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I am submitting herewith a thesis written by David Alan Hooper entitled "Cavitation of Mercury in a Centrifugal Pump." I have examined the final electronic copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Nuclear Engineering.

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Cavitation of Mercury in a Centrifugal Pump

Literature Review and Study

A THESIS
PRESENTED FOR THE
MASTER OF SCIENCE
DEGREE
THE UNIVERSITY OF TENNESSEE, KNOXVILLE

David A. Hooper
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ABSTRACT

Cavitation is a significant concern for the reliable operation of a centrifugal pump. Liquid metal flow loops are used in nuclear, chemical, metal forming, and liquid metal dynamo applications. Understanding of the cavitation characteristics of liquid metals is increasingly important to the design and operation of these facilities. One recent field of cavitation research has developed for mercury flow in spallation targets used in neutron sources. To further the understanding of mercury cavitation, a review of the existing literature on water cavitation, liquid metal cavitation, and mercury cavitation is performed. The mechanics of cavitation and the analytical methods applied to cavitation problems are discussed and analyzed. Acoustic data from the centrifugal pump for the mercury flow loop at the Spallation Neutron Source in Oak Ridge National Laboratory are examined.

TABLE OF CONTENTS

BACKGROUND	1
LIFE CYCLE OF CAVITATION BUBBLES	5
BUBBLE FORMATION	6
<i>Angle of Attack.....</i>	8
<i>Housing Shear.....</i>	12
<i>Fluid Acceleration</i>	12
<i>Thermal Energy Addition.....</i>	13
<i>Dissolved Gas</i>	13
<i>Bubble Formation Summary</i>	14
BUBBLE EVOLUTION	14
<i>Bubble Flow Patterns</i>	18
<i>Force Balance on Cavitation Bubble.....</i>	19
<i>Material Damage</i>	22
BUBBLE COLLAPSE	23
FORMS OF CAVITATION DAMAGE	28
THERMODYNAMIC CAVITATION SURGING	28
HYDRODYNAMIC CAVITATION SURGING	30
HYDRAULIC PERFORMANCE LOSS.....	33
CAVITATION EROSION	34
<i>Bubble Flow Characteristics</i>	36
MATHEMATICAL ANALYSIS.....	40
CLASSICAL APPROACH	40
NONDIMENSIONAL ANALYSIS AND SCALING FACTORS	41
CONCLUDING REMARKS ON CAVITATION THEORY	42
DIMENSIONLESS ANALYSIS PARAMETERS FOR SCALING MERCURY AND WATER	
CAVITATION INCEPTION.....	43
<i>Dimensionless Analysis in Fluid-Structure Interactions</i>	44
<i>Other Nondimensional Numbers and Performance Measures</i>	46
SMITHSONIAN PHYSICAL TABLES	49
FURTHER NONDIMENSIONAL ANALYSIS	51
COMPARISON OF MERCURY AND WATER PROPERTIES.....	56
SNS PUMP NOISE EVALUATION.....	68
FREQUENCY ANALYSIS	71
<i>Frequency Noise Comparison of 150 RPM and 400 RPM.....</i>	75
SIGNIFICANCE OF THE LOCATION OF DETECTORS	77
<i>Effect of the Orientation of the Accelerometers.....</i>	81
POSSIBLE REMEDIES FOR THE OBSERVED VIBRATIONS.....	86

CONCLUDING REMARKS	87
BIBLIOGRAPHY	88
APPENDIX APPENDIX A : LITERATURE REVIEW OF TURBULENT PRANDTL NUMBERS.....	92
APPENDIX A : LITERATURE REVIEW OF TURBULENT PRANDTL NUMBERS	93
VITA.....	101

LIST OF FIGURES

FIGURE 1 - BASIC CENTRIFUGAL SUMP PUMP DESIGN	2
FIGURE 2- NUCLEATION SITE OF A VAPOR BUBBLE	7
FIGURE 3 - BASIC IMPELLER SCHEMATIC AND FLUID FLOWPATH	10
FIGURE 4 -BLADE ANGLE OF ATTACK	11
FIGURE 5 - GENERAL SHAPE OF CAVITATION BUBBLE GROWTH AND COLLAPSE ..	17
FIGURE 6 - FORCES ACTING ON A CAVITATION BUBBLE IN A PUMP IMPELLER	20
FIGURE 7 - BUBBLE FLOWPATHS FROM BIRTH TO DEATH	24
FIGURE 8 - BUBBLE COLLAPSE AND LIQUID MICROJET	27
FIGURE 9 - VARIOUS CAVITATION PRESSURE TRANSIENTS	32
FIGURE 10 - PATTERN OF CAVITATION ASSOCIATED WITH CAVITATION EROSION	37
FIGURE 11 - CAVITATION PATTERN DURING HYDRODYNAMIC SURGING	38
FIGURE 12 - CAVITATION PATTERN FOR THERMODYNAMIC SURGING	39
FIGURE 13 - GRINDELL DATA FOR WATER-NAK CAVITATION CORRELATION	50
FIGURE 14 - VAPOR BUBBLES AGAINST A SURFACE IN WATER AND MERCURY	54
FIGURE 15 - DENSITY (WATER AT 100 BAR, MERCURY AT 1 BAR)	59
FIGURE 16 – THERMAL CONDUCTIVITY (WATER AT 100 BAR, MERCURY AT 1 BAR)	60
FIGURE 17 - SPEED OF SOUND (WATER AT 100 BAR, MERCURY AT 1 BAR)	61
FIGURE 18 - SPECIFIC HEAT CAPACITY (WATER AT 100 BAR, MERCURY AT 1 BAR)	62
FIGURE 19 - DYNAMIC VISCOSITY (WATER AT 100 BAR, MERCURY AT 1 BAR)	64
FIGURE 20 - KINEMATIC VISCOSITY (WATER AT 100 BAR, MERCURY AT 1 BAR) ..	65

FIGURE 21 - THERMAL DIFFUSIVITY (WATER AT 100 BAR, MERCURY AT 1 BAR) .	66
FIGURE 22 - PRANDTL NUMBER (WATER AT 100 BAR, MERCURY AT 1 BAR)	67
FIGURE 23 - FREQUENCY SPECTRUM OF THE SNS PUMP (ROTHROCK, 2006)	69
FIGURE 24 - 150 RPM DETAIL OF SNS PUMP (ROTHROCK, 2006).....	70
FIGURE 25 - FREQUENCY SPECTRUM OF THE SNS PUMP – OUTBOARD ACCELEROMETER (ROTHROCK, 2006)	78
FIGURE 26 - 150 RPM DETAIL – OUTBOARD ACCELEROMETER (ROTHROCK, 2006)	79
FIGURE 27 - INBOARD VERTICAL ACCELEROMETER (ROTHROCK, 2006)	82
FIGURE 28 - 150 RPM DETAIL INBOARD VERTICAL ACCELEROMETER (ROTHROCK, 2006)	83
FIGURE 29 - OUTBOARD VERTICAL ACCELEROMETER (ROTHROCK, 2006).....	84
FIGURE 30 - RPM DETAIL – OUTBOARD VERTICAL ACCELEROMETER (ROTHROCK, 2006)	85

