>
</table>

Table 3.3   Rotor blade two-dimensional characteristics

Impeller #3386 as depicted in Figure 3.6 was the original CCF design. These blades are 9.3% thick with cylindrical leading edges. Semi-elliptical trailing edges introduce the small negative camber listed in the table. A blade offset of 2.97 cm (10% D) from the rotor centerline gives an inlet angle of 45° with respect to the local tangent line.
The #3386 blade was superseded by the #3452 design, shown in Figure 3.7. The aft 38% of the latter blade profile is identical to the former, but the forward portion is cambered to reduce the inlet flow angle. Thus, the 4.3%
camber increases the eye diameter from 8.50 cm to 10.80 cm and decreases the inlet angle from 45° to 25°. This also slightly affects the blade thickness ratio.

Both impeller designs are based on the same tip diameter and the same centerbody cone. However, blade upper-surface contours are markedly different. The upper surface of the #3386 blades were lathe-cut after assembly of the rotor in order to match the contour of the LSRT upper housing; thus sharp corners exist between the side and top surfaces. In the #3452 design, these corners are substantially rounded to a radius that is approximately 30% of the blade thickness.

3.1.2.4 Rotor Drive System

Impeller rotation is managed by a geared, sealed shaft system driven by a step motor. This system is shown in Figure 3.8. The drive shaft is a 2.54-cm (1.00-in) diameter thick-walled tube that accommodates four 1.59-mm (0.063-in) stainless tubulations for transporting dye into the rotor assembly. Inside the LSRT lower housing, these lines can be connected to dye injection ports on the rotor disk surface or impeller blade surfaces.

The 40.6-cm (16.0-in) long shaft assembly passes through the Plexiglas® seal plate that encloses the lower housing. A stuffing box is bonded to the seal plate to prevent leakage around the rotating shaft. The stuffing box contains three rings of 6.4-mm (0.25-in) braided graphite cord that are compressed by a
gland follower. Four adjusting screws provide variable compression on the packing and thus variable seal.

Figure 3.8 LSRT microstepper-controlled drive system

At the lower termination of the shaft is a dye manifold, shaft coupler, and gear pair. Near the top center of Figure 3.8, a Plexiglas® manifold can be seen surrounding the shaft. This allows dye to be pumped through the interior lines in the rotating shaft for impeller visualizations. The shaft coupler is immediately below the dye manifold, and it ensures shaft alignment between the gearing and the impeller position. Various shims were used at the shaft coupler to adjust the axial position of the impeller and thus vary the endwall clearance, $c_w$. Directly below the shaft coupler is the idler gear, which creates a 4:1 ratio with the drive gear on the motor shaft.
A stepper motor actuates the drive system. This type of motor system satisfies the need for a low constant-torque requirement, low speed operation, and low-speed smoothness. A stepper motor was chosen over a hybrid servo or direct-drive servomotor based on its set-point speed control. The American Precision Industries® 42D-112-12A stepper motor delivers a static torque of 7.9 Nm (70 in-lbs). A Panther™ HI2 miniature microstepper controller, made by Intelligent Motion Systems®, is used as a driver, indexer, and power supply for the motor. The controller is configurable for resolutions between 200 and 51,200 steps/rev, resulting in maximum speeds between 6,000 and 23 RPM, respectively. For the current tests, the programmed motor resolution is 1,600 steps/rev. and the resulting maximum motor speed is 750 RPM. With the 4:1 gear ratio, the effective shaft resolution is 6,400 steps/rev at a maximum shaft speed of 187 RPM. Thus, the operation of the motor is precise and smooth.

Commands are issued to the controller through the use of a QuikPro™ handheld keypad/terminal. This accomplishes stand-alone motor operation, although RS-232 and RS-422 serial communication and control are also possible.

3.1.3 Instrumentation

The LSRT facility was instrumented to effectuate two purposes. One was performance mapping by using quantitative measurements such as pressure, flow rate, and temperature. The other purpose was qualitative flow visualization. This section describes the sensor hardware used to make the
measurements, the facility locations to enable those measurements, and the flow visualization techniques implemented.

3.1.3.1 Sensors and Hardware

The primary instrumentation hardware deals with measuring impeller speed, flow rate, housing gauge pressure, pressure (head) rise across the impeller, total pressure at the impeller inlet, and temperature. All sensors were operated within the linear calibration range, and the calibration curves used to reduce the data are found in Appendix E.

Impeller rotational speed is not measured; it is a function of the motor preset speed. Because of very low torque on the impeller by the stuffing box and the fluid, the motor executes the precise command stepping speed. This is the primary reason a stepper motor was chosen: so that a shaft encoder would not be necessary. Thus, the impeller speed is four times the motor speed, based on the gear ratio. This fact has been verified with a stroboscope; speed differences were less than the resolution of the strobe.

Loop flow rate is measured by a Potter Aero Corp.® 2C-50318 turbine flow meter. The horizontal pipeline constricts from a diameter of 10.2 cm (4.0 in) to 6.4 cm (2.5 in) to match the meter’s inflow diameter, and this requires a pipe 15 diameters in length upstream and 25 diameters downstream. The meter operates by using a Hall-effect sensor with a K-factor of 31.4 l/pulse (119 gal./pulse). The 2-wire, low-level magnetic pickup signal is processed by an
Omega\textsuperscript{®} DPF700 frequency counter/totalizer and is subsequently output as an analog voltage proportional to the rate. This setup is accurate within the linear range of 11–560 lpm (3–147 gpm). For higher viscosity fluids, the error in flow rate measurement is insignificant. Measurable errors in turbine K-factor of ±1\% full scale will occur at low flow rates (below 30\% maximum flow), but only if the kinematic viscosity is greater than 100 centistokes (Omega 1995, F-6). The fluids used in this study are within the accurate range of this setup. Figure 3.9 shows the turbine flow meter and digital rate meter.

![Figure 3.9 Inline turbine flow meter and digital rate meter](image)

Two pressure transducers are used to obtain the pressure data in the LSRT. A high-range 69-kPa (10-psig) gauge sensor is used to measure the housing pressure, which rarely exceeds 34 kPa (5 psig). This sensor, a Statham\textsuperscript{®} PDCR23 wet/dry silicon diaphragm strain gage transducer, has an accuracy of
0.50% full-scale output. The reference side is left open to the ambient pressure. A custom-built excitation/power supply/amplifier unit accompanies this transducer. This unit provides 12 volts of bridge excitation and amplifies the DC bridge output by a gain of 100 to deliver a signal in the range of ±7.5 V.

The impeller pressure rise is much lower, so a Validyne® DP-15 variable-reluctance differential sensor is used. This transducer has a diaphragm sized for 5 mmHg (0.1 psid), making it extremely accurate and sensitive (0.25 full-scale output) when considering non-linearity, hysteresis, and non-repeatability. Because this transducer has wet/wet capabilities, it was configured to measure $(p_{\text{OUT}} - p_{\text{IN}})$. This same sensor was used to sequentially measure the volute pressures, in which case the differential measurement was $(p_{\text{t}} - p_{\text{OUT}})$, as well as the inlet total pressure, in which case the differential was $(p_{\text{T}} - p_{\text{IN}})$. A model CD-15 carrier/demodulator provides the excitation signal and amplifies the output. The carrier signal is a square wave having a frequency of 3–5 kHz and amplitude of 5 VRMS. Demodulated DC output gain is over 50.

The only temperatures recorded were of the room and tank fluid. Ambient temperature in the room is measured with a mercury thermometer; this temperature changes very gradually so high sensitivity is not needed. The tank fluid temperature is measured with an Omega K-type (copper-copper) thermocouple with ice point reference. It has a 30-cm (12-in) long stainless probe tip that is inserted in a quiescent corner of the reservoir tank. A Bay
Laboratories® 5303 instrumentation amplifier boosted the thermocouple signal with a gain of 2,000 and filtering frequency of 0.1 Hz.

3.1.3.2 Housing Pressure Locations

Pressure is measured at seven locations in the LSRT and supply piping. There are six wall pressure taps in the volute/housing. The pump head (also called Δp) is measured by the differential between the discharge pressure and the inlet pressure to get the net increase in fluid pressure through the pump. These two pressure taps are located in the pipe mounted to the upper housing, just prior to the inlet expansion, and in the flange at the pump discharge plane. Both locations have the same pipe inner diameter, so there is no venturi effect on the measurement. The height of these tap locations is 11.24 cm (4.425 in) and 0.00 cm, respectively, as referenced from the housing split line.

Figure 3.10  Circumferential position measurement protocol
Four other wall pressure taps are located at orthogonal positions in the lower volute, spaced 90 degrees apart. These are used to measure the pressure rise as fluid issues around the volute passage. These positions are shown in Figure 3.10 and delineated in Table 3.4. The pressure tap at the 360° location is considered to be at the beginning of the diffuser section of the volute.

<table>
<thead>
<tr>
<th>Radial Position</th>
<th>r/R</th>
<th>Circumferential Position</th>
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<tr>
<td>(cm)</td>
<td>(in)</td>
<td></td>
</tr>
<tr>
<td>16.87</td>
<td>6.641</td>
<td>1.162</td>
</tr>
<tr>
<td>17.62</td>
<td>6.938</td>
<td>1.214</td>
</tr>
<tr>
<td>18.22</td>
<td>7.172</td>
<td>1.255</td>
</tr>
<tr>
<td>18.93</td>
<td>7.453</td>
<td>1.304</td>
</tr>
</tbody>
</table>

Table 3.4   Volute pressure tap locations

Total pressure was measured by using a pitot probe in the area between the flow conditioner and the LSRT inlet. Figure 3.11 shows the total pressure probe installation in the inlet pipe. The probe tip is made from 1.59-mm (0.063-in) diameter stainless tubing with a 32.5-mm (1.281-in) right-angle bend and a 45° internal chamfer at the end. It is soldered inside a 15-cm (6.0-in) length of 3.18-mm (0.125-in) diameter stainless tube (the horizontal piece). A Swagelok™ fitting with nylon ferrules allows the probe to be easily positioned and removed. This fitting is located 20.94 cm (8.245 in) above the housing split line such that the probe tip is 24.20 cm (9.526 in) above the split line.
3.1.3.3 Flow Visualization Instrumentation

Flow visualization was achieved with a four-channel dye injection system and miniature color CCD camera. The injector is a positive-pressure dye injection system with four 10-ml reservoirs as shown in Figure 3.12. A Porter® model 8286 regulator is used to pressurize all reservoirs to a gauge pressure of 20–35 kPa (3–5 psig) to force dye through the dye injection ports in the LSRT housing. Needle valves at the bottom of each reservoir are used to individually adjust the dye flow rates. This system is capable of delivering dye for 20–30 minutes of testing before requiring dye replenishment.
Figure 3.12 Four-component dye injection system

The dye is a composition of waterproof drawing ink, water, glucose, and small amounts of fat. To supply the glucose and fat, sweetened non-dairy coffee creamer were mixed with the water and ink in ratios of one part each. This mixture, which requires refrigeration, created bright dye filaments that did not immediately disperse. The streak patterns were filmed with the CCD.

A Connectix® Color QuickCam™ CCD was operated in still and video modes to capture steady and unsteady dye-streak patterns. This CCD is shown in Figure 3.13 and was used to perform flow visualization in the laboratory reference frame by capturing the flow patterns at the stationary upper housing surface. It has an image resolution of 640x480 and requires halogen floodlighting to produce quality images. Since the camera requires connection to a PC
computer, an eight-conductor PS-2 cable was spliced to the camera cord so that standard two-meter keyboard cables could be used to extend the camera up to 6 m from the computer. An adapter was fabricated to mount the CCD to a standard photography tripod.

![Figure 3.13  Miniature 640x480 color CCD camera](image)

<table>
<thead>
<tr>
<th>Circumferential Position (deg)</th>
<th>20° CCW Position</th>
<th>200° CCW Position</th>
<th>Radius (cm)</th>
<th>r/R</th>
</tr>
</thead>
<tbody>
<tr>
<td>20.00</td>
<td>200.00</td>
<td>10.16</td>
<td>0.6997</td>
<td></td>
</tr>
<tr>
<td>18.43</td>
<td>198.43</td>
<td>11.43</td>
<td>0.7871</td>
<td></td>
</tr>
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<tr>
<td>16.16</td>
<td>196.16</td>
<td>13.97</td>
<td>0.9620</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.5  Upper housing dye injection locations
Upper housing dye injection ports are located at two circumferential positions to enable flow visualization. These locations are based on the same reference used for the volute pressure taps (see Figure 3.10) and measured in the counterclockwise azimuthal direction. The coordinates are listed in Table 3.5. Small orifices of 1.0 mm (0.040 in) diameter are drilled through the housing inner surface, and 1.6-mm (0.063-in) diameter stainless tubes are bonded at the housing outer surface. These short tubes can be seen in Figure 3.5 on page 69.

Visualizations were also made with respect to the rotating reference frame. Four dye injection ports are circumferentially distributed at equal intervals between two blades of the rotor disk. These rotor disk dye ports are located at the same radial coordinates as the upper housing dye ports, and increase in radial position as the blade passage is traversed in the counterclockwise direction (when viewed from above). The dye was pumped into the manifold at the base of the rotating LSRT driveshaft and then traveled through small tubes located within the hollow driveshaft up to the dye ports in the rotor disk surface.

In this case, videos and still images were achieved by mounting a JVC compact video camera to the impeller driveshaft so that it turns at the same speed as the impeller. With the camera positioned below the LSRT housing, an unobstructed view up through the lower seal plate and rotor disk was possible. This technique was used to capture the flow patterns on the rotating impeller
disk surface between a blade passage. Digital image and video files were produced from the video tape with a PC-based frame-grabber.

3.1.4 Data Acquisition

Analog voltage outputs from the laboratory instruments are captured by a data acquisition system. A Pentium™-class computer is used to monitor a multifunction A/D board used for signal acquisition. The board, a 12-bit Computer Boards, Inc.® CIO-AD16, is configured to accept eight differential analog inputs in the range of ±10V. It also has three 24-bit channels of digital output that can be used to control laboratory test hardware for complete automation of the data acquisition procedure. The configuration of the eight A/D channels is summarized in Table 3.6.

<table>
<thead>
<tr>
<th>Analog Input Channel</th>
<th>Measurement</th>
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<tr>
<td>0</td>
<td>Gauge Pressure</td>
</tr>
<tr>
<td>1</td>
<td>NOT USED</td>
</tr>
<tr>
<td>2</td>
<td>Pump Pressure Rise</td>
</tr>
<tr>
<td>3</td>
<td>Loop Flow Rate</td>
</tr>
<tr>
<td>4</td>
<td>Tank Temperature</td>
</tr>
<tr>
<td>5</td>
<td>NOT USED</td>
</tr>
<tr>
<td>6</td>
<td>NOT USED</td>
</tr>
<tr>
<td>7</td>
<td>NOT USED</td>
</tr>
</tbody>
</table>

Table 3.6 Configuration of A/D board analog input channels
Figure 3.14 Functionality of program DatAcqRT

The function of the data acquisition program DatAcqRT is shown in Figure 3.14. This customized software provides a platform for transducer calibration, data acquisition, data "snapshot" capturing, and data reduction/analysis. The programming was done in the VisualBasic® environment and is based on a flexible relationship between module blocks, which were uniquely tailored to capture the phenomena of interest.

Acquisition Procedures

Data acquisition was accomplished in two modes: performance point measurements, which involved changing flow conditions between data points,
and survey runs, which were multiple data points acquired at fixed flow conditions. Performance point measurements were the most common mode of data acquisition and were used to map the pump curve at different impeller speeds. The procedure for this type of data acquisition involved setting the loop at maximum flow rate, which allowed trapped air to rise to the upper siphon tube and flow conditions to stabilize, and then starting the impeller rotation. After approximately five minutes, a data point was taken that simultaneously acquired all eight channels of analog voltages. The circulation motor power was incrementally reduced, decreasing the loop flow rate, and the acquisition procedure was repeated after 10–15 seconds of settling time. This produced about five data points between the maximum and minimum flow rates for each orifice plate. After these five points were acquired, the impeller speed was changed and the whole procedure was repeated. Upon completing the flow rate-impeller speed matrix, the next orifice plate was installed and the matrix repeated. Thus, after superimposing the data from each orifice plate, about 25 data points define the pump curve for each impeller speed. The data-averaging window was 10–12 seconds at sampling frequencies of 400–500 Hz.

Other situations, such as inlet flow quality surveys or volute pressure measurements, required the loop flow conditions to be held constant. Data that was acquired in this mode was corrected to the average flow conditions over the time interval, since small fluctuations occurred during the test run.
3.1.5 Inlet Flow Quality

Total pressure and velocity surveys were performed in the LSRT inflow pipe to ascertain the flow quality. The pitot probe used for these surveys, described in Section 3.1.3.1 and pictured on page 80, was installed only for the pipe flow surveys. It was removed for all subsequent performance tests.