BACKGROUND

The Spallation Neutron Source (SNS) at Oak Ridge National Laboratory is currently the most powerful spallation source of neutrons in the world. The facility operates by firing an intense pulse of high-energy protons into a liquid mercury target. As the neutrons collide with mercury nuclei, neutrons are dislodged from the mercury nuclei – a process termed spalling. With a targeted peak power level of 2 to 4 MW, the SNS target facility poses a significant engineering challenge in removal of thermalized beam energy and attenuation of shock waves produced by the proton bombardment of the mercury. The mercury is circulated in a flow loop by a centrifugal sump pump at volumetric flow rates up to approximately 380 gallons per minute to facilitate removal of the thermal energy, which is 97% of the beam energy.

Error! Reference source not found. shows the basic sump type centrifugal pump employed in the SNS target flow loop. The pump impeller is open, as shown in the picture included with **Error! Reference source not found.**, and is cantilevered on a shaft nearly one meter long. The mercury flows into the open impeller flow below, and is thrust horizontally through the pump outlet into the flow loop piping. The 25-kW drive motor is located vertically above the impeller, and directly drives the impeller by means of the cantilevered shaft. The shaft position is controlled by a radial thrust bearing located at the top of the sump well housing, and an axial thrust bearing within the motor housing.

Relatively little data exists regarding mercury flow in centrifugal pumps, particularly in comparison to water and other common fluids. Noise analysis of the SNS pump has indicated possibility of cavitation for flow rates above 200 gallons per minute. Cavitation can cause a wide range of harmful effects within a pump, from reduced flow efficiency or an increased wear rate of pump

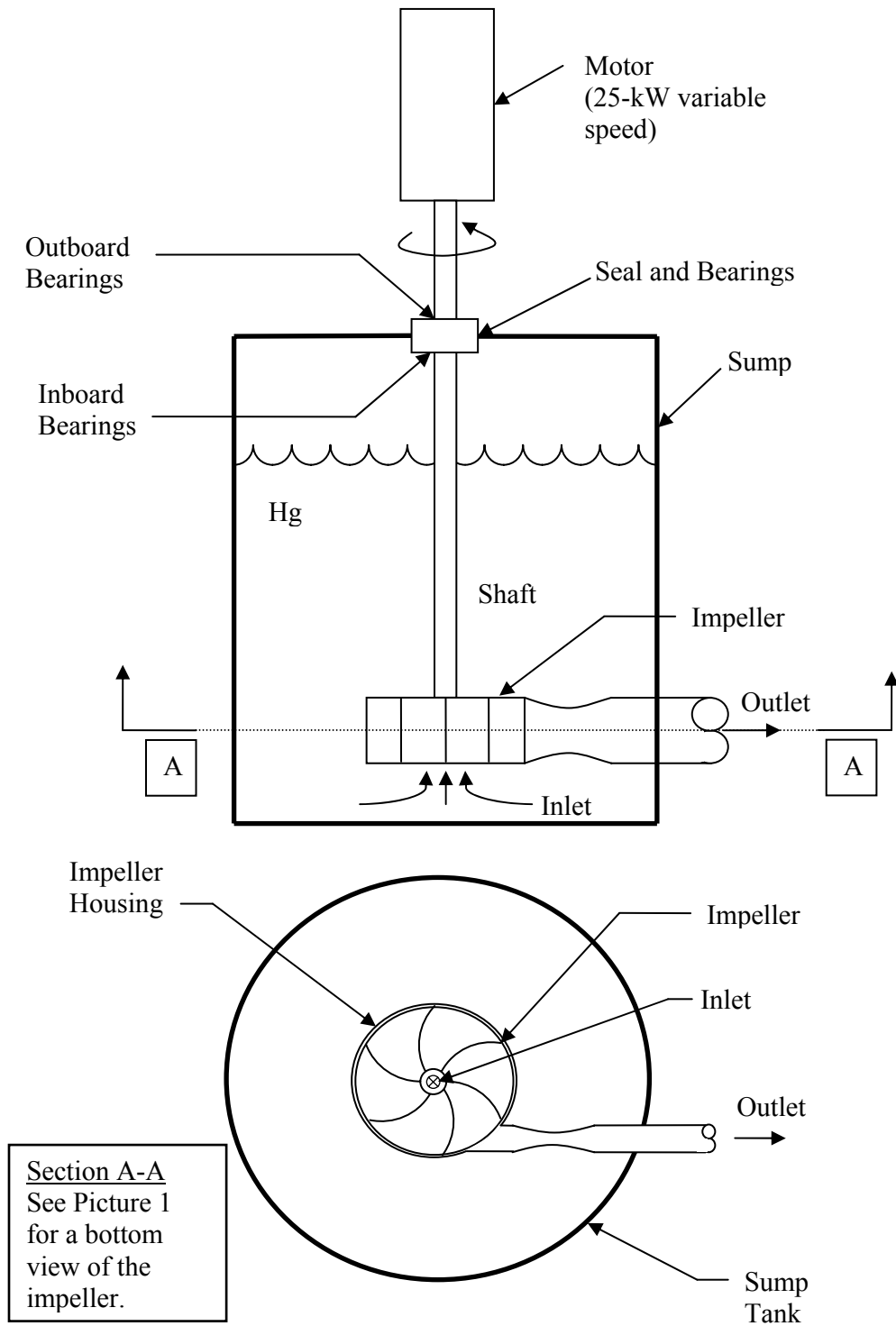


Figure 1 - Basic Centrifugal Sump Pump Design

Figure 1, cont.



Picture 1 - Bottom View of SNS Impeller

components to the outright destruction of the pump or breaching the flow loop. If cavitation is present, the entire flow loop may need to be shut down more frequently than desired in order to repair components that have suffered erosion, or to replace pump components. With such a wide range of possible consequences of cavitation, it is desirable to understand the mechanics of cavitation and the detection of cavitation, as well as methods to avoid cavitation or to minimize its harmful effects.

In order to better understand the possibility of cavitation in the SNS pump, an extensive review of the existing literature on cavitation has been performed. The mechanics of cavitation are discussed, and several models of cavitation are compared. The mechanisms of cavitation damage on flow components are evaluated. Since most existing cavitation data exists for water flow, research was performed to compare mercury flow to water flow in order to use water flow data in the evaluation of the mercury flow of the SNS pump. Finally, a vibration analysis of the SNS pump is examined to determine if cavitation is a significant concern, and to predict the consequences of any cavitation that may exist in the pump during operation.