Pipe flow surveys were achieved by traversing the pitot horizontally across the inside diameter (9.84 cm; 3.88 in) and measuring the change in total pressure. Pressures were measured with the low-range DP-15 transducer and referenced to the LSRT inlet pressure. The survey plane is 14.04 cm (5.53 in) downstream of the last flow-conditioning screen, and the LSRT inlet wall tap is 12.96 cm (5.10 in) directly downstream of the survey plane. Thus, the survey station is 54 cm (21 in) downstream of the downward bend in the supply pipe.

However, true velocities in the measurement plane could not directly be determined. Since the static tap is downstream of the total pressure measurement location, the velocity based on the difference \( (p_t - p_{in}) \) is not the actual velocity. Pressure losses along the pipe are a result of the still-developing viscous flow that has turned a corner in the pipe bend and then passed through the flow conditioner. Furthermore, the fact that the wall pressure tap is directly downstream of the pitot probe causes question as to the validity of the inlet pressure measurement. If wake artifacts from the pitot plane are present at the inlet plane, then the static pressure measurement will be lower than actual and the velocity will be higher than actual. The results verify that the inlet pressure
is lower than at the pitot probe. So for the purpose of these surveys, the term reference velocity will be used. This velocity is defined using Bernoulli’s equation:

\[
V_{\text{ref}} = \sqrt{\frac{2 \cdot (p_T - p_{\text{IN}})}{\rho}}
\]

Flow surveys were done for two flow rates: 104 and 242 lpm. These two conditions represent approximately half the maximum flow rate encountered in the impeller tests and the maximum flow rate, respectively. The corresponding bulk velocities through the pipe at these two flow rates were 22.8 cm/sec and 53.0 cm/sec (9.0 and 20.9 in/sec). These surveys were repeated for each fluid used in the impeller performance studies.

![Figure 3.15 Inlet velocity profiles at 104 LPM flow rate](image)
Figure 3.16 Inlet velocity profiles at 242 LPM flow rate

The results of the inlet pipe flow surveys for water and xanthan gum solutions are shown in Figure 3.15 and Figure 3.16. A slight asymmetry in the velocity profile is evident, particularly for water and especially at the higher flow rate. This deviation comes from the secondary flow generated by the pipe bend upstream of the measurement station. The negative radial coordinates in the figures designate the outer half of the pipe, with respect to the bend and the flow direction. This is the half of the pipe where secondary flows would be prominent, based on the centrifugal-induced pressure gradient in the pipe bend. It is obvious that the higher-viscosity xanthan gum solutions greatly reduce the
flow asymmetry. Normalized data for all surveys are compiled in Figure 3.17.

These results validate the acceptable quality of inlet flow profiles.

![Dimensionless inlet velocity profiles](image)

**Figure 3.17** Dimensionless inlet velocity profiles

### 3.2 RHEOLOGICAL MEASUREMENT LABORATORY

All non-Newtonian fluids were tested to characterize the shear-thinning viscosity with the help of the Rheological Measurement Laboratory (RML). The laboratory is part of the Center for Advanced Polymer and Composites Engineering (CAPCE) and is managed by the Department of Chemical Engineering at OSU. RML is a testing facility that helps researchers understand material behavior through characterization of fundamental rheological properties. Tests can be performed on solids, viscous liquids, and reacting...
systems to determine viscous, elastic, and viscoelastic properties of polymer substances. Although the RML boasts four computerized test systems, the focus of this section will be the Rheometrics RFS-II fluids spectrometer. The issues of data analysis and fluid handling are also addressed.

3.2.1 The Rheometrics RFS-II Fluids Spectrometer

The RFS-II is a dynamic analyzer that is used for high-accuracy, high-response viscosity measurements. A very sensitive air-lubricated force rebalance transducer enables testing of fluids having low viscosities approaching that of water. The user workstation and testing apparatus are shown in Figure 3.18, and the specifications are detailed in Table 3.7.

Figure 3.18  Computer-controlled RFS-II fluids spectrometer

91
<table>
<thead>
<tr>
<th>Tests Available</th>
<th>Steady Rate Sweep</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steady</td>
<td>Steady Rate Sweep</td>
</tr>
<tr>
<td>Dynamic</td>
<td>Step Shear Rate</td>
</tr>
<tr>
<td></td>
<td>Thixotropic Loop</td>
</tr>
<tr>
<td></td>
<td>Frequency Sweep</td>
</tr>
<tr>
<td></td>
<td>Temperature Sweep</td>
</tr>
<tr>
<td></td>
<td>Strain Sweep</td>
</tr>
<tr>
<td></td>
<td>Step Strain</td>
</tr>
<tr>
<td>Test Tools Available</td>
<td>Couette</td>
</tr>
<tr>
<td></td>
<td>Cone and Plate</td>
</tr>
<tr>
<td></td>
<td>Parallel Plate</td>
</tr>
<tr>
<td>Actuator Speed</td>
<td>Steady 0.001 - 100 rad/sec</td>
</tr>
<tr>
<td></td>
<td>Dynamic 0.001 - 500 rad/sec</td>
</tr>
<tr>
<td>Strain Rates Achievable</td>
<td>0.01482 to 1481 sec-1</td>
</tr>
<tr>
<td>Transducer Specifications</td>
<td>Torque 0.2 - 200 g-cm</td>
</tr>
<tr>
<td></td>
<td>±0.1°</td>
</tr>
<tr>
<td></td>
<td>Linearity 0.10%</td>
</tr>
<tr>
<td></td>
<td>Hysteresis 0.05%</td>
</tr>
<tr>
<td></td>
<td>Temperature Limits (with oven) (Ambient + 10° C) - 600° C</td>
</tr>
</tbody>
</table>

Table 3.7  RFS-II specifications

**Test Description**

The RFS-II rheometer consists of the control computer, system control unit, test station, and test control/analysis unit. A Couette tool with dimensions shown in Table 3.8 was used to produce a nearly pure shear field in the test fluids. When installed in the RFS-II, the cup is fixed to the lower rotational fixture, and the cylindrical bob is mounted on the upper stationary torque fixture. The cup is then turned at the specified speeds while the transducer measures the resulting torque on the bob. Thus, the cup RPM and bob torque are
used to compute the strain rate and shear stress, respectively. A Couette tool is the best test geometry for the fluids used, based on the capability to resolve low viscosities (near 1 cP) and the attainable high shear rates (over 1,400 sec\(^{-1}\)).

<table>
<thead>
<tr>
<th>Cup Diameter</th>
<th>26.88 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bob Diameter</td>
<td>25.00 mm</td>
</tr>
<tr>
<td>Bob Length</td>
<td>31.98 mm</td>
</tr>
<tr>
<td>Sample Volume</td>
<td>8 – 10 ml</td>
</tr>
</tbody>
</table>

Table 3.8   Couette tool geometry specifications

### 3.2.2 Data Analysis

The raw data for a viscosity test is cup speed and bob torque. Before viscosity behavior can be determined, the data must be converted to strain rate and wall shear stress. Then, the viscosity results from these two values. The shear stress at the surface of the bob is proportional to the measured torque, \(T\), and inversely proportional to the square of the bob radius, \(R_b\), and the length, \(L\):

\[
\tau_{\text{wd}} = \frac{T}{2\pi R_b^2 L}
\]

Because of the nonhomogeneous shear field that exists in the gap between the cylinders, the resulting velocity gradient is difficult to measure as a function of position. Instead, the velocity gradient is transformed to a derivative
of the stress with respect to rotation rate. Since the torque is proportional to the stress, this derivative can be written as:

\[ n = \frac{d\ln T}{d\ln \omega} \]

Then, the strain rate can be calculated at the surface of the bob. It is based on the rotation rate, \( \omega \):

\[ \dot{\gamma}_{wb} = \frac{2 \omega}{n(1 - \kappa^{2n})} \]

The gap ratio is denoted by the ratio of bob radius to cup radius, \( \kappa = R_b/R_c \).

Errors and Corrections

The predominant errors in Couette viscometry are due to end effects. End effects are due to the torsional shear flow on the bottom surface of the bob, which contributes to the total torque on it. The bob used in the current tests has a recessed cavity on the bottom that traps air as the bob is lowered into the cup, thus creating a negligible shear contribution from the bottom surface.

Two other potential problems associated with Couette viscometry are wall slip and viscous heating. Wall slip results from a lower viscosity layer forming on the surfaces and lowering the measured torque. This effect is only perceptible for concentrated suspensions and polymer solutions and was not a factor in the present study. Viscous heating was also not a factor, due to the low-viscosity, low-elasticity fluids used.
All measurements of the current fluids were performed at room temperature. The data-averaging window was 20 seconds with a delay of 5 seconds between measurements. Data were acquired for a clockwise rotation only, although dual-mode rotation was available.
CHAPTER 4

EXPERIMENTAL RESULTS OBTAINED

The design philosophy and skillful fabrication of the LSRT facility provided an excellent basis for obtaining precise and repeatable scale pump data and flow visualization images. Of particular importance to the project was the precise characterization of viscosity, accurate measurement of the overall pump pressure rise for Newtonian and non-Newtonian fluids, and the evaluation of the volute circumferential pressure distribution. Of secondary importance was the imaging of flow patterns in the transparent pump volute/housing. These will be respectively documented in the following sections.*

Fluid characterization of aqueous xanthan gum solutions will be presented in Section 4.1. Various concentrations from 200-1,200 PPM were studied as a suitable blood “analog” fluid; curve-fitting the data yielded the four parameters necessary for implementing Cross viscosity models. These results were crucial in evaluating the non-Newtonian effects of the fluids on the

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* To reiterate the nomenclature in usage, the term "IVAS pump" refers to the 1X-scale (or true-scale) pump fabricated at CCF, while the term "LSRT pump" refers to the 9X-scale pump test facility at OSU.
performance of the pump impeller. In addition to determining the properties of the test fluids used, time histories of viscosity behavior were used to establish the impact of aging and prolonged shearing on the present xanthan solutions.

Sections 4.2, 4.3, and 4.4 will present the performance results obtained using the LSRT impeller and volute test facility. These sections will deal with Newtonian and non-Newtonian pump curve mapping and volute pressure distributions. The pump curve mapping involved measuring the overall pump head rise for a number of different flow conditions and impeller geometries. The primary geometry variation, other than changing the impeller model, was the endwall clearance. The test matrices are shown in Table 4.1 and Table 4.2.

<table>
<thead>
<tr>
<th>Xanthan Gum Concentration</th>
<th>Impeller Endwall Clearance</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.9 mm</td>
</tr>
<tr>
<td>0 PPM (Water)</td>
<td>$0 \leq \text{Re} \leq 436,000$</td>
</tr>
<tr>
<td>600 PPM</td>
<td>$0 \leq \text{Re} \leq 282,000$</td>
</tr>
</tbody>
</table>

Table 4.1 LSRT impeller #3386 pump performance test matrix

<table>
<thead>
<tr>
<th>Xanthan Gum Concentration</th>
<th>Impeller Endwall Clearance</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>3.1 mm</td>
</tr>
<tr>
<td>0 PPM (Water)</td>
<td>$0 \leq \text{Re} \leq 428,000$</td>
</tr>
<tr>
<td>600 PPM</td>
<td>$0 \leq \text{Re} \leq 268,000$</td>
</tr>
<tr>
<td>1,200 PPM</td>
<td>$0 \leq \text{Re} \leq 224,000$</td>
</tr>
</tbody>
</table>

Table 4.2 LSRT impeller #3452 pump performance test matrix
Pressure measurements were also taken in the volute to evaluate its performance contribution. The matrix used is shown in Table 4.3 and represents a less exhaustive effort than the overall pump measurements. The results of these tests will be detailed in Section 4.4.

<table>
<thead>
<tr>
<th>Xanthan Gum Concentration</th>
<th>Impeller Design Model</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>#3386</td>
</tr>
<tr>
<td>0 PPM (Water)</td>
<td>Re = 303,000</td>
</tr>
<tr>
<td></td>
<td>$c_{ew} = 0.9 \text{ mm}$</td>
</tr>
<tr>
<td></td>
<td>$c_{ew} = 3.1 \text{ mm}$</td>
</tr>
<tr>
<td>600 PPM</td>
<td>$0 \leq \text{Re} \leq 282,000$</td>
</tr>
<tr>
<td></td>
<td>$c_{ew} = 0.9 \text{ mm}$</td>
</tr>
</tbody>
</table>

Table 4.3 LSRT volute performance test matrix

To ascertain the quality of flow through the LSRT, flow visualization experiments were performed on the inner surface of the transparent pump volute/housing and the transparent rotor disk. The nature of the flow patterns, which were revealed by dye streaks injected through surface ports, is discussed in Section 4.5. Flow visualizations were obtained by utilizing a CCD camera in still and video modes. The LSRT was operated at specific speeds of 0.25, 0.39, and 0.55 in water and 600-ppm xanthan gum solution. Matrices summarizing these tests are shown in Table 4.4 and Table 4.5.
4.1 BLOOD ANALOG FLUID FORMULATIONS

Several candidate blood analog fluids were discussed in Chapter 2. The primary blood characteristics of concern for this study are its strong shear-thinning behavior and weak viscoelastic effects. While some fluids, such as solutions of polyisobutylene in tetradecane, may closely match blood's elasticity and simulate its shear-thinning behavior, they have properties that are incompatible with safety concerns and have undesirable flow visualization properties. Therefore, xanthan gum was chosen for use as a non-Newtonian
blood analog fluid because it is well-documented, easy to handle, non-toxic, and optically clear. For this study, aqueous xanthan gum in concentrations of 600 and 1,200 ppm (0.06% and 0.12% by weight) were used in the LSRT facility.

4.1.1 Fluid Characterizations

In an attempt to match the properties of blood as closely as possible, several concentrations of aqueous xanthan gum were tested. Solutions of 200, 400, 600, and 1,200 ppm (by weight) were prepared in one-liter quantities for viscosity characterization using the RFS-II rheometer. Even such small volumes of polymer solution required careful measuring and mixing techniques. To ensure complete dissolution and homogeneous properties, the solutions were mixed for 48 hours on a roller mixer. After mixing, the solutions were allowed to sit for another 24 hours to promote polymer chain stabilization. A technical-grade xanthan gum from Colony Industries® was used.

Tests were conducted at room temperature using the Couette geometry so that high strain rates could be achieved. Shear stress was measured at strain rates up to 1,400 sec\(^{-1}\). The minimum measurable strain rate is dictated by the lower limit of the force transducer, about 0.004 g\(_f\)-cm of torque.

Results of the viscosity characterization tests are shown in Figure 4.1, along with blood data from Banerjee et al. (1991, 103-104). The selection of the proper concentration depends on which of blood's properties is the most important to match. For instance, if \(\eta_0\) is the most important property, then a
concentration over 1,200 ppm would be required. If $\eta_0$ effects predominate, then 860 ppm would suffice. However, the focus for this study is to match the degree of shear thinning, or the ratio $\eta_0/\eta_\infty$, so the best concentration would be 720 ppm. For blood, this viscosity ratio is 16.2. This compares with a value of 9.2 at 600 ppm and 53.3 at 1,200 ppm. Therefore, the two fluids of 600 and 1,200 ppm were chosen so that the zero-shear viscosity and the viscosity ratio could be bracketed. The curve-fitted Cross parameters are tabulated in Table 4.6.

Figure 4.1  Viscosity characterization of five non-Newtonian fluids

An important assumption about polymer solutions is the condition of a dilute solution. For this condition to be met, the intrinsic viscosity has to be a linear function of concentration, making the zero-shear viscosity a quadratic
function in C, as covered in Chapter 2. Figure 4.2 shows that the fluids used in the study maintain this requirement for dilute polymer solutions with an $R^2$ correlation of over 99%.

![Figure 4.2 Approximate quadratic viscosity variation with concentration](image)

The concentration similarity behavior also further validates the assumption of dilutancy and is shown in Figure 4.3. For this plot, the reference shear stress was taken at the onset of the lower Newtonian plateau at high strain rates. It is mathematically defined at the point where the viscosity is above the infinite-shear viscosity by a margin of 10% of the difference between $\eta_\infty$ and $\eta_0$. This corresponds to a reference shear of approximately 10 Pa for all xanthan
concentrations. The reference strain rates at this reference stress have been tabulated in Table 4.6.

Figure 4.3  Viscosity master curve for xanthan gum solutions

<table>
<thead>
<tr>
<th>Fluid</th>
<th>$\eta_0$ (cP)</th>
<th>$\eta_\infty$ (cP)</th>
<th>K (sec)</th>
<th>n</th>
<th>$\eta_r$ (cP)</th>
</tr>
</thead>
<tbody>
<tr>
<td>200 ppm Xanthan</td>
<td>4.20</td>
<td>1.58</td>
<td>0.050</td>
<td>0.315</td>
<td>1.63</td>
</tr>
<tr>
<td>400 ppm Xanthan</td>
<td>7.27</td>
<td>1.67</td>
<td>0.042</td>
<td>0.331</td>
<td>1.81</td>
</tr>
<tr>
<td>600 ppm Xanthan</td>
<td>16.69</td>
<td>1.82</td>
<td>0.079</td>
<td>0.363</td>
<td>2.16</td>
</tr>
<tr>
<td>1200 ppm Xanthan</td>
<td>125.81</td>
<td>2.36</td>
<td>0.409</td>
<td>0.363</td>
<td>3.78</td>
</tr>
<tr>
<td>Human Blood</td>
<td>56.00</td>
<td>3.45</td>
<td>1.007</td>
<td>-0.28</td>
<td>---</td>
</tr>
</tbody>
</table>

Table 4.6  A comparison of non-Newtonian fluids: Cross parameters
4.1.2 Viscosity Degradation

Having established the dilute nature of the xanthan gum solutions and the appropriate concentrations to be used in the LSRT facility, it was critical to verify the solutions' viscosity stability. The test fluids had to maintain constant properties for the duration of the tests, especially the viscosity characteristics. Viscosity breakdown results when the fluid endures influences that act to fissure the polymer chain networks. Two major factors are a concern for viscosity breakdown in this study: biological decomposition and shear-induced breakdown.

![Figure 4.4](image.png)

Figure 4.4  Aging effects on a 400-ppm xanthan gum solution
The biological decomposition was tested by performing aging tests on a laboratory-grade (BioChemika) xanthan gum made by Fluka®. The viscosity characterizations were performed on the same 400-ppm solution that was left open to the air. The results are shown in Figure 4.4. These data show that the useful life span of aqueous xanthan gum, without other additives, is about three weeks. Within this timeframe, very little change in the viscosity behavior is measured. Beyond 18 days, the viscosity shows signs of breakdown. After 31 days, a noticeable odor indicates biological decomposition.

![Figure 4.5 Prolonged shearing of a 600-ppm xanthan gum solution](image)

Sensitivity to prolonged shearing also tested the fluid's resistance to breakdown. These tests involved actual solutions used in the LSRT testing.

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which were made with Colony® xanthan gum. The results are shown in Figure 4.5 after the fluid was sampled during a series of impeller performance tests. According to these results, the xanthan gum used was very resistant to shear-induced breakdown. Moreover, the 13-hour duration of continuous shearing represents approximately two weeks' worth of real-time testing. This latter observation is compatible with the biological life span of the fluid samples.

4.2 LSRT NEWTONIAN IMPELLER PERFORMANCE

The first series of large-scale impeller tests was designed to establish the pump similarity behavior for a Newtonian fluid. For this effort, room temperature water (15 – 20°C) was used as the working fluid in the LSRT facility. To accomplish the performance testing, the flow rate was independently controlled using the loop circulation pump in conjunction with the downstream orifice plates, while the resulting pump head rise was measured for each impeller speed. This procedure is different than for the IVAS tests, where the pump was operated at a set speed while the flow rate and head rise were simultaneously measured. To obtain the Reynolds numbers listed (up to 436,000), the water tests were run at impeller speeds of 0 through 50 rpm.

An example of the data is shown in Figure 4.6 and Figure 4.7. The first shows the dimensional pump curve data for impeller #3386 at an endwall gap of 0.9 mm, the smallest clearance tested. These data show a marked rising head characteristic behavior, especially at higher rotor speeds, and thus indicate that
the pump peak efficiency may occur at moderate values of flow rate. The rising head characteristic means the pump head actually increases as the flow rate is increased from the shutoff condition. This is due to the hydrodynamic efficiency of the volute such that its contribution to the total pump head rise is higher at nonzero flow rates. Despite the quantitative and apparent qualitative changes in the pump curve with varying rotor speed, the data collapses onto a common similarity curve as shown in Figure 4.7. This behavior holds true despite the fivefold increase in impeller Reynolds number, which correspond to 50, 40, 30, 20, and 10 rpm, respectively. A speed of zero rpm is not represented here because it results in undefined values of pressure and flow coefficient.

Figure 4.6 LSRT impeller #3386 Newtonian pump curve, $c_w=0.9$ mm
Figure 4.7 LSRT impeller #3386 Newtonian similarity, $c_{ew}=0.9$ mm

The rest of this section will describe the on-design conditions of the IVAS and LSRT pumps, based on the dimensionless performance coefficients and similarity parameters, and the geometry effects of impeller design and endwall clearance on the overall pump performance.

4.2.1 On-Design Characteristics

The design point of the IVAS pump has been established at a Reynolds number of 90,000 (true-scale speed of 2,850 rpm) and specific speed of 0.4.\* These two similarity parameters relate to the dynamic scaling and thus must be produced by the LSRT to match the IVAS design point. This was accomplished

\* These criteria are based on CCF requirements and performance of early prototypes.
by interpolating the pump curve data in Figure 4.6 to yield the operating conditions (flow rate and impeller speed) at specific speeds of 0.25, 0.39, and 0.55. These results, shown in Figure 4.8, demonstrate the linear behavior between flow rate and speed for constant values of specific speed.

![Figure 4.8 Impeller #3386 Newtonian operation chart, $c_{ew}=0.9$ mm](image)

By using this chart, the LSRT can be operated such that the conditions match the dynamic similarity of the IVAS pump. If the LSRT is operated anywhere along the middle curve of Figure 4.8, the specific speed will be equal to the design-point value of 0.4. The Reynolds number, which depends on only one performance parameter (impeller speed), determines the LSRT design speed. In the LSRT, the size is increased to 916% of the true-scale value and the
viscosity is decreased to 27%, such that the LSRT design speed is 10.4 rpm. Thus, dynamic similarity is maintained between the IVAS design point and the LSRT design point. Comparisons of the true-scale and large-scale pump design points are summarized for impellers #3386 and #3452 in Table 4.7 and Table 4.8.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>IVAS Impeller</th>
<th>LSRT Impeller</th>
</tr>
</thead>
<tbody>
<tr>
<td>Scale Factor</td>
<td>1.000</td>
<td>9.162</td>
</tr>
<tr>
<td>Test RPM</td>
<td>2,850</td>
<td>10.36</td>
</tr>
<tr>
<td>Flow Rate (LPM)</td>
<td>4.51</td>
<td>16.58</td>
</tr>
<tr>
<td>Head Rise (mmHg)</td>
<td>101.2</td>
<td>0.123</td>
</tr>
<tr>
<td>Reynolds Number</td>
<td>90,000</td>
<td>90,000</td>
</tr>
<tr>
<td>Specific Speed</td>
<td>0.398</td>
<td>0.394</td>
</tr>
<tr>
<td>Flow Coefficient</td>
<td>0.063</td>
<td>0.083</td>
</tr>
<tr>
<td>Head Coefficient</td>
<td>0.542</td>
<td>0.660</td>
</tr>
</tbody>
</table>

Table 4.7 Comparison of impeller #3386 Newtonian design point

<table>
<thead>
<tr>
<th>Parameter</th>
<th>IVAS Impeller</th>
<th>LSRT Impeller</th>
</tr>
</thead>
<tbody>
<tr>
<td>Scale Factor</td>
<td>1.000</td>
<td>9.162</td>
</tr>
<tr>
<td>Test RPM</td>
<td>2,850</td>
<td>10.36</td>
</tr>
<tr>
<td>Flow Rate (LPM)</td>
<td>4.03</td>
<td>15.52</td>
</tr>
<tr>
<td>Head Rise (mmHg)</td>
<td>93.5</td>
<td>0.116</td>
</tr>
<tr>
<td>Reynolds Number</td>
<td>90,000</td>
<td>89,000</td>
</tr>
<tr>
<td>Specific Speed</td>
<td>0.400</td>
<td>0.397</td>
</tr>
<tr>
<td>Flow Coefficient</td>
<td>0.057</td>
<td>0.078</td>
</tr>
<tr>
<td>Head Coefficient</td>
<td>0.500</td>
<td>0.625</td>
</tr>
</tbody>
</table>

Table 4.8 Comparison of impeller #3452 Newtonian design point
4.2.2 Effect of Impeller Geometry Variation

In addition to changing the impeller design, the endwall clearance with respect to the upper housing was varied to determine the effect on Newtonian similarity. This is an important study because a strong effect on the similarity laws will yield a strong effect on the scaling laws. Results from the variation in impeller design also produce clues for better designing the next impeller configuration. Impeller #3386 was tested at endwall clearances of 0.9 and 4.0 mm and impeller #3452 was tested at an endwall clearance of 3.1 mm.

Installing different impeller blades on the rotor disk varied the impeller geometry. This effort required breaking the seal on the upper and lower pump housings to remove the rotor disk, so complete tests involving Newtonian and non-Newtonian fluids were performed on impeller #3386 before #3452 was installed and tested. Changing the endwall clearance was a simpler procedure. Shims were inserted at the driveshaft coupler to vary the impeller vertical position in the housing.

Figure 4.9 shows the variation in dimensional pump curve data for the highest three impeller speeds. The results for impeller #3386 show that increasing the endwall clearance decreases the pumping effectiveness. The difference is very small at low flow rates but it increases drastically between about 40% and 80% of the maximum flow rate. More interestingly, the worst of the three configurations is impeller #3452, which was supposed to be an
improvement over design #3386. Even with an endwall clearance 0.9 mm less than that of impeller #3386, the pump performs at a deficit of 10% – 20%.

![Graph showing flow rate vs. pressure rise for different RPM and impeller designs](image)

**Figure 4.9** LSRT geometry effect on Newtonian pump curve data

Similarity behavior of the three configurations is shown in Figure 4.10. The relative dimensionless differences between the geometries are about the same proportion as in the dimensional pump curves. It is evident that each configuration has a unique similarity law. The flow and pressure coefficients are presented in Figure 4.11 and Figure 4.12. These plots are consistent with the similarity trends in Figure 4.10, showing the progressive decrease in pump performance, and that the flow and pressure coefficients are both functions of the specific speed. This functionality emphasizes the utility of similarity laws.
Figure 4.10  LSRT geometry effect on Newtonian similarity laws

Figure 4.11  LSRT geometry effect on Newtonian flow coefficient
These results firmly establish the Newtonian similarity behavior for all the geometries tested. Plots of the performance coefficients show that $v_s$ dictates specific values of $\phi$ and $\psi$, making it the single sufficient similarity parameter that scales the IVAS and LSRT pump performance.

4.3 LSRT NON-NEWTONIAN IMPELLER PERFORMANCE

Having established the LSRT pump similarity behavior for a Newtonian fluid, the next step was to perform identical measurements using non-Newtonian fluids. For this effort, room-temperature aqueous xanthan gum solutions having a strong shear-thinning region were used as the working fluid. This procedure was slightly different than for the Newtonian fluid tests, such
that a 60-rpm pump curve was added to the test matrix in order to keep the Reynolds numbers in the range of interest. This resulted in a maximum Reynolds number that depended on the concentration used: 268,000 for the 600-ppm xanthan gum and 224,000 for the 1,200-ppm xanthan.

The following two figures show a single non-Newtonian impeller test. Figure 4.13 and Figure 4.14 show the dimensional pump curve data and similarity behavior for impeller #3452 at an endwall gap of 3.1 mm, the only clearance tested for this impeller. These data reveal the same characteristic rising head behavior as the Newtonian tests. The dimensional data does not reveal any significant qualitative differences, and the pump curves appear very similar to the Newtonian curves for impeller #3452 in Figure 4.9.

![Figure 4.13 LSRT impeller #3452 non-Newtonian pump data, 600 ppm](image)
However, the dimensionless performance coefficients shown in Figure 4.14 reveal qualitative differences. The data no longer abides by the single similarity law that was sufficient for the Newtonian cases. Even at the Reynolds numbers that overlap, the similarity behavior is no longer valid. Only for values of the Reynolds number equal to or greater than 178,000 do the similarity laws appear to hold.

Based on these observations, a higher concentration of 1,200-ppm xanthan gum was prepared and tested. The goal of these tests was to evaluate the relationship between the xanthan gum concentration, which directly affects the degree of non-Newtonian character and elasticity of the fluid, and the
deviation from the similarity laws, which correspond to a particular critical (minimum) Reynolds number.

4.3.1 Effect of Fluid Concentration Variation

The subsequent fluid concentration tests involved progressively increasing the xanthan gum concentration from zero to 600 to 1,200 ppm for one impeller geometry. Impeller #3452 was tested at a single endwall rotor clearance of 3.1 mm using identical test matrices for each of the three fluid concentrations. This same range of xanthan gum concentrations was not reproduced for impeller design #3386. The results presented here will span only impeller speeds of 30, 40, and 50 rpm.

When comparing the pump curve data for water and 600-ppm xanthan gum, very little difference is detected. In fact, at speeds below 30 rpm, the differences are negligibly small and represent errors on the same order of magnitude as experimental scatter. Thus, results for impeller speeds below 30 rpm will not be presented here. As shown in Figure 4.15, the differences arising from the 600-ppm solution for impeller speeds above 30 rpm are on the order of a few percentage points when compared to the Newtonian curvefit line.

However, the results for the 1,200-ppm xanthan gum solution show significant dimensional differences, unlike the weaker non-Newtonian fluid. The deviation is as much as 30% lower than the Newtonian data for the 30-rpm pump curve, and the error appears to be independent of the flow rate. Another
observation is that the magnitude of the error is roughly the same regardless of
the impeller speed.

Figure 4.15 LSRT concentration effect on non-Newtonian pump curves

The impact on the pump performance coefficients is shown in the
comparisons of Figure 4.16, Figure 4.17, and Figure 4.18. It is clear that the non-
Newtonian fluids do not consistently follow the same similarity behavior as the
Newtonian fluids and that this deviation increases with concentration.

Therefore, the xanthan gum concentration has a progressive impact on
the criteria for similarity behavior. These results show that the critical Reynolds
number increases with the xanthan gum concentration, so that a greater range of
data lies outside of the region of similarity behavior.
Figure 4.16  LSRT concentration effect on non-Newtonian similarity

Figure 4.17  LSRT concentration effect on non-Newtonian flow coefficient
In addition, for the data represented above the critical Reynolds number, there appears to be different similarity laws in effect that are dependent upon the concentration. This uniqueness in similarity appears to be much like the deviation observed for the impeller geometry variations. This point is demonstrated by plotting only the data above the critical point, that which collapses onto a common similarity curve. For example, the points shown in Figure 4.19 are representative for data above 10 rpm for water, above 40 rpm for 600-ppm xanthan gum, and 60 rpm for 1200-ppm xanthan gum.
4.3.2 Departure From Similarity Behavior

It is desirable to determine with greater accuracy the point of departure from the similarity laws. The results presented to this point were based on varying the Newtonian and non-Newtonian Reynolds numbers by changing the impeller rotational speed. This was done, and data was generated for each of the three fluids and each of the three geometries by adjusting the impeller speeds between 0 and 60 rpm at intervals of 10 rpm. The results show that the point at which the similarity characteristic fails (i.e. the minimum, or critical Reynolds number) occurs at impeller speeds between 0 and 10 rpm for water. For 600-ppm xanthan gum solution, the departure point occurs between 30 and 40 rpm.
and for 1,200-ppm xanthan gum solution, between 50 and 60 rpm. This corresponds to Reynolds numbers in the range of 0–86,000 for water, 134,000–178,000 for 600-ppm xanthan, and 187,000–224,000 for 1,200-ppm xanthan gum solution.

Having obtained well-defined similarity curves for each fluid and each geometry, point measurement tests were conducted on impeller #3452 to more accurately assess the break point for similarity characteristics. Individual points were sampled in the impeller speed range of interest for the three fluids used in Section 4.3.1. The data points that were sampled using water as the test fluid are shown superimposed on the pump curve data in Figure 4.20. These points represents impeller speeds of 0, 2, 4, 6, 8, 10, 15, and 20 rpm.

![Graph showing data points for critical Reynolds number, water](image)

**Figure 4.20**  Data points for critical Reynolds number, water
The data were then reduced to dimensionless pressure and flow coefficients. Figure 4.21 shows the results for water. It is clear that the minimum Reynolds number is represented by an impeller speed of 8 rpm; the critical value is more accurately identified as 71,000. Above this critical Reynolds number, the data no longer abides by the Newtonian similarity law.

The same result is true for the non-Newtonian similarity laws. The 600-ppm xanthan gum solution is shown in Figure 4.22. These data points correspond to impeller speeds of 30, 32, 34, 36, 38, 40, 42, and 44 rpm, and the critical Reynolds number is approximately 170,000. Figure 4.23 shows the result for 1,200-ppm xanthan gum, which represent speeds of 46, 48, 50, 52, 54, 56, 58, and 60 rpm. The critical Reynolds number is approximately 194,000.