LIFE CYCLE OF CAVITATION BUBBLES

The phenomenon of cavitation can be described in three phases: bubble formation, bubble evolution, and bubble collapse. Bubble formation describes the physical creation of vapor bubbles within a liquid, and the conditions under which bubble creation might occur. Bubble evolution describes the growth and behavior of cavitation bubbles. Bubble collapse describes the destruction of the cavitation bubble and subsequent return of energy into the fluid. Each phase in the life of a bubble exhibits unique properties, and plays a key role in the overall effects of cavitation on fluid flow and on machinery wear. Here, each phase is assessed individually to better understand its unique properties and how those properties can be seen in the results of cavitation.

In pump applications, the static pressure in a region of liquid is often expressed in terms of *head*. If a point in a fluid at some depth “h” in the fluid is considered, the gauge pressure at that point in the fluid is given by Equation 1:

$$P = \rho gh$$

Equation 1- Static Pressure in a Column of Fluid

Here, ρ is the fluid density and g is acceleration due to gravity. In incompressible fluids, ρ and g can both be treated as constants, and the gauge pressure P is directly proportional to the height h of fluid above the point of interest. The equation may also be rewritten by solving for h :

$$h = \frac{P}{\rho g}$$

Equation 2 - Height of Liquid as a Function of Pressure

In this form, the liquid height is referred to as “head”. By scaling the liquid pressure with two constants, the pressure is effectively given in units of length. The concept is similar to barometric pressure measurements, where atmospheric pressure is measured by the height of fluid that can be supported in a column by the atmospheric pressure.

Bubble Formation

In general, the formation of a vapor bubble within a liquid requires a nucleation site. An impurity within the liquid or a defect on a surface in contact with the liquid can provide such a location. Gas molecules dissolved within a liquid may also serve as a nucleation site in a phenomenon known as gaseous cavitation. Gaseous cavitation is a concern in centrifugal pump systems with a high concentration of dissolved gas. In a centrifugal pump, the nucleation sites for cavitation bubbles are typically found on the impeller blades or the impeller housing (Grist, 1999). In Figure 2, a nucleation site in water is shown. The liquid cannot completely penetrate the microscopic surface defect due to the surface tension of the liquid. The volume of the defect is filled with vapor at slightly elevated pressure. The size of the vapor bubble can grow or shrink as the fluid pressure varies, but the vapor bubble will remain. If the liquid pressure (head) in the region of the surface defect falls below the vapor pressure of the fluid and overcomes the pressure defect between the liquid and gas, the vapor bubble will expand until either the fluid pressure increases above vapor pressure, or a force (such as fluid friction or buoyancy) detaches a portion of the bubble from the surface. The nucleation sites are a product of the manufacturing process of materials, and are always present in industrial applications. With nucleation sites present in the material, the risk of bubble formation in the liquid on wetted walls will always be present for regions where the liquid pressure falls below vapor pressure.

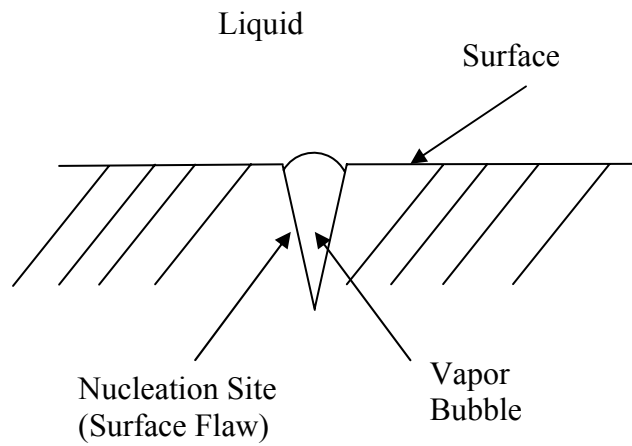


Figure 2- Nucleation Site of a Vapor Bubble

In a centrifugal pump, the most common location for the onset of cavitation is at the impeller inlet (Grist, 1999). Here, the fluid accelerates due to the motion of the impeller, and the system pressure drops to what is often the lowest pressure in the entire flow loop. For this reason, the head at the impeller inlet is the standard criteria for noncavitating pump operation; if the head at the inlet is too low, cavitation will occur. This head is termed Net Positive Suction Head (NPSH).

Net Positive Suction Head is the sum (in head equivalents) of the static pressure and kinetic energy of the fluid minus its vapor pressure at the pump inlet. In Equation 3, the inlet pressure is assumed to be gauge pressure, and is corrected by the addition of atmospheric pressure.

$$NPSH = \frac{p_l + p_{atm}}{\rho \cdot g} + \frac{v_1^2}{2g} - \frac{p_v}{\rho \cdot g}$$

Equation 3 - Net Positive Suction Head

As long as NPSH is positive at all points, bubble formation will never occur; the pressure is sufficient to maintain the liquid phase of the fluid. Unfortunately, many factors can affect the fluid pressure in localized regions within the pump, such that localized regions within the pump may experience a much lower head than the NPSH measured at the inlet.

Angle of Attack

The blade tip of the impeller is the main location for local head loss. Here, as the fluid meets the impeller blade, vortices may form and low-pressure pockets may develop because the fluid has to flow around the impeller blades. All impeller blades will cause a dip in pressure in the fluid, but the blade's angle of attack (θ) can play a very significant role in the magnitude of pressure loss. If

the blade's angle of attack and rotational speed do not match the relative velocity of the fluid, a low pressure region will occur as the fluid accelerates to match the impeller blade. A basic impeller schematic is given in Figure 3; a diagram of the angle of attack is given in **Error! Reference source not found.**

If the angle of attack (θ) is too great, the blade will push the fluid on the leading edge, causing a rise in fluid pressure in the leading region. In this case, however, the fluid entering the trailing region of the impeller blade must accelerate to fill the volume behind the impeller blade. The fluid acceleration causes a reduction in static pressure and may cause trailing-edge loss of NPSH. This is the most common location for bubble formation.

Conversely, if the angle of attack is too shallow, the fluid filling the trailing region of the impeller will be forced to slow down, causing a rise in trailing-edge pressure. The fluid at the leading edge will now experience a temporary drop in pressure as it turns past the impeller blade, and can develop cavitation bubbles if the problem is significant enough.

The angle of attack is a design parameter that is determined by the pump manufacturer, and is a straightforward geometric exercise. For most pumps operating near design conditions, the angle of attack matches the incoming flow vector. If a check of the angle needs to be performed, Equation 4 is useful:

$$\dot{V}_{ideal} = \frac{\omega D_i A_{inlet}}{\tan(\theta)}$$

Equation 4 - Angle of Attack

In this form, the equation is solved for the volumetric flowrate, " $\dot{V}$ ", of the fluid. " ω " is the angular speed of the pump, " D_i " is the inlet diameter, " A_{inlet} " is the impeller inlet area at the blade tips, and " θ " is the angle of attack.