![Diagram](image-url)

Figure 4.21 Critical Reynolds number, water
Figure 4.22  Critical Reynolds number, 600-ppm xanthan

Figure 4.23  Critical Reynolds number, 1,200-ppm xanthan
4.4 VOLUTE PRESSURE DISTRIBUTIONS

Volute pressures were measured at selected flow conditions as another means to validate the large-scale pump results. Since CCF designed the volute for a specific operating condition of $v_{sp}=0.40$, the pressure distributions circumferentially around the volute should validate this design point. Furthermore, at conditions above and below the design point, specific trends in pressure distribution should be observed.

To accomplish these measurements, both impeller designs were tested at impeller Reynolds numbers of around 300,000. The specific speeds of 0.25, 0.39, and 0.54 were chosen to bracket the design point of the IVAS pump, and two trade studies were performed. The first study compared the Newtonian volute pressure distributions for the two different impellers. Tests were performed using water, and both impeller designs were operated at 34 rpm for varying flow rates. The second study involved a comparison between test fluids for one impeller design. Impeller #3386 was operated at 34 rpm in water and 60 rpm in 600-ppm xanthan gum while the volute pressures were measured.

The Newtonian volute pressure distribution is shown for both impellers in Figure 4.24. Impeller #3386 was tested at a Reynolds number of 303,000, and impeller #3452 was tested at a Reynolds number of 283,000. The plotted pressures are referenced to the inlet pressure, which was measured at the 0° circumferential position, and do not include the overall pump pressure rise, $\Delta p$. 

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These distributions show a relatively constant circumferential pressure difference at the design condition for both impellers, indicating that the impeller does the bulk of the work in raising the pressure. At the design specific speed, the pump rise for impeller #3386 is 1.31 torr with about 77% coming from the impeller (between the inlet and the 90° location). For impeller #3452, pump rise is 1.19 torr with 73% coming from the impeller.

Figure 4.24 LSRT geometry effect on Newtonian volute pressures

At off-design flow conditions, the ratios of pressure rise change. The specific speed of 0.25 represents a low flow condition that drops the percentage of impeller contribution to the pump head rise. In this case, the rotors account for 71% and 64%, respectively, of the total pump head rise. As a result, the
volute provides a larger contribution of the pressure, but the adverse pressure gradient in the direction of rotation makes this situation undesirable and thus prone to additional losses. In the case of high relative flow rates, the impeller performs too much work (94% and 89%, respectively) such that the volute acts as a diffuser, directly resulting in pressure loss.

![Graph showing the effect of viscosity on volute pressures](image)

**Figure 4.25** LSRT viscosity effect on volute pressures, impeller #3386

A comparison was also made, using the geometry of impeller #3386, between Newtonian and non-Newtonian fluids. The results of Figure 4.25 show the differences in pressure levels. The xanthan gum tests were conducted at 60 rpm, while the water tests were conducted at 34 rpm to achieve approximately the same Reynolds number. Although the pressure magnitudes are very
different, the trends remain the same with the design condition resulting in a relatively constant volute pressure distribution. The ratio of impeller/volute pressure contribution is the same between the two fluids, even though the xanthan gum tests produce about three times the overall pump head rise (3.88 torr versus 1.24 torr). Dimensionless pressure coefficients are plotted in Figure 4.26. Both tests yielded an overall pressure coefficient of 0.62.

Figure 4.26  LSRT viscosity effect on volute coefficients, impeller #3386

4.5 SURFACE FLOW VISUALIZATIONS

The flow visualization experiments were limited to the configuration involving impeller #3386, and were accomplished with dye injections through surface ports. These injection ports, located at the inner surface of the upper
pump housing and the upper surface of the rotor disk, have been described geometrically in Chapter 3. Flow patterns that are visualized here were obtained by utilizing various techniques with the CCD in still and video modes. The LSRT was operated in the similarity regime at specific speeds of 0.25, 0.39, and 0.55. Impeller speeds and flow rates required to achieve these specific speeds in Newtonian and non-Newtonian fluids are listed in Table 4.9. Flow visualizations were performed only in water and 600-ppm xanthan solutions; higher-concentration solutions were too cloudy to produce clear images of the flow patterns.

<table>
<thead>
<tr>
<th>Targeted Specific Speed</th>
<th>Water</th>
<th>600-ppm Xanthan Gum</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>RPM</td>
<td>LPM</td>
</tr>
<tr>
<td>$v_{Sp} = 0.25$</td>
<td>33.96</td>
<td>19.3</td>
</tr>
<tr>
<td>$v_{Sp} = 0.39$</td>
<td>33.96</td>
<td>51.9</td>
</tr>
<tr>
<td>$v_{Sp} = 0.55$</td>
<td>33.96</td>
<td>85.2</td>
</tr>
</tbody>
</table>

Table 4.9 LSRT speed and flow rate for visualization tests

4.5.1 *Pump Housing Dye Streak Patterns*

The pump housing dye injection ports are located in the endwall region, between 70 and 96% of the impeller radius. Two sets of four dye ports are located at 20° and 200° counterclockwise from the volute exit plane, when viewed from above. Dye streaks from these ports revealed patterns in the
surface flow behavior, which was under the influence of the rotational motion of
the passing impeller blades and the adverse pressure gradient produced by the
centrifugal forces through the impeller. These tests were done for two endwall
clearances (\(c_{ew}=0.9\) and 4.0 mm) and two fluids (water and 600-ppm xanthan).

To obtain reliable measurements of the surface flow angles, multiple CCD
still images were considered for digital averaging. Images were captured by
programming the CCD to obtain a series of 10 exposures over a span of 13
seconds. The total time span of the snapshots represented between 7 and 13
revolutions of the impeller, depending on which fluid was used. Image quality
was maximized to a digital resolution of 640x480 and 16-bit color was
implemented with a high sharpness setting using the CCD software. The
following color settings were used throughout these flow visualization tests
(maximum value is 256): brightness=129, white level=110, black level=151,
saturation=120, hue=automatic. The low light image filter was enabled and the
light meter was center-weighted.

The images were then digitally post-processed using PaintShop Pro™
(version 5.01) to obtain averaged data. Averaging the 10 images was
accomplished by performing 9 sequential arithmetic additions of each raw
image to obtain a composite image. Thus for each operation, two images were
digitally added, pixel-by-pixel for each color value, and then multiplied by a
factor of 0.5 to effect a time-averaged image. This process was performed with a
zero bias on all color channels. Color value clipping was unnecessary for these operations. Finally, post-processing the hue, saturation, and contrast levels helped to make the color dye streaks appear clearer for angle measurements, which were done using a draftsman’s protractor on the enhanced color printouts of the images. The multiple raw images were saved as bitmaps, but the post-processed composites were stored in compact JPEG format. It is estimated that the deviation in angle measurements using this technique is ±3°.

Digital video data was also produced in order to discern the unsteady nature of the housing surface flows. The CCD was programmed to record 30-60 seconds of data at approximately 10.1 frames per second. The total time span of the video represented between 17 and 60 revolutions of the impeller, depending on which fluid was used. Image quality was set to 640x480 resolution and 16-bit color (with high sharpness) using a VIDECC compression protocol, which permitted compact storage in AVI-formatted files. The color settings were: brightness=124, white level=96, black level=151, saturation=120, hue=automatic. The low light image filter was also enabled for the video mode.

The following flow visualization images were taken in water and show the averaged dye streaks, which were colored white to contrast them with the rest of the background image. These images were obtained at the design specific speed of 0.39. The impeller is seen to be rotating in the counterclockwise direction, with the center of rotation located above the image. Figure 4.27 and
Figure 4.28 show the streak patterns for the tighter endwall clearance at the 20° and 200° CCW positions, respectively. In these images, the second black line from the left is the radial reference line used for angle measurements at each circumferential position. In Figure 4.27, a wide white stripe at the left of the image indicates the 0° CCW position, and the radial discharge passage is seen branching off the volute in the lower right hand corner.

One of the more remarkable observances is that the dye streaks are spiraling inward toward the impeller inlet, not toward the outlet. The expectation is that the fluid would flow toward the impeller exit due to the net flow through the impeller. However, the angles of these dye streaks are all less than 90° (i.e. tangential flow) and it is apparent that the surface flow is heavily affected by the adverse radial pressure gradient. The pressure gradient, in which the pressure increases in the radial direction, results from the centrifugal nature of the pump. Figure 4.28 shows that the tendency for inward flow is slightly greater at 200° than at the 20° CCW position. This indicates flow asymmetry.

The resulting dye streak patterns for the larger endwall clearance are shown in Figure 4.29. When the endwall clearance is increased from 0.9 to 4.0 mm, the dye streaks reveal a much more pronounced inward flow. This would indicate that the viscous-dominated flow at the surface is relieved of the endwall effect, but still encounters the pressure gradient effect. Thus, the surface flow becomes less tangential and more radial inward.
Figure 4.27  Housing dye streaks at 20° CCW, $c_{ew}=0.9$ mm

Figure 4.28  Housing dye streaks at 200° CCW, $c_{ew}=0.9$ mm
In general, the flow angles depended first on the endwall clearance and second on the circumferential position. Very little difference was noted throughout the range of pump operating conditions that were included in this study. Table 4.10 and Table 4.11 show the angle measurements, indexed by radial position of the dye injection ports, for three specific speed values using water as a test fluid. These results show no clear trends regarding the variation in specific speed, and with a few exceptions, are within the precision of the measurement technique. Furthermore, the results for the xanthan gum solution were also within the precision of the measurements, showing no significant deviation between the two fluids. However, it must be noted here that these
flow visualizations were performed only within the regime of similarity behavior.

<table>
<thead>
<tr>
<th>r/R</th>
<th>V&lt;sub&gt;SP&lt;/sub&gt; = 0.25</th>
<th>V&lt;sub&gt;SP&lt;/sub&gt; = 0.39</th>
<th>V&lt;sub&gt;SP&lt;/sub&gt; = 0.55</th>
</tr>
</thead>
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</tr>
<tr>
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<td>78</td>
<td>80</td>
</tr>
<tr>
<td>0.87</td>
<td>87</td>
<td>82</td>
<td>81</td>
</tr>
<tr>
<td>0.96</td>
<td>90</td>
<td>89</td>
<td>84</td>
</tr>
</tbody>
</table>

Table 4.10 Housing streak angles at 20° CCW, c<sub>e</sub> = 0.9 mm

<table>
<thead>
<tr>
<th>r/R</th>
<th>V&lt;sub&gt;SP&lt;/sub&gt; = 0.25</th>
<th>V&lt;sub&gt;SP&lt;/sub&gt; = 0.39</th>
<th>V&lt;sub&gt;SP&lt;/sub&gt; = 0.55</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.70</td>
<td>78</td>
<td>80</td>
<td>81</td>
</tr>
<tr>
<td>0.79</td>
<td>73</td>
<td>78</td>
<td>75</td>
</tr>
<tr>
<td>0.87</td>
<td>69</td>
<td>78</td>
<td>76</td>
</tr>
<tr>
<td>0.96</td>
<td>74</td>
<td>85</td>
<td>81</td>
</tr>
</tbody>
</table>

Table 4.11 Housing streak angles at 200° CCW, c<sub>e</sub> = 0.9 mm

4.5.2 Rotor Disk Dye Streak Patterns

While the housing flow visualizations of Section 4.5.1 were performed with respect to a fixed reference frame, the rotor flow visualizations were performed with respect to the rotating reference frame. Dye was injected through three of the four dye ports in the rotor disk surface (at 70, 79, and 87%
of the impeller radius) to capture flow patterns between two adjacent impeller blades. The dye was pumped into the impeller driveshaft manifold using the four-component dye injection system. In this case, analog videos were produced, rather than digital still/video images, using an untethered compact video camera that was mounted to the impeller driveshaft. With the camera positioned below the LSRT housing, an unobstructed view up through the lower seal plate and rotor disk was possible.

The analog video footage was converted to digital still and movie formats. First, the compact Video8 format was converted to VHS format for use in standard VCR equipment. Second, an analog video workstation with a Miro® PC-board digitizer was utilized to capture individual JPEG frames and convert the VHS format to digital AVI format. Still images were captured at full-frame size (640x480 resolution) and were post-processed to enhance the color dye streaks. The digital videos were rendered at a 320x240 resolution and a 30 frames-per-second rate. Easily-obtained Intel Indeo® video decompression filters (version 5.04) were utilized to create movie files of reasonable size (less than 1.4 MB). The duration of the movies are 4-5 seconds (between 2 and 5 rotations of the impeller).

Examples of the digital images that were converted to 8-bit grayscale coloring are shown in Figure 4.30, Figure 4.31, and Figure 4.32. The figures show the rotor disk surface flow (viewed from below) in water for three different
specific speeds. Since the impeller is rotating in a clockwise direction, the fluid appears to be flowing in a counterclockwise direction from the perspective of the rotating video camera. The dye injection tubes can be clearly see in the figures, as can a portion of the dark inlet region (lower left-hand corner). The far-right dye injection tube was not used in these tests. Along the left of each figure is one impeller blade; another is seen in the lower right-hand corner.

The dye streak images show a clear dependence on the pump operating conditions and lower specific speeds correlate with higher degrees of flow recirculation in the blade passage. This observation is especially poignant for the first and second dye ports near the pressure side of the left impeller blade. At a specific speed of 0.25, the flow near this blade is outward away from the blade surface in a slightly upstream orientation. As the specific speed is increased to the design point of 0.39, the recirculation zone near the left blade decreases, although the second dye streak still exhibits a largely tangential flow direction (see Figure 4.31). However, at the highest specific speed of 0.55 as shown in Figure 4.32, the dye streak patterns reveal a much greater degree of outward radial flow. This is the commonly accepted mode of operation for centrifugal pumps at higher specific speeds. The third dye port from the left shows backward tangential flow patterns at all specific speeds, indicating an influence from the right impeller blade. This blade apparently produces strong separation regions on the suction side that do not dispel at the higher pump flow rates.
Figure 4.30  Rotor disk water dye streaks, $v_{sp}=0.25$

Figure 4.31  Rotor disk water dye streaks, $v_{sp}=0.39$
Figure 4.32  Rotor disk water dye streaks, $v_{sp} = 0.55$
CHAPTER 5

DISCUSSION OF THE RESULTS

The results of the non-Newtonian performance tests in the LSRT have demonstrated a measurable shift in the dynamic scale for a centrifugal impeller. While this is the primary issue and the main focus of the current study, many other factors related to this work are salient to the continuing efforts at the Cleveland Clinic to develop a viable LVAD system like the IVAS pump. These factors include the feasibility of pseudo-3D and true-3D computational fluid dynamics in a centrifugal pump, the validity of large-scale pump testing for prototype studies, and the determination and applicability of pump scaling laws for design iteration.

Often, scaling laws are used in design to size "rubber" pumps that are within 50% of the baseline configuration geometry. It is not very common to apply pump scaling laws when such disparate geometric sizes are present, mainly because of concerns about the shift in peak efficiency to higher flow rates for larger pumps. This shift in efficiency occurs because of the disproportionate scaling of the pump motor power and the hydraulic power required, but it will
not be considered in this discussion because power measurements were not conducted. Instead, only the hydraulic dynamic scale (that is, the flow through the pump) will be considered with respect to its impact on similarity.

Because the scale comparisons presented here are based on a 916% sizing of the impeller and volute geometry, the issue of scaling validity must be resolved first because it is expected to exert a greater impact on the performance predictions than the non-Newtonian effects. Therefore, in addition to presenting some three-dimensional computational results, this chapter discusses comparisons between the true-scale and large-scale pump performance and the impact of the geometric size on the applicability of Newtonian scaling laws. Once these have been established as valid for the purpose of this study, the magnitude of the non-Newtonian effects on pump scaling laws will be presented.

5.1 SOME 3D COMPUTATIONAL RESULTS

As part of the larger IVAS effort at OSU, an extensive computational effort was undertaken to simulate the three-dimensional Newtonian flow through the IVAS impeller and volute. Under the direction of Professor Shoichiro Nakamura, these efforts progressed from 1D bulk-flow analyses to 2D and pseudo-2D computations and finally to full 3D direct numerical simulations. Selected results from these very recent computations, which are currently in preparation for publication, will be summarized here to
demonstrate the complexity of the impeller and volute flow fields. The details contained herein, which include many particulars of the flow field previously unknown, will also support some of the assumptions made about the IVAS pump operation envelope.

The code development implemented solutions to the three-dimensional, unsteady, incompressible, Newtonian Navier-Stokes equations. Direct simulation has yielded solutions that resolve the scales of boundary layers, vortices, and the time-dependence of unsteady flows. Most importantly, the solutions were obtained with a significantly reduced value of the numerical viscosity, which is a necessary artifact in Navier-Stokes solution algorithms. The geometry modeled a single flow passage from the inlet eye, through a single IVAS impeller blade passage, and into a simplified model of the volute. The computational domain was 66x30x21 in dimension in the radial, azimuthal, and axial directions, respectively.