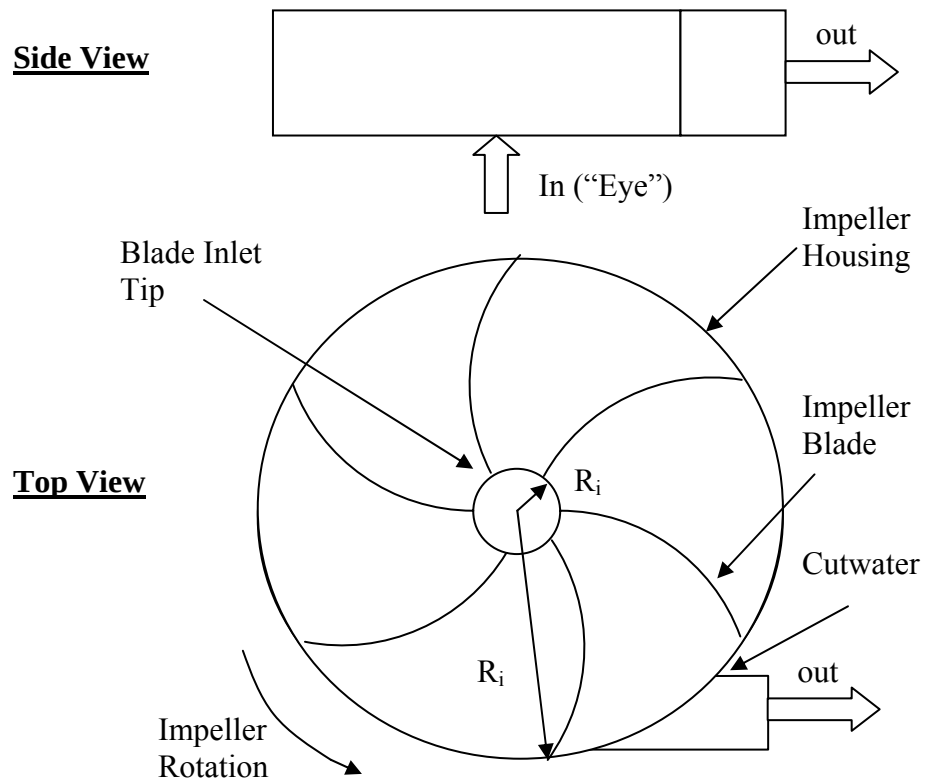


Figure 3 - Basic Impeller Schematic and Fluid Flowpath

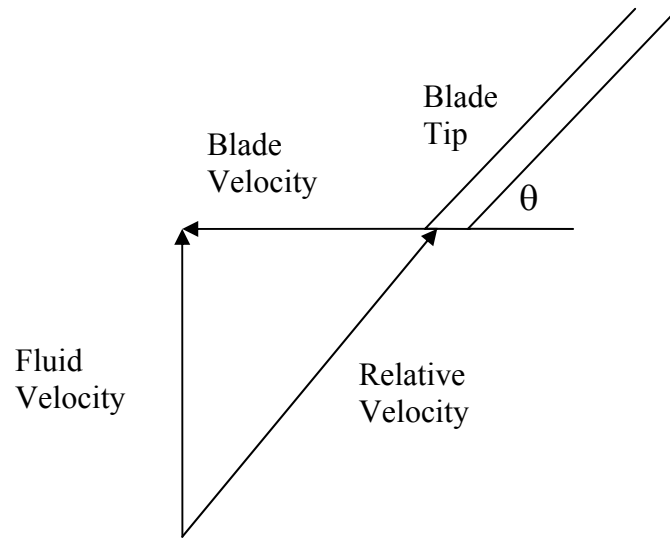


Figure 4 -Blade Angle of Attack

Housing Shear

In the open impeller design, the impeller blade must pass close to the stationary housing (Grist, 1999). A small amount of the pumped fluid will be drawn through the space between the blades and the housing, and will experience a very large shear stress. If the shear stress is sufficient to cause cavitation, the blade (and possibly the housing) will experience erosion. This erosion will eventually wear enough material away to open the clearance and reduce the shear stress. However, the pump performance will degrade as a result of fluid leaking from the high pressure side of the blade to the low pressure side of the blade.

Fluid Acceleration

The premise of a centrifugal pump is to do work on the fluid, leading generally to increased velocities and kinetic energy. The kinetic energy is converted into static pressure by decelerating the fluid in an expanding volute, which is very much like a diverging nozzle. As the fluid moves through the pump volute, the pump blades may add kinetic and potential energy to the fluid. Usually, the fluid will initially experience a drop in static pressure as it accelerates. The more strongly the fluid is accelerated, the greater this pressure drop will be. For very high-speed pumps, the pressure drop can be very large and cause the local pressure to drop to zero. For this reason, pump manufacturers tend to prefer larger, slower-turning impeller blades to smaller, faster blades for high-volume flowrates. Though the larger impeller is bulkier, the magnitude of the flow kinetic energy will be lower, and the pressure drop across the blades will be smaller.

Thermal Energy Addition

A relatively infrequent cause of loss of NPSH is the addition of thermal energy to the fluid. As the fluid's temperature increases, the vapor pressure of the fluid increases. As vapor pressure increases, the NPSH necessarily decreases. The thermal energy can be introduced in a variety of ways; since all real mechanical processes are not thermally ideal, the fluid will experience a gain in thermal energy, though usually slight if the fluid is incompressible. A small gain in thermal energy may significantly increase the risk of cavitation for fluids with low specific heats or in low pressure systems. In these systems, little energy is required to produce a large volume of vapor from a small volume of liquid. For example, condensate booster pumps on feed water in power plants typically operate with inlet suction below 1 atmosphere of pressure. Under this circumstance, a modest amount of thermal energy is necessary to convert a small portion of the liquid water to vapor. The vapor is of very low density, and may occupy a large part of the pump volume.

For most forms of cavitation, thermal energy addition does not play a major role in loss of NPSH. However, as will be seen later, thermal energy addition can become the critical factor in the most dangerous form of cavitation exhibited by centrifugal pumps: thermodynamic cavitation surging. Because of the devastating effects associated with this cavitation, the concept of thermal energy contributing to cavitation is considered.

Dissolved Gas

For many flow loops, some gases may be dissolved in the liquid. For example, the water in a public water fountain recirculation loop will have dissolved nitrogen, oxygen, and argon gases from the exposure of the water to the

atmosphere. If dissolved gases are present during the formation of a cavitation bubble, some of the gas will contribute to the bubble formation. The presence of dissolved gases can have three effects on flow cavitation: increased susceptibility to bubble inception, increased growth rate of the bubble, and incomplete collapse of the bubble.

The inclusion of dissolved gases raises the vapor pressure of the liquid, thereby reducing the NPSH of the fluid (Equation 3). With the reduced NPSH, less reduction in static pressure is necessary to cause cavitation, and the flow will be more susceptible to cavitation inception.

Finally, if dissolved gases are present, the bubble may not completely collapse. If the bubble has traveled from the region of formation to a region where dissolved gases are near saturation in the liquid, the gases in the cavitation bubble will not readily dissolve back into the liquid. This resistance to dissolving may result in a partial collapse of the bubble rather than a complete collapse of the bubble.

Bubble Formation Summary

Cavitation starts with bubble formation, and cavitation is prevented by preventing bubble formation. Preventing cavitation is focused on the mechanisms behind bubble formation. Cavitation can be designed completely out of most pumping systems if the general factors that can drive NPSH down are known.

Bubble Evolution

Once the bubbles form, a regular pattern of existence is usually followed (**Error! Reference source not found.**). Sometimes the bubble will remain

trapped at the point of inception; this is more commonly seen on the suction side of the impeller blade. However, the most common event is for the bubble to travel along the flowpath. The bubble may travel a short distance or a relatively long distance (e.g. a foot or more), depending on its growth rate and the speed of the fluid flow.

The two most important components of the evolution of a cavitation bubble are the growth rate and the location of the bubble. The ultimate size of the bubble is related to the growth rate of the bubble, which is related to the conditions of bubble inception (Tillner, 1993). For example, if a region exists where the NPSH drops just low enough to allow bubble formation, the impetus for bubble development is relatively weak. Growth rate will be limited, and the bubble will not become very large before moving into a higher pressure region. However, if the NPSH is significantly below zero, the conditions favor rapid bubble growth and the bubble may become large.

In **Error! Reference source not found.**, the bubble growth and collapse rate are seen as functions of time. In this case, the bubble formation was produced by acoustic cavitation (Shah, 1999). Though the motivation is different, the actual process of bubble creation and growth is the same. The bubble is created by the local reduction of static pressure below vapor pressure. The growth period of the cavitation bubble is noticeably longer than the time required for collapse, a phenomenon that contributes greatly to the damage potential of the bubbles.

The movement of the bubble is a function of the flow characteristics around the bubble. Sometimes, if a bubble is formed on the suction side of the impeller, the liquid flow near the bubble develops vorticity as the impeller passes by; this vortex can keep the bubble trapped in its original location. In fact, a

steady-state condition can be achieved where a vapor pocket trails the impeller blade. If a steady-state vapor pocket is created, the fluid velocity profile in the

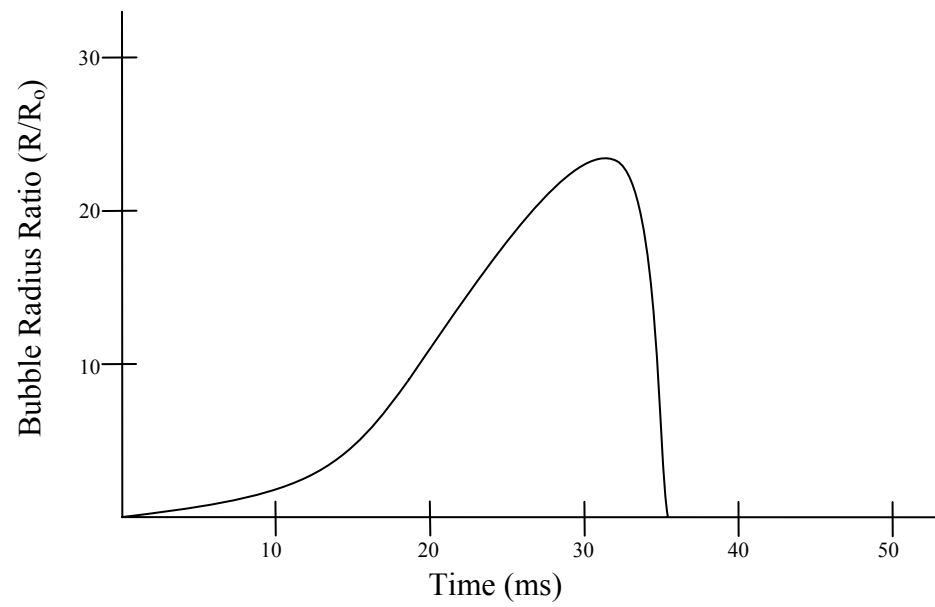


Figure 5 - General Shape of Cavitation Bubble Growth and Collapse

impeller will be affected; large vapor pockets may substantially reduce the fluid flow and the effectiveness of the pump.

More commonly, the bubbles move away from the point of inception and become a part of the fluid flow. Since fluid flow in pumps is typically highly turbulent, the actual flowpath can be rather chaotic; discussion of the general flow is therefore easier than the exact flow. In general, the bubble may travel in three different patterns.

Bubble Flow Patterns

Once a bubble is formed, it will usually travel within the fluid according to the path of least resistance. The most common flow pattern is for the bubble to move through the impeller along the intended flowpath of the fluid. If the bubble does travel within the fluid, it will eventually leave the localized region of low pressure that caused the bubble to form; once the bubble enters a region of sufficient head, the bubble will collapse.

If relatively few bubbles are generated, their contribution to the flow may be of minor importance – little to no reduction in generated head or fluid flowrate may be detected. However, if a larger number of bubbles are generated and are traveling together; their collective ability to occupy large volumes will cause a noticeable reduction in the total flowrate of fluid through the pump. Additionally, the compressibility of bubbles can reduce the efficiency of the pump, resulting in a reduction in generated head.

If the total fluid flow is relatively slow – usually when the pump is operating at less than 50% of design speed, the bubbles may find the least resistive path by traveling backwards into the pump inlet. Backflow of bubbles seems counterintuitive, but one must remember that the lowest pressure point in a

system is generally the pump inlet. By analyzing the pressure of the system in the following analysis and in Figure 6, the ability for bubbles to travel backward during low flow rates can be seen.

$$P_{total} = P_{static} + P_{dynamic}$$

$$P_{total} = P_{static} + \frac{1}{2} \rho_{liquid} v^2$$

Equation 5 - Simplified Pressure Balance on Vapor Bubble

Force Balance on Cavitation Bubble

The bubble vapor density is small relative to the liquid, so the bubble does not have significant kinetic energy. Without kinetic energy, the bubble accelerates in the direction of the local force gradient, which is a sum of the local fluid forces and the local pressure gradient as shown in Figure 6. With a low flow, the pressure gradient in the pump may be stronger than the friction force of the flowing liquid, and accelerate the bubble upstream to the pump inlet. If the flow is not fast enough to entrain the bubble flow, the bubbles will travel to the point of lowest pressure. This behavior is called “surging” (Grist, 1999), and is discussed later. (This discussion does not include more complex forces that are present, such as turbulence forces, buoyancy forces, and virtual mass accelerations. However, the concept remains the same when these forces are included. The bubble will accelerate in the net direction of force.)