Velocity vectors contained in a horizontal plane near the blade endwall region are shown in Figure 5.1. The axial index is 16 of 21 cells distributed across the blade span; this is located at the blade upper edge. The counter-clockwise vectors at the right are relative volute velocities with respect to the rotating impeller. In the laboratory frame of reference, the impeller would be rotating in a clockwise direction. As such, the upper boundary is a blade pressure surface
and the lower boundary is a blade suction surface. The cluster of vectors along the left side of the flow domain represents the inlet flow at the impeller eye.

Figure 5.1  Horizontal flow plane near blade upper edge [reproduced from Nakamura and Ding 1999]

This diagram shows the essentially chaotic flow that occurs in the impeller blade passage near the endwall region. One example is the separation bubble that is evident at the leading edge of the suction surface. The bubble is short, reattaching quickly, but induces a jet onto the opposing pressure surface. However, the pressure surface has an apparent separation zone at about the
half-chord location. This local flow pattern agrees very well qualitatively with the rotor disk surface flow visualizations of Figure 4.31 on page 138. Finally, the mixing between the blade passage flow and the volute flow is very non-uniform. It seems that effective discharge flow is present in only about 50% of the blade passage at the impeller discharge plane.

![Vertical flow plane halfway between blades](image)

**Figure 5.2** Vertical flow plane halfway between blades
[reproduced from Nakamura and Ding 1999]

A vertical cross-section of the velocity vectors between blades is shown in Figure 5.2. The azimuthal index is 15 of 30 cells distributed across the blade
passage. This view shows the dramatic formation of the Ekman layer near the inlet plane. This separated, reverse-flow, annular region of high angular velocities results from the low-specific-speed, low-flow operating conditions of the pump. Although it can be amplified or minimized, the Ekman layer flow characteristic never disappears and contributes significantly to the operating point of best efficiency.

These computational results are significant from the standpoint of qualitative flow behavior. Although they were performed using a Newtonian fluid assumption, it is expected that the net flow patterns are reasonably represented. The remaining discussion in this chapter will be based on the understanding obtained from these computational results and will draw on some of the flow features depicted here.

5.2 NEWTONIAN COMPARISON OF SCALE-PUMP DATA

CCF made true-scale pump performance data available, as presented in Appendix D, and a comparison with the large-scale data is desirable. Thus, the results for impellers #3386 and #3452 are compared for different Newtonian fluids. The LSRT data is based on using room-temperature water as a working fluid, while the IVAS data involves a 40/60 ratio of glycerin and water at 37° C. Some portions of the LSRT data discussed here have also been presented in Chapter 4. These scale comparisons of the performance data are shown for both impeller designs in the next two figures.
The scale comparison for impeller #3386 is shown in Figure 5.3. In this case, the LSRT data is represented by performance coefficients based on an endwall clearance of 4.0 mm. Only impeller speeds of 10, 20, and 30 rpm are depicted at flow rates of 0–140 lpm. The specific gravity of 0.998 and viscosity of 1.02 cP are based on a reservoir water temperature of 19.4° C. This yielded Reynolds numbers in the range 87,000–260,000 as shown in the plots. On the other hand, the IVAS data represents test conditions of 2,500, 3,000, and 3,500 rpm and flow rates of 0–13.6 lpm. The fluid had a specific gravity of 1.113 and a viscosity of 3.7 cP, which resulted in Reynolds numbers of 79,000–111,000.

![Newtonian scale comparison of impeller #3386](image)

Figure 5.3 Newtonian scale comparison of impeller #3386
Figure 5.4 shows the scale comparison for impeller #3452. The LSRT data is represented here by performance coefficients based on an endwall clearance of 3.1 mm, the only clearance tested. The same impeller speeds of 10, 20, and 30 rpm and flow rates of 0–137 lpm are shown. The reservoir water temperature was 18.8°C, resulting in a specific gravity of 0.998 and viscosity of 1.03 cP. Thus, the plotted Reynolds numbers are 86,000–257,000. IVAS data represents similar test conditions as previously, at flow rates of 0–12.1 lpm. The fluid had a specific gravity of 1.113 and a viscosity of 3.52 cP, which resulted in slightly higher Reynolds numbers of 83,000–116,000.

![Newtonian scale comparison of impeller #3452](image)
Note for these comparisons the overlap in Reynolds numbers of 87,000–111,000. This overlap ensures the dynamic similarity is matched such that the corresponding similarity laws can be compared. Each scale pump demonstrates a clear trend in similarity behavior for a range of Reynolds numbers. A range of specific speeds of 0–2 is represented in each case. In these figures, the discrepancy between scale pumps is as low as 8% and as high as 30%. Although the large discrepancy is of some concern, there are some purely scale-related issues that influence the comparison of the two pumps. Differences in relative surface roughness, scale tolerances, inflow and outflow connection geometry, and secondary impeller flow modeling can affect the performance considerably. However, one vestige of flow anomaly was unaccounted for in the LSRT tests that was conspicuous in the operation of the IVAS pump. Preswirl was noticeable in the inflow conduit of the IVAS device, but the LSRT flow conditioner largely repressed this effect. The issue of pump losses and the differing effects on the scale pumps will be addressed in Sections 5.2.1 and 5.2.2.

5.2.1 Prerotation Corrections

The effect of preswirl or prerotation is to decrease the impeller’s pumping effectiveness. This concept has been demonstrated in the discussion of Euler’s equation for an impeller with the inclusion of inlet swirl velocities (see Section 2.2.1.2 on page 23). When the impeller induces an upstream rotation in the inlet flow, the overall addition of angular momentum through the impeller is
decreased. More critically, the relative rotational velocity decreases the inlet flow angles so that the blading is less effective at imparting the angular momentum, as does decreased incidence on an isolated airfoil. Thus, the impeller exerts a lower overall torque on the fluid at lower flow rates.

Prerotation corrections were thus applied to the LSRT data in an effort to reconcile the differences between the LSRT and the IVAS measurements. Of the factors that adversely affect the shutoff head (zero-flow pump pressure rise), namely prerotation and flow through the secondary impeller, the prerotation is assumed to be the larger influence. Considering the very low flow rates through the secondary impeller, less than 1% of the pump flow, this assertion is valid. Therefore, prerotation corrections were applied to the LSRT data to match the shutoff head between the two scale pumps. The corrections were used according to the prescribed procedure as presented in Chapter 2. For LSRT impeller #3386, adding a 45% prerotation effect was sufficient to match the shutoff head of IVAS impeller #3386, and this result is shown in Figure 5.5.
Figure 5.5 Corrected Newtonian scale comparison of impeller #3386

Figure 5.6 Corrected Newtonian scale comparison of impeller #3452
The percentage of prerotation is determined by the prerotation factor, f. This factor describes the magnitude of the swirl velocity with respect to the impeller rotation, which represents a theoretical maximum amount. Therefore, the 45% prerotation assumed for impeller #3386 indicates that the fluid is rotating at 0.45 times the rate of the impeller at the eye diameter. Due to the high degree of viscous effects, this theoretical limit is never achieved in practice.

The level of prerotation required for LSRT impeller #3452 was slightly higher at 50%, and the corrected data is shown in Figure 5.6. This fact, which could indicate that IVAS impeller #3452 demonstrates a slightly higher level of prerotation than #3386, would also justify the inferior performance of the former when compared to the latter.

Preswirl corrections to the LSRT data seem to account for some of the discrepancies between the two scaled pumps. While pump performance plots agree better only at low-flow conditions, the isolated pressure coefficients agree better over the entire operating range of the pumps. The resulting corrected pressure coefficients are plotted against the specific speed in Figure 5.7 and Figure 5.8. The overall agreement is much improved.
Figure 5.7  Corrected Newtonian pressure coefficient of impeller #3386

Figure 5.8  Corrected Newtonian pressure coefficient of impeller #3452
Prerotation corrections are justified by the known presence of preswirl in the IVAS tests, although the magnitude is currently unknown. However, the levels of prerotation that result in good agreement, 45% and 50% of the rotational velocity at the impeller eye, do not represent reasonable stipulations about the inlet flow quality in the small-scale pumps. These values are extremely high for pumps of this type, and prerotation cannot be the only major influence in the performance discrepancies. There appear to be additional contributions to the pump losses that may account for the scale differences in flow coefficient, if not the pressure coefficient, especially at higher flow rates (30–70% of the maximum flow coefficient).

5.2.2 Additional Pump Losses

Several other differences between the two scale pumps may contribute, in part, to the performance discrepancy. Of these, the factors of relative surface roughness, scale tolerances, and inflow/outflow connection geometry would affect the pump head at nonzero flow rates. This is because these losses are proportional to the square of the bulk velocity in the same manner as pipe losses. It is acknowledged that all of these factors are present in the current scale pump tests.

The surface finish of the IVAS pump was not accurately scaled in the LSRT pump. In fact, these aspects actually improved by at least an order of magnitude. The IVAS data in Appendix D was generated using prototype pump
housings and impeller surfaces that were polished to the specification of a 14G surface finish. The corresponding surface roughness is thus a standard deviation of at most ±14 microns and the roughness-to-diameter ratio is 1.12x10⁻⁵. By contrast, the LSRT surfaces were polished to an estimated 8L surface finish with a corresponding surface deviation of ±8 microns. This means the roughness-to-diameter ratio is 7.00x10⁻⁷, or more than 16 times proportionately smoother than the IVAS surfaces. In the case of turbulent pipe flows, a roughness decrease of 16X can cause a 10-20% decrease in friction factor. Furthermore, the roughness losses in a pipe flow will vary with the square of the velocity, so higher flow rates will produce much higher pressure losses. Therefore, decreasing the scaled surface roughness could significantly ameliorate the expected flow losses through the impeller and housing, particularly at higher flow rates.

Also, the tolerances were not accurately scaled. In general, the tolerance dimensions used to specify the fabrication of the IVAS pump were matched in fabricating the LSRT pump. This means that scaled part tolerances are improved by a factor of more than 9, and losses associated with part fit are reduced in the same manner as with surface roughness.

It has been assumed that the geometry of the inflow and outflow conduits also contributes to the pump performance differences. The IVAS prototype testing involved quick-disconnect fittings at the inflow and outflow conduits. These connections create minute differences in the flow passage dimensions.
immediately upstream and downstream of the pump. At the inlet, a slight increase followed by a sharp decrease in the conduit diameter is evident just upstream of the quick-disconnect fitting. In the pump housing, the flow passage decreases about 7% at the inlet plane before increasing again through the impeller inlet region. Overall, the conduit diameter is about 114% of the inlet diameter. The outflow connection exhibits a sudden expansion at the discharge plane that could adversely affect pump output. The volute discharge passage increases to its maximum diameter and maintains a constant section for 0.22 diameters before the quick-disconnect fitting. At this point, there is a sudden expansion to 127% of the discharge diameter. Then the diameter gradually decreases to about 116% of the discharge diameter. This fitting thus acts as a flow diffuser, creating a higher static pressure at the discharge as a result of the larger conduit, but there is a dynamic pressure loss associated with the sudden viscous expansion. This added backpressure could minimally reduce the pump flow rates. Overall, the inflow and outflow geometries of the IVAS prototype pumps may have contributed minimally, if not negligibly, to the differences in measured performance.

However, installed performance of the IVAS pump is expected to be measurably reduced. The inflow and outflow assemblies #3566 and #3567 as shown in Figure 5.9 represent the implantable configuration. In addition to the possible performance losses just outlined, further complications of the flow
quality in the inflow conduit could originate in the upstream cannula tip and the sharp 120° bend in the tube. This geometry would likely induce strong secondary flow patterns in the bend that would propagate downstream to the inlet plane location. Without a flow conditioning device, these secondary flows would definitely exacerbate the degree of preswirl at the inlet and possibly enhance the tendency for unsteady, separated, recirculating inlet flow such as the Ekman layer discussed in Section 5.1.

Figure 5.9 IVAS outflow and inflow conduits

For the LSRT tests, losses generated by inflow and outflow conduits have been minimized by the design and fabrication of the facility. By sizing the LSRT
to 916% of the IVAS prototypes, the diameter at the inlet and discharge planes coincided with the inner diameter of standardized 10.16-cm (4.0-inch) schedule 80 PVC pipe. At the inlet, the inflow pipe was bonded into the pump housing and finished for a hydraulically smooth transition. This short section is the dark pipe mounted to the inlet in Figure 3.5 on page 69. Although the union fitting introduces a small amount of flow disturbance, there is no converging-diverging geometry in the inflow pipe. The inline flow conditioner is directly upstream of the union fitting, so the flow through the inlet region is expected to be uniform.

The LSRT discharge passage is designed to be equally smooth. At the end of the volute discharge passage (i.e., the point of constant area), a flange is bolted to the pump housing. A section of PVC pipe is bonded to this flange, and the flange inner diameter matches the discharge diameter within the scaled tolerance of the pump (refer to Figure 1.7 on page 13). This creates a smooth transition from the volute discharge to the discharge pipe in a similar manner as the inlet pipe. It is thus expected that the velocity in the pipe matches the velocity at the discharge plane.

Secondary impeller flow modeling can also affect scaled performance because it contributes to pump pressure losses. The IVAS pump has a secondary impeller to promote a small amount of lubricating flow through the journal bearing. The secondary impeller draws less than 1% of the flow down through the journal bearing between the fixed centerbody and the rotor assembly and
pumps it back up the annular passage between the rotor assembly and fixed outer wall. This return passage is shown in the cross section in Figure 5.9 at the periphery of the impeller. However, the LSRT does not model this geometry. The journal bearing is not modeled, and the nosecone rotates with the impeller disk, as described in Chapter 3. Furthermore, the return passage is not modeled, being replaced by a radial extension of the rotor disk at the entry to the volute passage. This occurs at the impeller discharge plane. Having the secondary impeller flow could contribute theoretically to a loss in pump head, although this effect is likely to be very small when compared to a rotor assembly without a secondary impeller flow. It is the other, larger contributors to the performance deficit, as addressed in the previous sections, that could affect the applicability of the pump scaling laws.

5.2.3 Applicability of Pump Scaling

All of the losses discussed in the previous two sections bring into question the validity of the pump scaling laws for Newtonian fluids. And, although several possible sources for error have been qualitatively if not quantitatively delineated, it still remains to be seen if the LSRT engenders adequate performance for application of Newtonian pump scaling laws. This issue will now be addressed.

Within the specified operating envelope of the IVAS pump, the LSRT presents satisfactory scaled-performance results with respect to the normalized
pressure (pressure coefficient); for simulating the design point, the LSRT presents excellent scaled performance results. The latter point will be discussed first. As presented in Section 4.2.1, the design point of both IVAS pumps corresponds to a specific speed of about 0.4, which establishes the design flow coefficient of about 0.06 and the design pressure coefficient of about 0.50. At this specific operating point, the LSRT data corrected for finite preswirl shows good agreement compared to the IVAS data at the design point. Both LSRT impeller #3386 and #3452 corrected data is within 4% of the IVAS data. Considering that the differences in friction factor and slight variations in geometry remain unaccounted for, this conformity is accepted as excellent agreement. At pump operation below the design point, the agreement between the two scaled pumps is extremely close as a result of the preswirl procedure that corrected the data by matching the shutoff head.

The upper end of IVAS pump operation is defined by a specific speed of about 0.7 for both impeller geometries. This corresponds to a flow coefficient of 0.12 and a resulting lower pressure coefficient of 0.40. At these conditions, the deviation in the scaled pump pressure coefficient is about 11% for impeller #3386 and 10% for impeller #3452. This agreement is considered very good in the absence of further data corrections.

Based on the good agreement in pressure coefficient between the two scaled pumps, the applicability of the similarity scaling laws has been
established for a Newtonian fluid, although it appears to be somewhat less reliable for flow coefficient predictions. The LSRT data has been corrected to account for a known preswirl present in the IVAS data, but no other corrections have been applied despite the differences in surface roughness. Now that the pump scaling laws have been established for Newtonian fluids, the discussion will shift to the magnitude of the effects of a non-Newtonian fluid having shear-thinning properties.

5.3 NON-NEWTONIAN EFFECTS ON PUMP SCALING

It has been widely accepted that the effects of a non-Newtonian fluid exert a second-order influence on the flow field in a rotodynamic LVAD. Based on Bodonyi's boundary layer work presented in Section 2.3.4 on page 49, a strong shear-thinning fluid similar to blood does not influence shifting the position of the separation point. At most, shear-thinning fluids precipitate significant changes in boundary layer thickness and skin friction coefficient, but nothing yet suggests a major deviation in laminar-to-turbulent transition point or boundary layer separation location. Therefore, it is expected from these calculations that a developing boundary layer in an otherwise inviscid flow would have a noticeably different form for a non-Newtonian fluid than a Newtonian one. The thicker boundary layer and higher skin friction would create measurable differences in the surface pressure distribution. However, for
a fully-developed pipe flow that is viscous-dominated in character, these effects 
would be expected to be minimal and probably immeasurable.

However, the results of Chapter 4 have suggested three ramifications of 
implementing a shear-thinning fluid in a large-scale centrifugal pump model. In 
these points, the Weissenberg number* is used to describe the degree of non-
Newtonian character:

- For each value of the Weissenberg number, a unique similarity law exists.
- The critical Reynolds number increases with Weissenberg number.
- The deviation from the Newtonian similarity law increases with the 
magnitude of the Weissenberg number.

Results presented in Chapter 4 show that above a critical impeller speed, 
similarity behavior is exhibited, and that this critical speed increases with the 
Weissenberg number. These observations are summarized in Table 5.1 and 
Table 5.2. Obviously, increasingly non-Newtonian (e.g. viscoelastic) fluids as 
indicated by the Weissenberg number effectuate flows with increasing degrees 
of viscous effects such that lower-Reynolds number phenomena are observed at 
higher actual Reynolds numbers. Specifically, the point of departure from 
Newtonian similarity at a Reynolds number below 71,000 is observed at higher 
Reynolds numbers in the viscoelastic fluids, namely 170,000–194,000. Another

* Note that the Weissenberg number is proportional to the rotational speed and the zero-shear 
viscosity of the fluid. In this application, it is useful as an indicator of non-Newtonian character 
because it is a stronger function of fluid concentration, which results in a 125-fold change in 
viscosity in this study, rather than impeller speed, which has a 7-fold variation in this study.
classic example of this type of behavior is observed with respect to freestream turbulence in a wind tunnel; turbulence generally tends to degrade an airfoil's performance in the same manner as higher Reynolds number effects. Thus, it would be possible for data acquired at a higher turbulence level and lower Reynolds number to be dynamically similar to data acquired at a lower turbulence level and higher Reynolds number.

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Min. RPM</th>
<th>Min. We</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>8</td>
<td>0.00</td>
</tr>
<tr>
<td>600 PPM Xanthan</td>
<td>38</td>
<td>5.07</td>
</tr>
<tr>
<td>1200 PPM Xanthan</td>
<td>52</td>
<td>57.09</td>
</tr>
</tbody>
</table>

Table 5.1 Conditions for similarity behavior: Weissenberg number

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Minimum RPM</th>
<th>Minimum Reynolds Number</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Based on η₀</td>
</tr>
<tr>
<td>Water</td>
<td>8</td>
<td>71,000</td>
</tr>
<tr>
<td>600 PPM Xanthan</td>
<td>38</td>
<td>44,000</td>
</tr>
<tr>
<td>1200 PPM Xanthan</td>
<td>52</td>
<td>4,000</td>
</tr>
</tbody>
</table>

Table 5.2 Conditions for similarity behavior: Reynolds number

The third ramification of a shear-thinning fluid is that the deviation from Newtonian similarity progressively increases in magnitude as the Weissenberg number increases. This effect is seen in Figure 4.19 on page 121, and the data is cropped in Figure 5.10 to show the deviation in pump similarity over the
operating range of the IVAS pump. Based on the smooth curve fits through the data, the similarity law for 600-ppm xanthan gum deviates 0–3% from the Newtonian curve. For the higher-concentration xanthan gum, the deviation from the Newtonian similarity law is 4–11%. Regardless of the definition of the Reynolds number used, the same conclusions can be made: increasing the concentration of the xanthan gum solution increases the viscous interaction in the overall flow field and is manifest in the reduction of the pressure coefficient, the reduction of the flow coefficient, or both.

![Graph](image)

Figure 5.10 LSRT non-Newtonian effects in pump operating envelope
5.3.1 The Magnitude of Non-Newtonian Effects

This study documents that measurable differences in pump performance exist between Newtonian and non-Newtonian fluids. Thus far, the results show that the similarity laws are affected and the magnitude of deviation depends on the level of "non-Newtonian character," as it has been discussed at the beginning of Section 5.3. Now the magnitude issue will be addressed, because it is evident that the non-Newtonian effects in a centrifugal heart pump model are decidedly more influential than the historical second-order assumption that has been made in the literature.

With respect to the impact on the similarity laws, the higher-concentration xanthan gum solution engendered at most an 11% performance deficit over the range of operating conditions of the pump. Depending on the other parameters that affect the performance, this deficit may or may not be of first-order importance, despite the significant percentage. To compare with other factors affecting performance, the geometry trade studies were considered. The geometry effect is seen in Figure 4.10 on page 113; this data is cropped in Figure 5.11 to show the deviation in pump similarity over the same operating range as represented in Figure 5.10. This shows a deviation of approximately 10% resulting from first-order variations in the impeller geometry. The geometry changes were significant in magnitude: increasing the blade camber from $-2.5\%$ to $4.3\%$ and increasing the endwall clearance by over 300%. 

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5.3.2 The Appropriate Reynolds Number Definition

With the results and conclusions presented thus far, the issue of Reynolds number definition must now be elaborated. While the critical Reynolds numbers are listed in Table 5.2 using both reference viscosities, the results of Chapter 4 have been presented under the assumption of using the infinite-shear viscosity in defining the Reynolds number. This has been done based on the regime of pump operation and the subsequent shear magnitudes encountered in the impeller and volute.

For the specific application of the current study, the infinite-shear viscosity ($\eta_\infty$) is the most appropriate choice for defining the Reynolds number.
Mann and Tarbell (1990, 711) have suggested that arterial blood flows encounter average wall shear rates of over 100 sec\(^{-1}\) and much higher maximum values. Based on a blood viscosity model like that presented in Figure 2.12 on page 54, it may be assumed that circulatory flow is predominantly Newtonian in nature at a viscosity of 3.5–4.0 cP. This is not necessarily an accurate assessment for the xanthan gum polymer solutions used in centrifugal pumps like the IVAS and LSRT. At strain rates of 100 sec\(^{-1}\), the viscosity of the 600-ppm solution is 4.0 cP, which is within 15% of the infinite-shear value.* The viscosity of the 1,200-ppm solution is 11.9 cP at this strain rate, which is within 8% of the infinite-shear value. However, the shear rates in the centrifugal IVAS pump are expected to be much higher than the average arterial values even though isolated separation zones in the impeller will produce zero shear rates locally. If the assumption is implemented that IVAS shear rates are at least an order of magnitude higher, then the xanthan gum viscosities are within 5% of the infinite-shear values for each concentration.

In using the infinite-shear viscosity in the Reynolds number definition, the most appropriate blood analog fluid would be selected to match this value. The present xanthan gum solution of 1,200 ppm has an infinite-shear viscosity of 2.36 cP that is within 2% of the infinite-shear value for blood, thus making it the most appropriate analog fluid.

* The percentage is based on the difference in viscosity values as referenced to the difference between the zero-shear and the infinite-shear viscosity.
By employing the current definition for the Reynolds number based on the infinite-shear viscosity, the proportionate increase in viscous interaction with xanthan gum concentration becomes evident. It has already been established that high shear rates dominate the pump performance behavior. So the LSRT performance resulting from using the 1200-ppm xanthan gum solution as the test fluid will be within 11% of the performance of a Newtonian fluid having a viscosity of 2.36 cP. However, the non-Newtonian fluid will experience localized regions of low shear rate, such as at separation locations on the impeller blades, and the effect will be that of a higher local viscosity. Therefore, the effect of the sum of all of these localized regions is seen to be a first-order effect when considering the pump performance overall. As the concentration of the xanthan gum increases, the low-shear effects accumulate, thus accounting for the apparent increase in viscous interaction and the progressive departure from Newtonian similarity laws. This also accounts for the increase in the critical Reynolds number. At a given reference Reynolds number, a non-Newtonian fluid will contribute a higher degree of viscous effects than a Newtonian fluid, and the result is a departure from Newtonian similarity.
CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

In an effort to model the dynamic similarity of the Cleveland Clinic's Innovative Ventricular Assist System (IVAS) blood pump, the Large-Scale Rotor Testbed (LSRT) was developed and implemented at the Aeronautical and Astronautical Research Laboratory at The Ohio State University. With the LSRT tool, data was acquired on a 9X model of the true-scale pump, and the effects on the pump performance of various non-Newtonian fluids were experimentally assessed. Solutions of xanthan gum in water were used in 600–1,200 ppm concentrations to vary the degree of non-Newtonian qualities. These fluids proved to be effective and stable in simulating the shear-thinning viscosity behavior of whole blood.

Two 9X impeller geometries were tested at three different values of the endwall clearance in the same volute/housing (CCF design #3374). The LSRT impeller models (CCF designs #3386 and #3452) were operated at the same dynamic scale as the true-scale IVAS impellers. This corresponds to specific
speeds of 0.4–0.7 and yields flow coefficients of 0.06–0.12 and pressure coefficients of 0.4–0.5.

Both the tool (LSRT facility) and the task (experimental directives and procedures) have proven effective and fruitful and have demonstrated the potential for use in future research. These assets have combined to demonstrate that the non-Newtonian effects of a natural polymer solution are equal in magnitude to the geometry effects in a large-scale centrifugal heart pump model. The summary and conclusions of this study will be delineated in Section 6.1. Recommendations for facility improvements will be outlined in Section 6.2, while recommendations for further test programs will be presented last in Section 6.3.

6.1 SCALING LAW OBSERVATIONS

The overall goal of this documented research was to experimentally evaluate the non-Newtonian effects on the similarity laws in a centrifugal heart pump. Similarity laws are applied as scaling relationships in the design process. Thus, to achieve this goal, four objectives were sequentially met in the LSRT facility: observation of Newtonian similarity behavior, validation of Newtonian similarity laws for use in scaling, observation of non-Newtonian similarity behavior, and evaluation of the non-Newtonian effects on the scaling laws.

First and foremost, the LSRT facility promoted accurate and repeatable indications of Newtonian centrifugal pump similarity behavior. Routine
calibrations of the instrumentation were performed to ensure the accuracy of the measurements. Despite the wide range of operating conditions of impeller speeds from 10–50 rpm and flow rates from 0–210 lpm, resulting in Reynolds numbers of 80,000–440,000, the dimensionless performance parameters coincided with similarity behavior. The plots of the dimensionless pump curve (pressure coefficient versus flow coefficient) showed extremely defined similarity laws, with the data collapsing onto a common curve with a scatter of less than 2% of the zero-flow value. The quality of the data and repeatability of these observations is testimony to the effectiveness of the LSRT as a tool. However, these results are only efficacious if the 9X scaling laws of the LSRT correspond to the 1X scaling laws of the IVAS.

Second, the Newtonian scaling laws measured in the LSRT are in reasonable agreement with those measured in the IVAS prototypes. IVAS data as presented in Appendix D shows a higher degree of scatter in the data (3–4% on the dimensionless pump curve) but still exhibits a well-defined similarity behavior. However, an initial comparison of the dimensionless pump curve data shows an 8–30% difference over the operating range of the IVAS pump. One factor that impacts this comparison is the variation in the inflow quality, particularly the differences in inlet preswirl, between the two scale pumps. The LSRT has an inline flow conditioner to create uniform inlet flow, while the IVAS does not, resulting in significantly higher levels of prerotation in the latter.
pump. When a prerotation correction procedure is implemented for the LSRT data (inducing 45–50% preswirl), the errors in pressure coefficient scaling drop to 3–9% over the operating range of the IVAS pump. However, these are excessive and unrealistic levels of preswirl in the inlet. The errors in flow coefficient scaling remain unaffected by the corrections and are in the range 3–17%. Thus, the LSRT provides a reasonable representation of pressure scaling laws, when prerotation effects are taken into consideration, and a fair representation of flow scaling laws.

Similarity behavior was also observed for non-Newtonian fluids, corroborating the third stated objective. The Newtonian experiments used room-temperature water as the test fluids such that the viscosity was around 1 cP. For the non-Newtonian experiments, aqueous xanthan gum solutions in 600- and 1,200-ppm concentrations were implemented as the test fluids. These fluids had infinite-shear viscosities around 2 cP, but the zero-shear viscosities ranged from 16 cP to over 100 cP, respectively. For these fluids, the Weissenberg number is used to determine the level of elasticity, while the ratio of zero-to-infinite shear viscosities is used to determine the level of non-Newtonian character. Both properties are interrelated through the solution concentration. Despite the range of operating conditions of impeller speeds from 10–60 rpm and flow rates from 0–230 lpm, resulting in Reynolds numbers of 45,000–268,000, conformity to similarity laws were observed. This was true even at the highest Weissenberg
numbers and concentrations. The plots of the dimensionless pump curve showed the same magnitude of scatter as the Newtonian cases (less than 2%). However, the critical Reynolds number at which the data departs from the similarity law increased notably with fluid concentration.

Fourth, the trends in scaling laws for non-Newtonian fluids were established in terms of the change in critical Reynolds number and the departure from Newtonian similarity laws. The critical Reynolds number for water in the LSRT pump is about 71,000, below which the similarity law is not followed. For the non-Newtonian fluids, the critical Reynolds number increases to 170,000 at 600-ppm concentration and 194,000 at 1,200 ppm. The impeller speeds that correspond to these values dictate Weissenberg numbers of 5 and 57, respectively. Furthermore, there is a well-defined trend in departure from the Newtonian similarity law with increasing concentration. As the xanthan gum concentration increases, the established similarity law deviates further from the Newtonian curve. Based on smooth lines fitted through the data, there is a deviation of 0–3% over the range of IVAS operation for a 600-ppm xanthan solution. At 1,200 ppm, the deviation increases to 4–11% over the operating range of the IVAS pump. These performance deficits are approximately the same magnitude as those due to gross geometry variations, and therefore can be considered to be of first order, rather than second order.
In meeting the four stated objectives, the LSRT facility is a proven tool for non-Newtonian fluid testing. Some potential improvements and proposed future studies will now be presented.

6.2 RECOMMENDED FACILITY IMPROVEMENTS

Although the LSRT proved to be an effective and reliable research tool, there are some recommendations for improvement that would make it an exceptional research facility for testing centrifugal blood pump impellers in Newtonian or non-Newtonian fluids. The improvements, which involve two configuration adjustments, a few instrumentation enhancements, and some equipment upgrades, would result in the facility hypothetically modeled in Figure 6.1.

As stated in Chapter 3, the configuration of the LSRT as tested was not the optimum arrangement for a dedicated flow loop facility. Because the LSRT shared the same laboratory space with the OSU 6x18-Inch Rheology Tunnel, some compromises were made to accommodate both facilities. The first major proposed change in configuration would be to orient the LSRT so that the test fluid flows up, rather than down, through the pump housing. This would alleviate a few bothersome problems related to purging the air from the system that are encountered during operation. In the current configuration, the purging of air has to be performed against the main pump flow, which flows downward. Thus, the dynamic pressure of the main flow works against the buoyancy of the
air. This proves sometimes difficult through the screens and honeycomb in the flow conditioner, particularly with the higher-viscosity fluids.

Figure 6.1 Proposed configuration of the LSRT facility

Another benefit of inverting the flow direction is the minimizing of trapped air in the loop. Because the flow travels upward, the reduction in line pressure resulting from pipe and pump losses would correspond with the decreasing hydraulic grade line. In the current configuration, pipe losses accumulate to reduce pressure in the downward direction (generally), which works against the increasing hydraulic pressure. These two opposing effects
sometimes cause air to be trapped at a lower point in the line where there is a relatively high localized pressure loss, such as that experienced in a wake or recirculation zone. A generally upward flow through the LSRT housing would simultaneously align the gradients in pressure loss, hydraulic grade, and buoyancy. Thus, purging the air from the flow loop would be much easier.

The second major proposed configuration change would be to raise the reservoir fluid level above the LSRT housing to create a "full wet" test chamber. Ideally, the reservoir fluid level would be raised to one meter higher than the split line of the LSRT housing such that a positive-pressure condition is established everywhere in the loop at all flow rates. At zero flow, this would create a gauge pressure in the housing of about 9800 Pa (1.4 psig). This constant hydraulic grade would help to simplify the purge process and improve the flow control. The current configuration relies on an orifice plate flow restrictor to generate enough backpressure so that negative pressures do not induce air in the flow loop. The constant backpressure of the elevated reservoir would eliminate the need for orifice plates and thus enable continuous flow rate control with the circulation pump power setting.

A few instrumentation and equipment acquisitions would improve the operation of the LSRT facility. An estimation of the costs involved is listed in Table 6.1. Instrumentation needs are a pressure multiplexer and a new flow meter. The addition of a ScaniValve™ pressure multiplexer would enable up to
24 different housing and line pressure measurements at each run condition. This would be an improvement over the current manual pressure connections. The appropriate ScaniValve™ is a 24-port J7 model with 3.2-mm (0.125-in) tubulations. Quick-disconnect fittings would improve the task of pressure port plumbing. The only other necessary instrumentation is a turbine flow meter that is compatible with the proposed circulation pump. This would have a body fabricated from PVC with 7.6-cm (3.0-in) bolt flanges for connection to the flow loop.