Since a cavitation bubble is in a vapor phase, its density is much less than that of the surrounding fluid. With a negligible mass, the bubble is very sensitive to the external forces acting on the bubble and will travel in the direction of net force. Considering only the two forces shown in Figure 6, the pressure force on

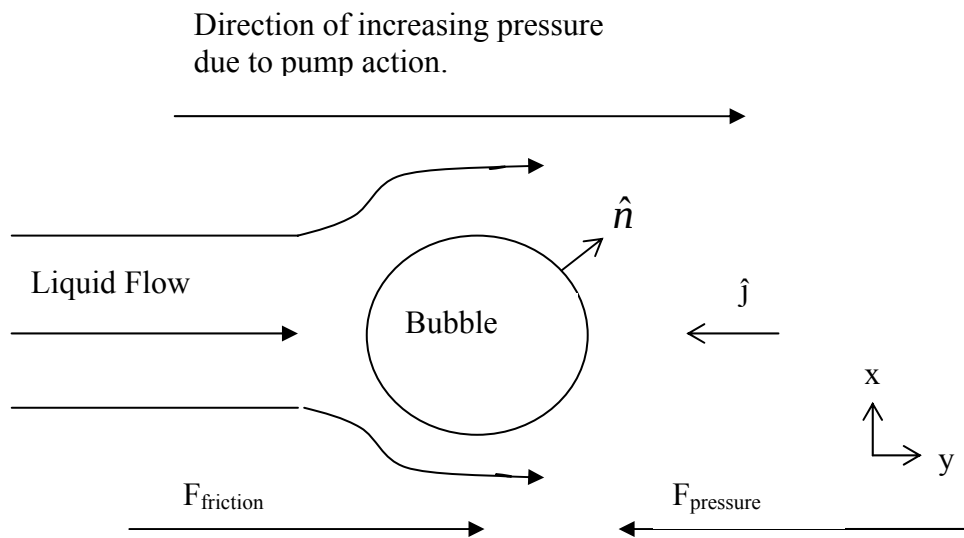


Figure 6 - Forces Acting on a Cavitation Bubble in a Pump Impeller

the bubble is given by the surface integral of the pressure gradient acting on the bubble (Equation 6).

$$\vec{F}_{friction\ engineering} = C_D (\vec{u}_{liquid} - \vec{u}_{bubble})^2 \left(\frac{1}{2} \right) \rho_{liquid} \frac{\pi D_{bubble}^2}{4}$$

Equation 6 - Pressure Force on a Cavitation Bubble

Here, $\hat{n}$ is the vector normal to the bubble surface, and $\hat{j}$ is the direction of the pressure gradient.

The friction force can be approximated using a drag coefficient (C_D).

$$\vec{F}_{pressure} = \oint_{bubble\ surface} P_{liquid} \hat{n} \cdot \hat{j} ds$$

Equation 7 - Friction Force on a Cavitation Bubble

The drag coefficient varies with the bubble shape, and the fluid acting on the bubble, but some drag coefficients have been experimentally determined. For water, Wallis proposed that several correlations for drag coefficients based on the Reynolds number and the Weber number. For turbulent flows, the Wallis drag coefficient for water was $C_D = We/3$ (Scheper, 2003). The $\frac{\pi D_{bubble}^2}{4}$ term is the frontal area of a spherical bubble, and is determined by the size of the bubble.

Lastly, if bubbles are produced at a very high rate, the bubbles may remain inside the impeller and occupy the entirety of the impeller volume. This condition is known as “vapor lock”, and will stop the flow of fluid through the pump.

Vapor lock is a condition in a cavitation process known as thermodynamic cavitation surging (Grist, 1999). Typically, vapor lock is problematic for pumps with a high energy density and fluids operating near saturation pressure, and with processes involving viscous fluids. In both cases, the pump may add sufficient energy to a localized region in the fluid to cause large-scale vaporization.

The vapor mass that is created is dependent on the amount of energy added to the local region of fluid and the heat of vaporization of the fluid:

$$\dot{m}_{\text{vapor}} = \frac{\dot{Q}}{h_{fg}}$$

Equation 8 - Mass Flowrate of Vapor

The rate of increase of vapor volume is proportional to the rate of increase of vapor mass:

$$\dot{V}_{\text{vapor}} = \frac{\dot{m}_{\text{vapor}}}{\rho_{\text{vapor}}}$$

Equation 9 - Volumetric Flowrate of Vapor

In a low pressure system, the vapor density is particularly low, and the rate of volume increase of the vapor can be very large. In low pressure systems, like condensate pumps in power plants, the risk of cavitation leading to vapor lock is high due to the large volume occupied by the vapor phase.

Material Damage

No matter how the bubble may propagate, the bubble collapse is what can cause damage to mechanical surfaces within the pump. The process of damage is

discussed with bubble collapse, but the likelihood of damage is a function of the bubble's location at the time of collapse.

The closer a bubble is to a surface, the greater its potential for damage. Depending on the flow, the bubble may travel to the middle of the fluid flow, where its collapse will leave all surfaces unaffected. However, the bubble may also travel closely along a surface, where its collapse will release damaging energy within the vicinity of the surface.

Two potential flowpaths are illustrated in **Error! Reference source not found.**. In one case, the bubble may travel away from the surface, where its collapse will have no effect on the surface. In the other case, the bubble travels back toward the surface and collapse in very close proximity. The bubble will travel back to the surface if the fluid pressure increases with distance from the surface or if net shear and turbulent forces create a “lift” force acting toward the surface. This pressure gradient can also play a major factor in the mechanism of surface damage during the bubble collapse.

Bubble Collapse

Once the cavitation bubble has reached its maximum growth and is in a region where the local head is positive, the bubble will collapse, and the vapor inside the bubble will return to liquid phase. Using energy considerations, Rayleigh demonstrated that the whole kinetic energy of the liquid may be described as (Shay, 1999):

$$\frac{1}{2} \rho \int_R^{\infty} \left(\frac{dr}{dt} \right)^2 4\pi r^2 dr = 2\pi \rho \left(\frac{dR}{dt} \right)^2 R^2$$

Equation 10 - Liquid Kinetic Energy During Bubble Collapse

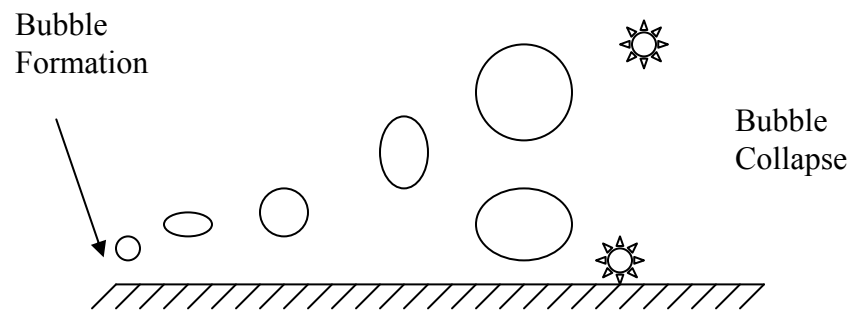


Figure 7 - Bubble Flowpaths From Birth to Death

Here, R is the radius of the bubble boundary as a function of time, and dr/dt is the velocity of the liquid at a distance r from the center of the bubble, where r is greater than R . For most of the duration of the bubble collapse, this equation has been proven to adequately describe the velocity of the bubble wall. However, as the bubble wall radius approaches zero, the velocity of the bubble wall approaches infinity. To avoid this physically impossible situation, an adiabatic collapse is generally considered more realistic. Using an adiabatic collapse, the velocity of the bubble wall can be described as:

$$\frac{3}{2} \rho \left(\frac{dr}{dt} \right)^2 = P_m (Z - 1) - Q \frac{(Z - Z^\gamma)}{1 - \gamma}$$

Equation 11 - Bubble Wall Velocity for Adiabatic Collapse

Here, Z is the ratio R/R_{max} , and γ is an experimental constant. This approach solves the infinite velocity problem as the radius approaches zero, and has been shown to provide good agreement for water. Experimentation has not been performed to verify the equation for mercury. The substantially greater surface tension of mercury may cause the collapse of a bubble in mercury to behave differently than in water, but Equation 11 does provide realistic wall velocities and satisfies the energy balance for the bubble collapse.

For water, the thermodynamic process line for collapse of a vapor bubble is generally considered to be adiabatic (Tillner, 1993; Shah, 1999). As a result, the collapse rate is much faster than the growth rate. In **Error! Reference source not found.**, the acoustic bubble collapses with a slope nearly approaching vertical, and on a time scale roughly an order of magnitude faster than the growth rate. The rapid bubble collapse will transfer the heat of vaporization of the bubble to the surrounding fluid and mechanical work is done to the fluid equal to:

$$\pi \int_{R_{bubble}}^0 \frac{4}{3} R_{bubble}^2 \Delta P dR = \dot{V} dP_{Bubble_to_liquid}$$

Equation 12 - Work Performed by Bubble Collapse

where ΔP is the pressure difference between the liquid and the vapor.

If the bubble has traveled close to a material surface, the pressure gradient across the bubble is such that the pressure of the fluid between the bubble and the surface is lower than the pressure on the side of the bubble opposite the surface (the far side of the bubble) (Tillner, 1993). As the bubble collapses, the pressure gradient will push liquid from the far side of the bubble through the middle of the bubble, where the collapse accelerates the fluid in a highly organized micro-jet toward the surface. In water, the jets are speculated to reach speeds of up to 200 meters/second. If the bubble is close enough to the surface, this jet will impinge on the surface. Unless the surface has sufficient strength to withstand the jet, the force of the fluid impinging upon the surface will shear material from the surface, causing a pitting corrosion (Tillner, 1993).

As the density of the fluid increases, the damaging potential of the bubble collapse increases. For example, Mercury, which has a density roughly 13.6 times greater than water, usually exhibits a cavitation erosion rate at least an order of magnitude faster than that of water (Tillner, 1993). The material of the surface will also play a factor, with specialized materials like Stellite exhibiting a much slower erosion rate than generic materials like cast iron (Tillner, 1993).

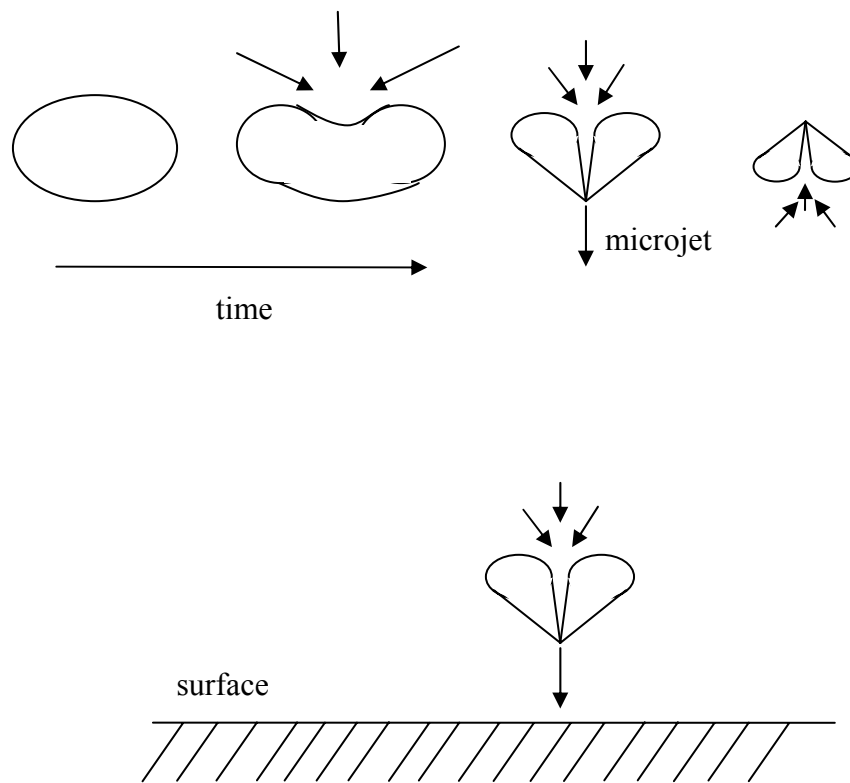


Figure 8 - Bubble Collapse and Liquid Microjet

FORMS OF CAVITATION DAMAGE

In a pump, flow cavitation has the potential to cause two general undesirable effects: reduced pump efficiency and material damage. The reduced pump efficiency can be seen in the loss of head provided by the pump, and by the reduction in flow of the fluid. Generally, these effects are considered limitations on the operations of a pump, and are avoided by choosing pumps known to provide the performance required. The material damage that can occur from cavitation can drastically reduce the life of pump components, resulting in excessive pump shutdowns to repair the damage or to replace the components. In the worst case, cavitation can locally pressurize a system to levels high enough to destroy containment of the flow loop.

There are four general categories of cavitation damage (Grist, 1999). All four forms are typically the result of improper design or operation, and can usually be avoided with the proper engineering. Here, the mechanism of each form of cavitation damage is described, and their common indicators are discussed.

Thermodynamic Cavitation Surging

Thermodynamic cavitation is usually the most violent and catastrophic form of cavitation a pump can experience (Grist, 1999). Here, the liquid inside the impeller rapidly vaporizes, or “flashes”. As the newly formed gas tries to expand, a very large increase in pressure is experienced in the system. This pressure spike creates a shock wave that travels through the fluid and can breach the piping barrier at any sufficiently weak points. Commonly, such a weak point is found near the pump inlet, at the connection between the pump inlet header and the system piping. The pipe burst can then release a large amount of liquid and

vapor that can be lethal to nearby personnel. Additionally, the shock wave can dislodge equipment. Though all forms of cavitation are generally undesirable, thermodynamic surging is the most destructive.