<table>
<thead>
<tr>
<th>Item</th>
<th>Acquisition Cost</th>
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<tbody>
<tr>
<td>ScaniValve Pressure Multiplexer</td>
<td>2,205.00</td>
</tr>
<tr>
<td>Turbine Flow Meter</td>
<td>593.00</td>
</tr>
<tr>
<td>Non-Metallic Circulation Pump</td>
<td>1,331.00</td>
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<tr>
<td>AC Motor Controller</td>
<td>860.00</td>
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<tr>
<td>Storage Tank</td>
<td>772.00</td>
</tr>
<tr>
<td>Tank Mixer</td>
<td>515.00</td>
</tr>
<tr>
<td><strong>TOTAL COST</strong></td>
<td><strong>$6,276.00</strong></td>
</tr>
</tbody>
</table>

Table 6.1 A cost estimate of LSRT facility upgrades

The biggest equipment needs are a new fluid reservoir, a non-metallic circulation pump, and an AC motor controller. The proposed reservoir would be mounted on a 1-m frame and would consist of an 1100-l (300-gal) LDPE storage tank with an air motor-driven tank mixer to aid in the preparation of large batches of polymer solution. Tank dimensions would be a diameter of 1.09 m (43
in) and a height of 1.50 m (59 in). The non-metallic circulation pump would
eliminate the presence of scale in the flow loop. The current pump is the only
metal component in the facility, apart from the stainless steel body of the flow
meter. The proposed circulation pump is made from high-strength plastic and
capable of 1,900-lpm (500-gpm) flow rates and 100-kPa (15-psi) pressures. The
pump would require an appropriate AC motor controller for variable-speed
operation. The power output would be required to be at least 7,500 W (10 hp) to
drive the pump at maximum flow rate.

6.3 RECOMMENDED TEST PROGRAMS

The results of the current study provide some interesting justifications for
further research. In addition to this motivation, the improvements in the LSRT
facility would enable a new series of tests to be performed on the existing
geometries. The most promising tests would involve a comprehensive variation
in xanthan concentration, comprehensive measurement of pressure distributions
throughout the inlet and housing, and concerted flow visualization efforts.

Since only two data points were established for non-Newtonian fluids
(600- and 1200-ppm xanthan gum), a larger range of fluid concentrations is
proposed. This series of tests would focus on the change in critical Reynolds
numbers in the LSRT for one impeller geometry, design #3452. Xanthan fluid
concentrations of 300, 900, 1,500, and 1,800 ppm would supplement the results
already obtained at 600 and 1,200 ppm. Results from this testing would yield a
higher-resolution trend between the critical Reynolds number and fluid concentration that would bracket the behavior of blood. Then a generalized non-Newtonian scaling law could be formed for these types of pumps using dilute xanthan gum solutions.

Improvements in LSRT instrumentation would enable comprehensive measurements of the pressure distribution through the inlet and in the volute. Inlet pressures would be useful in indicating the nature of the Ekman layer and could reveal levels of preswirl. This would involve up to 8 pressure measurement locations between the inflow conditioner and the leading edge of the impeller blades, including the current inlet pressure tap. Four more pressures could be measured using the dye injection ports between 70 and 96% of the impeller radius. The proposed volute measurements would require 8 pressure taps in the volute passage and 4 taps in the discharge passage. The volute taps would be arranged in a circumferential pattern around the volute at intervals of 45° (twice as many as currently used) and would be located at the bottom of the passage so that air would not rise into the pressure tubing. The pressure taps in the discharge passage would be equally spaced between the throat and the discharge plane where the outlet pressure is currently measured. These pressure measurements would be more useful in quantifying the pump's Newtonian performance and establishing congruent scaling laws between the 1X and 9X pumps, rather than establishing non-Newtonian scaling relationships.
Finally, the transparency of the LSRT housing permitted effective dye injection flow visualizations. This property could be exploited to obtain CFD validation data. Some techniques that have already been implemented involve surface tufting and dye injections, but more advanced methods could be used. The geometry is such that a laser light sheet could be used to illuminate the flow in the horizontal plane at the separation of the upper and lower housings, effecting particle image velocimetry (PIV) measurements from above the LSRT. With maximum fluid velocities on the order of 1 m/sec, a low-power laser (e.g., a 5-W argon ion laser) can be pulsed using a simple beam chopper, rather than more exotic pulse techniques. The images would yield velocity vector maps in a blade passage, from 50–100% of the impeller radius, and could be used to validate the CFD techniques employed to generate the velocity maps shown in Chapter 5.
APPENDIX A

SELECTIVE GLOSSARY OF MEDICAL TERMINOLOGY

The following medical terms have been used in this document. They are briefly defined here to clarify the application. Most of these definitions were obtained from the Reston Encyclopedia of Biomedical Engineering Terms (Graf and Whalen, 1977).

<table>
<thead>
<tr>
<th>Term</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Allograft</td>
<td>A surgically implanted non-native tissue or organ.</td>
</tr>
<tr>
<td>Aneurysm</td>
<td>A permanent dilation of an artery, usually with rupture of the internal and middle coats.</td>
</tr>
<tr>
<td>Antigen</td>
<td>Any substance that brings about the formation of antibodies.</td>
</tr>
<tr>
<td>Aorta</td>
<td>The main artery of the body, approximately one inch in diameter at the widest point.</td>
</tr>
<tr>
<td>Arrhythmia</td>
<td>A deviation from the normal rhythm of the heartbeat, including fibrillation and tachycardia.</td>
</tr>
<tr>
<td>Atrium</td>
<td>A chamber of the heart receiving venous blood. There are two atria of the heart.</td>
</tr>
<tr>
<td>Brachiocephalic trunk</td>
<td>The artery that supplies the head and right arm.</td>
</tr>
<tr>
<td>Cardiomyopathy</td>
<td>A disease of the heart muscle that is not a result of a specific infection.</td>
</tr>
<tr>
<td>Term</td>
<td>Description</td>
</tr>
<tr>
<td>------------------</td>
<td>--------------------------------------------------------------------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Coronary arteries</td>
<td>The arteries that supply blood to the myocardium.</td>
</tr>
<tr>
<td>Carotid arteries</td>
<td>The two arteries of the neck that supply blood to the brain.</td>
</tr>
<tr>
<td>Diastole</td>
<td>The period of the cardiac cycle during which the ventricles fill with blood. Diastolic pressure is thus the minimum blood pressure generated by the heart.</td>
</tr>
<tr>
<td>Erythrocyte</td>
<td>A red blood cell.</td>
</tr>
<tr>
<td>Hematocrit</td>
<td>Erythrocyte volume fraction in the blood. Does not necessarily indicate the blood’s ability to carry oxygen.</td>
</tr>
<tr>
<td>Hemoglobin</td>
<td>A heme protein that transports oxygen and carbon dioxide; constitutes 99% of the protein content of erythrocytes.</td>
</tr>
<tr>
<td>Hemolysis</td>
<td>A separation of the hemoglobin from the red blood cell that thus hinders the ability to carry oxygen. It is sometimes caused by trauma to the blood.</td>
</tr>
<tr>
<td>Idiopathic</td>
<td>When an event occurs without an apparent cause.</td>
</tr>
<tr>
<td>Immunosuppression</td>
<td>The act of suppressing the body’s tendency to produce antibodies as a response to antigens.</td>
</tr>
<tr>
<td>Infarction</td>
<td>A localized area of dead tissue that results from a restriction of circulation to the area.</td>
</tr>
<tr>
<td>Inferior vena cava</td>
<td>The major vein returning blood from the head and arms.</td>
</tr>
<tr>
<td>Ischemia</td>
<td>The effect of constricting the flow of oxygenated blood to tissue.</td>
</tr>
<tr>
<td>Leukocyte</td>
<td>A white blood cell.</td>
</tr>
<tr>
<td>Mitral valve</td>
<td>The valve that separates the left atrium from the left ventricle.</td>
</tr>
<tr>
<td>Myocardium</td>
<td>Heart tissue; the middle muscular layer of the heart wall.</td>
</tr>
<tr>
<td>Term</td>
<td>Definition</td>
</tr>
<tr>
<td>----------------------</td>
<td>-----------------------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Oxygenator</td>
<td>A device that mechanically enhances the solubility of oxygen in the blood.</td>
</tr>
<tr>
<td>Papillary muscles</td>
<td>Muscles controlling the mitral and tricuspid valves.</td>
</tr>
<tr>
<td>Pericardium</td>
<td>The membranous tissue sac that surrounds the heart.</td>
</tr>
<tr>
<td>Perioperative</td>
<td>Occurring during a surgical procedure.</td>
</tr>
<tr>
<td>Plasma</td>
<td>The noncellular liquid that suspends erythrocytes and leukocytes.</td>
</tr>
<tr>
<td>Platelet</td>
<td>A megakaryocyte cytoplasm responsible for clotting.</td>
</tr>
<tr>
<td>Pulmonary arteries</td>
<td>The arteries that transport blood to the lungs.</td>
</tr>
<tr>
<td>Septum</td>
<td>A wall of tissue separating the right and left halves of the heart.</td>
</tr>
<tr>
<td>Subclavian artery</td>
<td>The artery that supplies the head and left arm.</td>
</tr>
<tr>
<td>Superior vena cava</td>
<td>The major vein returning blood from the trunk and legs.</td>
</tr>
<tr>
<td>Systole</td>
<td>The time interval of the cardiac cycle during which the ventricles are in contraction. Systolic pressure is thus the maximum blood pressure generated by the heart.</td>
</tr>
<tr>
<td>Thromboembolism</td>
<td>Sudden blocking of an artery as a result of a dislodged blood clot.</td>
</tr>
<tr>
<td>Thrombosis</td>
<td>The formation of a clot in a blood vessel. The clot itself is known as a thrombus.</td>
</tr>
<tr>
<td>Tricuspid valve</td>
<td>The valve that separates the right atrium from the right ventricle.</td>
</tr>
<tr>
<td>Ventricle</td>
<td>A chamber of the heart that ejects blood. There are two ventricles in the heart.</td>
</tr>
</tbody>
</table>
APPENDIX B

BOUNDARY LAYER CALCULATIONS

The numerical computation of 2D non-Newtonian boundary layer development was performed by Bodonyi (1997) and is presented here. This work attempts to answer the following question: What is the effect of non-Newtonian fluid viscosity model on the separation point and the boundary layer thickness? To this end, the governing two-dimensional laminar boundary layer equations were solved numerically.

For boundary layer calculations, it is common to write the equations purely in terms of two localized orthogonal velocity components: in the streamline direction and normal to it. Thus, the impressed external pressure gradient produces the desired result for a body of finite thickness. The continuity equation and the momentum equations reduce considerably for two-dimensional incompressible flow:

\[ \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \]

\[ \rho \left( u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) = -\frac{dp}{dx} + \frac{\partial}{\partial y} \left( \eta \frac{\partial u}{\partial y} \right) \]
In these equations, \( u \) and \( v \) are the components of the velocity in the streamline direction, \( x \), and the normal direction, \( y \), respectively; \( p \) is the surface pressure (assumed to be known), \( \rho \) is the fluid density, and \( \eta \) is the coefficient of dynamic viscosity. For Newtonian analysis, \( \eta \) represents the constant fluid viscosity, while for non-Newtonian analysis, it represents the shear-dependent viscosity function. These equations are supplemented by the boundary conditions shown in Table B.1.

- No-slip condition at the wall: \( u = v = 0 \) at \( y = 0 \) (for all \( x \))
- External velocity profile at edge of boundary layer: \( u \to u_e(x) \) as \( y \to \infty \) (for all \( x \))
- Initial boundary layer velocity profile: \( u = u_0(y) \) at \( x = x_0 \) (for all \( y \))

Table B.1 Computational boundary conditions

It is imperative to note that these equations cannot be solved analytically with a closed-form solution. Thus they have been rewritten using finite-differences (FD). The numerical scheme incorporates second-order accurate central differencing along with the Thomas algorithm. Since the controlling equations are nonlinear, an iterative process is implemented and applied repeatedly until acceptable error criteria are met. A Newtonian linearization technique, where the velocity update term \( U_\delta^i \) is replaced by \( U_j^i + \delta U_j \) and terms involving higher powers of \( \delta U_j \) are neglected, is used to generate the finite difference form of the controlling equations:

\[
x_i = x_0 + (i-1) \cdot \Delta x, \quad i = 1, 2, \ldots, l
\]
\[ y_j = (j-1) \cdot \Delta y, \quad j = 1, 2, \ldots, N \]

\[ R_j \cdot \delta U_{j+1} + S_j \cdot \delta U_j + T_j \cdot \delta U_{j+1} = Z_j, \quad j = 2, 3, \ldots, N - 1 \]

The coefficients \( R_j, S_j, T_j, \) and \( Z_j \) of this characteristic tri-diagonal system are defined by:

\[ R_j = h_j - \frac{1}{2} \Delta y \left[ \left( \frac{\partial h}{\partial y} \right)_j + \alpha F_j + \gamma \frac{\partial F}{\partial x} \right] \]

\[ S_j = -2 \left[ h_j + \beta (\Delta y)^2 U_j + \gamma \frac{(\Delta y)^2}{\Delta x} U_j \right] \]

\[ T_j = 2h_j - R_j \]

\[ Z_j = -h_j \left[ U_{j-1}^{i-1} - 2U_i^{i-1} + U_{j+1}^{i-1} + U_{j-1}^i - 2U_j^i + U_{j+1}^i \right] \]

\[ - \left( T_j - h_j \right) \left[ U_{j+1}^{i-1} + U_{j+1}^{i-1} - U_{j-1}^{i-1} + U_{j-1}^i \right] \]

\[ - 2\beta (\Delta y)^2 \left[ 1 - \frac{1}{2} (U_j^{i-1})^2 \right] - \gamma \frac{(\Delta y)^2}{\Delta x} (U_j^{i-1})^2 \]

\[ + \left[ \beta (\Delta y)^2 + \gamma \frac{(\Delta y)^2}{\Delta x} \right] (V_j^i)^2 \]

In these finite difference equations, \( h_i \) represents the normalized non-Newtonian viscosity law, and \( \Delta x \) and \( \Delta y \), the grid spacing in the \( x \) and \( y \) directions, respectively. The functions \( \alpha, \beta, \) and \( \gamma \) are defined by:

\[ \alpha = \sqrt{x_i} \cdot \frac{d}{dx} \left( \frac{U}{\sqrt{x_i}} \right) \]

\[ \beta = x_i^2 \left( \frac{dU}{dx} \right)_i \]
The function $F$ is defined as:

$$
F(x, y) = \int_0^y U(\xi) \, d\xi
$$

Thus, $F$ is recursively calculated from previous quantities at each $x$ coordinate:

$$
F_j^i = F_{j-1}^i + \frac{1}{2} \Delta y \left[ U_{j-1}^i + U_j^i \right] \quad j = 1, 2, \ldots, N
$$

Clearly, this algebraic system is nonlinear. Thus, an initial guess was selected for the velocity distribution $U$ (and $F$) so that the coefficients $R$, $S$, $T$, and $Z$ could be calculated. The resulting tri-diagonal matrix system was solved for the $\delta U$ values, which were then used to compute new, updated values for $U$. This process was repeated at each increasing $x$ coordinate ($i$-index) until the following terms were checked for convergence:

$$
|\delta U_j^i| < \varepsilon, \quad j = 1, 2, \ldots, N
$$

The convergence criteria used has an error limit of $\varepsilon < 10^{-6}$. 

\[\gamma = x_i U_{e_i}^2\]
APPENDIX C

PROPERTIES OF WATER

For the data reduction of the LSRT test data, the properties of water were calculated based on the measured reservoir fluid temperature. Figure C.1 shows the variation in specific gravity ($g_{sp}$) for a wide range of temperatures based on the ASME steam tables as referenced by Avallone and Baumeister (1987, 6-10).

Figure C.1  Temperature dependence of the specific gravity of water
The smooth fourth-order curvefit through the data points yields the following representation of the specific gravity:

\[ g_{sp} = \sum_{i=0}^{4} A_i T^i \]

The coefficients \( A_i \) that give the best fit are listed in Table C.1.

<table>
<thead>
<tr>
<th>( i )</th>
<th>( A_i )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>9.9987x10^{-01}</td>
</tr>
<tr>
<td>1</td>
<td>6.4457x10^{-05}</td>
</tr>
<tr>
<td>2</td>
<td>-8.5614x10^{-06}</td>
</tr>
<tr>
<td>3</td>
<td>6.8726x10^{-08}</td>
</tr>
<tr>
<td>4</td>
<td>-3.5466x10^{-10}</td>
</tr>
</tbody>
</table>

Table C.1 Curvefit coefficients for the specific gravity of water

This approximation to obtain the fluid density was used for both water and aqueous xanthan gum test fluids. Xanthan gum was added in relatively weak concentrations (600–1200 ppm), even for the higher-viscosity tests. This resulted in, at most, 1.18 kg (2.60 lbs) of xanthan gum added to 984 liters (260 gal). Therefore, the density change was less than 0.1%. With ambient temperatures in the range of 17–22 °C, the density variation was also less than 0.1%.

Variation in water viscosity is much more pronounced at room temperatures, as shown in Figure C.2. For the LSRT tests, viscosity varied approximately 13%.
Figure C.2  Temperature dependence of the viscosity of water

A fifth-order polynomial curvefit was used to approximate water viscosity:

\[ \eta = \sum_{j=0}^{5} B_j T^j \]

The coefficients \( B_j \) that give the best fit are listed in Table C.2.

<table>
<thead>
<tr>
<th>( j )</th>
<th>( B_j )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>1.7529x10^{-00}</td>
</tr>
<tr>
<td>1</td>
<td>-5.4587x10^{-02}</td>
</tr>
<tr>
<td>2</td>
<td>1.0893x10^{-03}</td>
</tr>
<tr>
<td>3</td>
<td>-1.3705x10^{-05}</td>
</tr>
<tr>
<td>4</td>
<td>9.5625x10^{-08}</td>
</tr>
<tr>
<td>5</td>
<td>-2.7656x10^{-10}</td>
</tr>
</tbody>
</table>

Table C.2  Curvefit coefficients for the viscosity of water

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APPENDIX D

IVAS PROTOTYPE TEST DATA

True-scale IVAS prototype pumps were tested at CCF during the same timeframe as the LSRT work. These pump configurations tested are known generically as the CCF model 3100 blood pump. Although this work was not done by the author, the results of the performance mapping are presented here for reference purposes. This data is used to compare pump performance scaling effects.

The tested pump housings and impellers were fabricated from Cibatool™ 5170 using a stereo lithography rapid prototyping technique. IVAS pump housing part #3374 was designated the baseline volute for the tests, and the two impeller designs implemented were #3386 (flat rotor blades) and #3452 (curved rotor blades). The IVAS impeller diameter is 3.170 cm (1.248 in) for both geometries. These geometries are shown in detail in Chapter 3. The one major difference between IVAS and LSRT impellers was that the IVAS centerbody remained stationary, while the LSRT centerbody rotated with the blading.
Data is presented here that demonstrates the performance differences between the two impeller geometries when operated at representative run conditions. A variation of viscosity is also shown to produce no significant deviation in the Newtonian operation of an IVAS pump utilizing impeller #3386.

D.1 IVAS IMPELLER GEOMETRY COMPARISON

True-scale pumps were operated using CCF's IVAS impeller designs #3386 and #3452. Results for IVAS impeller #3386 were obtained using pump build #224A, which was tested on 8 May 1997. The Newtonian test fluid was a solution of 40% glycerin and 60% water having a viscosity of 3.7 cP. The results for IVAS impeller #3452 were obtained using pump build #242, tested on 22 April 1998, based on a fluid viscosity of 3.52 cP. Both impellers were operated at 2,500, 3,000, and 3,500 rpm, while pump output was measured for a test fluid temperature of 37° C.

The dimensional pump curve data is compared in Figure D.1. Impeller design #3386 consistently produces about 10% more output than design #3452. This is evident in the curvefits of the dimensional performance by applying a smooth, sixth-order polynomial. This reveals the general trend and distinguishes between the different datasets.

The dimensionless performance coefficients are shown in Figure D.2, and the 10% deficit is apparent. It was anticipated that impeller #3452 would be an improvement over impeller #3386 due to the 44% reduction in inlet angle. These
results seem to refute that assumption. These conclusions are reiterated by the plots of the flow and pressure coefficients in Figure D.3 and Figure D.4. The flow coefficient is much more agreeable than the pressure coefficient when comparing the two impeller designs. The bigger differences occur at low flow conditions (specific speeds less than 0.5).

Figure D.1  IVAS pump curve data for both impellers
Figure D.2  IVAS performance coefficients for both impellers

Figure D.3  IVAS performance flow coefficient for both impellers
D.2 IVAS IMPELLER #3386 VISCOSITY STUDY

Impeller #3386 was implemented in the IVAS prototype baseline volute to evaluate the effect of varying viscosity on the pump output. These results were obtained using pump build #224B, which was tested on 3 June 1997. The viscosity trade study was accomplished by modifying the composition of the test fluid. Viscosity was varied by changing the ratio of glycerin to water (normally set to 40/60) to yield Newtonian viscosities of 2.50, 3.52, and 4.58 cP.
Figure D.5 depicts the pump curves for the viscosity trade study. These results show no significant trend based on viscosity variation, and the scatter in the data is consistent with the deviation in the experimental measurements.

![Graph showing pump curves for different viscosities and RPMs](image)

**Figure D.5**  Viscosity effect on IVAS impeller #3386 pump curves

As shown in Figure D.6, the range of variation produces no significant deviation from the similarity law for this impeller. This scatter seems worse than that of the dimensional curves, but it is consistent with the precision of the measurements. The curvefit is a smooth sixth-order polynomial for visual trending.
Figure D.6  Viscosity effect on IVAS impeller #3386 coefficients
APPENDIX E

SENSOR CALIBRATION DATA

Calibration relationships for all instrumentation used for experimental measurements in this study are presented here. The sensors were used to measure housing pressures, loop flow rate, and fluid temperature. With the exception of the flow meter, all instrumentation was calibrated on a monthly basis to ensure accuracy and repeatability of the results.

Housing gauge pressure was measured by a Statham PDCR23 transducer connected to the LSRT discharge flange. The calibration, shown in Figure E.1, was performed by connecting it to a 1-liter water reservoir that was varied in height from zero to about 13 feet, creating a calibration curve of a better than 99% linear correlation. The calibration curve as a function of voltage is thus:

\[ P_g = 1.3207 \cdot (V + 0.0483) \]

The calibration curve of the Validyne DP-15 differential pressure transducer is shown in Figure E.2. The positive and negative sides of this transducer were connected to two 0.5-liter reservoirs, in which the water levels were varied. The difference in water levels was in the range ±2.5 inches.
Figure E.1  Calibration of a Statham PDCR23 pressure transducer

Figure E.2  Calibration of a Validyne DP-15 pressure transducer
Thus, the wet-wet pressure differential of the DP-15 can be expressed as a function of measured voltage using the equation:

\[ \Delta P = 0.01008 \cdot (V - 0.0531) \]

A Potter 2C-50318 turbine flow meter with a Hall-effect sensor was used to determine loop flow rate; its calibration is shown in Figure E.3. The calibration was performed by conducting drain tests from a 35-gal HDPE barrel raised to a height of eight feet with a forklift. The same turbine meter leg used in the LSRT flow loop was connected to the barrel in a vertical orientation, with a butterfly valve at the bottom end. Two lines were drawn on the barrel, a “start” line and a “stop” line, to bracket a 29.5-gal volume. A reference line was drawn halfway between the start and stop lines.

Figure E.3 Calibration of a Potter 2C-50318 turbine flowmeter
A series of water tests were conducted at different valve settings. The valve was opened while the drain time was measured with a stopwatch (according to the fluid level reaching the start and stop lines). When the fluid level reached the reference line, the frequency of the turbine was measured with a frequency counter/totalizer. The flow rate results were corrected for a small amount of constant leakage. The turbine meter flow rate and the rate meter’s analog output voltage are represented as:

\[ \text{GPM} = 0.5042f + 0.5089 \]
\[ \text{Voltage} = 0.0504f + 0.0405 \]

Thus, the flow rate as a function of the analog output voltage is:

\[ \text{GPM} = 9.9957 \times (V + 0.0104) \]

As part of the flow metering of the LSRT, orifice plate flow restrictors were implemented to control rate and backpressure. To ensure the accuracy of the turbine flow meter, the housing gauge pressure had to be in the range 1.4–5.2 psig (72–269 mmHg). This was required so that air would not form in the flow meter leg.

The orifices had sharp upstream edges with a 45° chamfer on the downstream side. Plates were made from 0.25-in thick aluminum with orifice diameters of 0.383” (9.7 mm), 0.550” (14.0 mm), 0.756” (19.2 mm), 1.019” (25.9 mm), and 1.250” (31.8 mm). They were installed between two four-inch PVC pipe flanges. Resulting orifice calibration loop flow rates and housing gauge pressures are shown in Figure E.4 and Figure E.5.
Figure E.4  Calibration flow rate of orifice plate flow restrictors

Figure E.5  Calibration pressure of orifice plate flow restrictors
Reservoir fluid temperature was monitored to determine water specific gravity and viscosity. This was done with a K-type thermocouple and instrumentation amplifier. Fluid temperature was calibrated by measuring the amplified signal (gain=2,000) of an ice point and a boiling point. Calibration results are shown in Table E.1. The boiling point is less than 212° F in Columbus, based on the altitude of 1,000 feet.

<table>
<thead>
<tr>
<th>Voltage</th>
<th>Degrees (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.2539</td>
<td>32</td>
</tr>
<tr>
<td>8.1201</td>
<td>200</td>
</tr>
</tbody>
</table>

Table E.1  K-type thermocouple calibration points

Thus, the linear equation for the thermocouple calibration is given as:

\[ T = 21.3572 \cdot (V + 1.2444) \]
APPENDIX F

LSRT IMPELLER #3386 DATA

This section contains the data for LSRT tests on impeller design #3386.

Two fluids were used as the test medium: water and 600-ppm aqueous xanthan gum. The endwall clearance, the gap between the rotating blades and the upper housing designated by \( c_{ewr} \), was set to 0.9 and 4.0 mm for each fluid. Taking into consideration the impeller speed and infinite-shear viscosity, the maximum Reynolds number achieved was 436,000 for water and 282,000 for xanthan gum.

Each data point represents a measurement of impeller rpm, housing gauge pressure, pump pressure rise, loop flow rate, and fluid temperature. At a clearance of 0.9 mm for both fluids, the circumferential distribution of volute pressure was also measured. Thus, the following charts are presented for each run condition:

- Dimensional Pump Curve (Pump Head vs. Flow Rate)
- Dimensionless Performance (Pressure Coefficient vs. Flow Coefficient)
- Flow Coefficient vs. Specific Speed
- Pressure Coefficient vs. Specific Speed
- Pump Operating Curves
- Volute Pressures (selected conditions)
- Volute Pressure Coefficients (selected conditions)
F.1 WATER TESTS

These tests were conducted using water as the test fluid. Pressure rise was measured for impeller speeds of 0–50 rpm and flow rates of 0–58 gpm.

F.1.1 Endwall Clearance of 0.9 mm

This data represents LSRT runs 135–158 acquired on 9 February 1999.

Figure F.1 Impeller #3386 pump curves: water, $c_w=0.9$ mm
Figure F.2  Impeller #3386 coefficients: water, $c_{ew}=0.9$ mm

Figure F.3  Impeller #3386 flow coefficient: water, $c_{ew}=0.9$ mm
Figure F.4  Impeller #3386 pressure coefficient: water, $c_w=0.9$ mm

Figure F.5  Impeller #3386 operating curves: water, $c_w=0.9$ mm
Figure F.6  Impeller #3386 volute pressures: water, $c_{ow}=0.9$ mm

Figure F.7  Impeller #3386 volute coefficients: water, $c_{ow}=0.9$ mm
**F.1.2 Endwall Clearance of 4.0 mm**

This data represents LSRT runs 159–182 acquired on 13–15 February 1999.

![Graph](image)

**Figure F.8** Impeller #3386 pump curves: water, $c_{ew}=4.0$ mm

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Figure F.9 Impeller #3386 coefficients: water, $c_w=4.0$ mm

Figure F.10 Impeller #3386 flow coefficient: water, $c_w=4.0$ mm
Figure F.11  Impeller #3386 pressure coefficient: water, $c_w=4.0$ mm

Figure F.12  Impeller #3386 operating curves: water, $c_w=4.0$ mm
F.2 600-PPM XANTHAN GUM TESTS

These tests were conducted using xanthan gum as the test fluid. Pressure rise was measured for impeller speeds of 0–60 rpm and flow rates of 0–70 gpm.

F.2.1 Endwall Clearance of 0.9 mm

This data represents LSRT runs 206–239 acquired on 9–12 March 1999.

Figure F.13 Impeller #3386 pump curves: xanthan, $c_{ew}=0.9$ mm
Figure F.14  Impeller #3386 coefficients: xanthan, $c_w=0.9$ mm

Figure F.15  Impeller #3386 flow coefficient: xanthan, $c_w=0.9$ mm
Figure F.16  Impeller #3386 pressure coefficient: xanthan, $c_{ew}=0.9$ mm

Figure F.17  Impeller #3386 operating curves: xanthan, $c_{ew}=0.9$ mm
Figure F.18 Impeller #3386 volute pressures: xanthan, $c_{ew}=0.9$ mm

Figure F.19 Impeller #3386 volute coefficients: xanthan, $c_{ew}=0.9$ mm
F.2.2 Endwall Clearance of 4.0 mm

This data represents LSRT runs 241–269 acquired on 13 March 1999.

Figure F.20 Impeller #3386 pump curves: xanthan, \( c_{ew} = 4.0 \) mm
Figure F.21  Impeller #3386 coefficients: xanthan, $c_w=4.0$ mm

Figure F.22  Impeller #3386 flow coefficient: xanthan, $c_w=4.0$ mm

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Figure F.23  Impeller #3386 pressure coefficient: xanthan, \( c_w=4.0 \) mm

Figure F.24  Impeller #3386 operating curves: xanthan, \( c_w=4.0 \) mm
This section contains the data for LSRT tests on impeller design #3452. Three xanthan gum solutions in concentrations of zero (water), 600, and 1200 ppm were used as the test fluid. The endwall clearance, the gap between the rotating blades and the upper housing designated by \( c_{ew} \), was set to a constant 3.1 mm for all fluids. Taking into consideration the impeller speed and infinite-shear viscosity, the maximum Reynolds number achieved was 428,000 for water, 268,000 for 600-ppm xanthan, and 224,000 for 1200-ppm xanthan gum.

Each data point represents a measurement of impeller rpm, housing gauge pressure, pump pressure rise, loop flow rate, and fluid temperature. For water, the circumferential distribution of volute pressure was also measured.

Thus, the following charts are presented for each run condition:

- Dimensional Pump Curve (Pump Head vs. Flow Rate)
- Dimensionless Performance (Pressure Coefficient vs. Flow Coefficient)
- Flow Coefficient vs. Specific Speed
- Pressure Coefficient vs. Specific Speed
- Pump Operating Curves
- Volute Pressures (selected conditions)
- Volute Pressure Coefficients (selected conditions)
G.1 WATER TESTS

These tests were conducted using water as the test fluid. Pressure rise was measured for impeller speeds of 0–50 rpm and flow rates of 0–56 gpm. This data represents LSRT runs 270–293 acquired on 30 March 1999.

Figure G.1  Impeller #3452 pump curves: water
Figure G.2  Impeller #3452 coefficients: water

Figure G.3  Impeller #3452 flow coefficient: water
Figure G.4  Impeller #3452 pressure coefficient: water

Figure G.5  Impeller #3452 operating curves: water
Figure G.6  Impeller #3452 volute pressures: water

Figure G.7  Impeller #3452 volute coefficients: water
G.2 600-PPM XANTHAN GUM TESTS

These tests were conducted using xanthan gum as the test fluid. Pressure rise was measured for impeller speeds of 0–60 rpm and flow rates of 0–62 gpm. This data represents LSRT runs 360–389 acquired on 23 April 1999.

![Graph of pressure rise vs flow rate for different impeller speeds](image)

Figure G.8 Impeller #3452 pump curves: 600-ppm xanthan
Figure G.9  Impeller #3452 coefficients: 600-ppm xanthan

Figure G.10  Impeller #3452 flow coefficient: 600-ppm xanthan
Figure G.11  Impeller #3452 pressure coefficient: 600-ppm xanthan

Figure G.12  Impeller #3452 operating curves: 600-ppm xanthan
G.3 1200-PPM XANTHAN GUM TESTS

These tests were conducted using xanthan gum as the test fluid. Pressure rise was measured for impeller speeds of 0–60 rpm and flow rates of 0–61 gpm. This data represents LSRT runs 390–433 acquired on 26–28 April 1999.

Figure G.13 Impeller #3452 pump curves: 1200-ppm xanthan
Figure G.14  Impeller #3452 coefficients: 1200-ppm xanthan

Figure G.15  Impeller #3452 flow coefficient: 1200-ppm xanthan
Figure G.16  Impeller #3452 pressure coefficient: 1200-ppm xanthan

Figure G.17  Impeller #3452 operating curves: 1200-ppm xanthan
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