The destructive potential of the energy applied to a fluid by a pump can be surprisingly great. For example, in a recirculation pump for a pressurized water reactor (PWR) or boiling water reactor (BWR), the pump may apply about 5,000 horsepower of energy to a fluid volume of approximately 10 liters. With this, the energy addition density is:

$$\begin{aligned}\bar{E} &= \frac{\dot{Q}_{pump}}{V_{fluid}} = \frac{5,000hp}{10l} \\ \bar{E} &= 500hp/l \\ \bar{E} &= 373 * 10^6 W / m^3\end{aligned}$$

Equation 13 - Energy Addition Density

In contrast, the average power addition density in a reactor core may be about $54.1 * 10^6 \text{ W/m}^3$ in a BWR, or about $105 * 10^6 \text{ W/m}^3$ in a PWR (Todreas, 1990). The pump clearly does not add more energy than the core (due to a much smaller fluid volume), but the concentration of added energy can be sufficient to vaporize the liquid, particularly if the liquid is close to saturation pressure.

The basic condition that creates thermodynamic surging is the rapid transfer of very large amounts of energy into the fluid, and subsequent conversion of the energy into thermal energy. For example, suppose a high-energy pump is operating when flow is rapidly stopped (e.g. a valve is suddenly closed or a break in the line occurs upstream). With the sudden stoppage of flow, the pump energy will be transferred through the impeller into the fluid. Since the fluid has nowhere to go, the energy will be entirely deposited into the fluid within the

impeller, causing the fluid to heat up. If the energy is sufficient, the fluid may gain enough energy to convert to vapor, and cause thermodynamic cavitation.

Even if thermodynamic cavitation surging does not destroy line equipment or cause a rupture, the system response is very characteristic; there is a very sharp increase in the inlet pressure that will plateau at some high level. If the condition is not alleviated, the pressure will remain high. If pressure relief is available, the system will experience occasional drops in pressure; the pressure will then cycle between moments of high pressure and low pressure in a very typical and easily identifiable fashion. A pressure history can be seen in **Error! Reference source not found.**

The keys to avoiding cavitation surging for large pumps or very fast pumps include procedures that prescribe gentle startups and shutdowns to avoid rapid transfers of thermalizing energy into the fluid. If sudden flow stoppages are a risk, consideration should be given to pressure relief systems near the pump to limit the pressure transient and to protect the system.

Hydrodynamic Cavitation Surging

Hydrodynamic surging is another relatively violent form of cavitation, though it does not have the same potential for damage as thermodynamic surging. If a pump is operated for abnormally low flowrates, cavitation bubbles may start to form at the inlet tip of the impeller blades. The bubbles are formed from a low NPSH and from the shear between the fluid and the impeller inlet tip due to the angle of attack of the fluid. Because the flowrate is low, the bubbles may travel back into the impeller inlet flow backwards through the loop for a short distance to a location of low pressure. The bubbles usually act in concert; a train of bubbles will escape the impeller and travel back through the impeller inlet, and then reverse course and flow back into the impeller. The cycle will repeat regularly unless relief is provided.

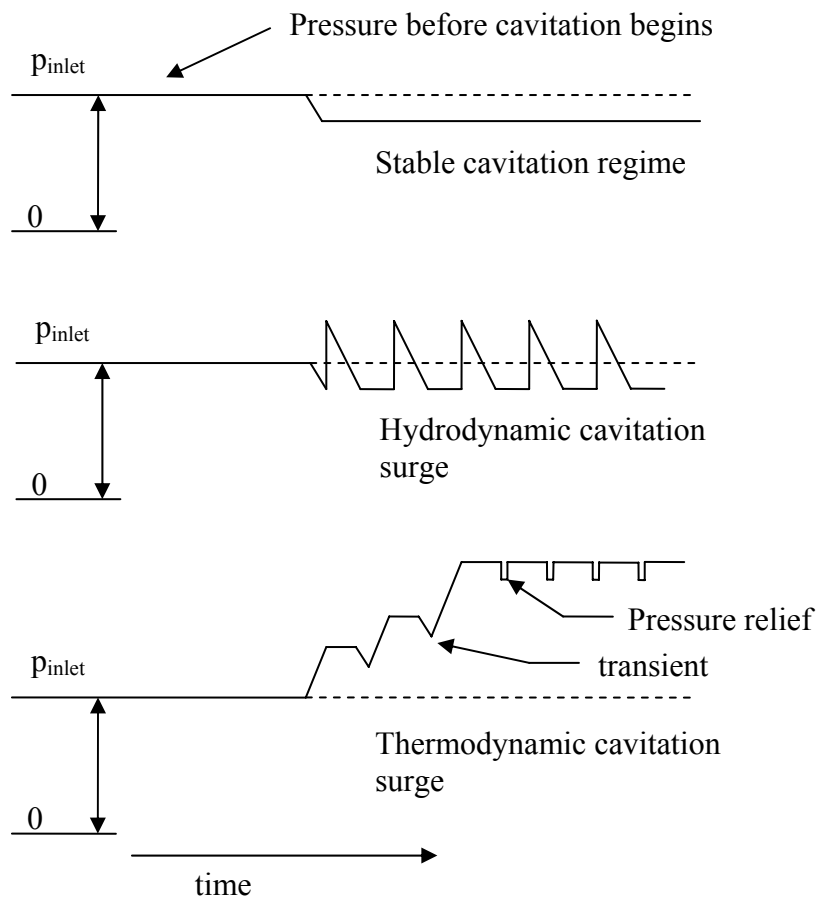


Figure 9 - Various Cavitation Pressure Transients

The cyclic pattern of bubble backflow and recirculation produces a very distinctive “chugging” sound and a corresponding inlet pressure pattern (Grist, 1999). The inlet pressure will exhibit a series of regular spikes for the duration of the cavitation event. The pressure spikes are the result of the acceleration of the bubbles into the inlet, with pressure diminishing as the bubbles flow back in the flow direction.

The most common reason for hydrodynamic surging is the attempt to operate a pump too far below its design operating speed. For example, a pump may be operated at half speed to maintain a slow loop circulation during down times. As a result, the angle of attack may be mismatched with the flow. The system NPSH may drop too low to prevent cavitation and the low speed is unable to force the bubbles to follow the normal flow path. Since the inlet is usually the point of lowest pressure in the system, the bubbles will flow to the low pressure region.

Hydrodynamic surging is usually avoided by design. The easiest solution is to use two half-size pumps; the two pumps can run in parallel during full-speed operation, and the system can be run on one pump for slow-speed operation. Having two pumps can provide a redundancy, where one pump can maintain limited circulation if the other pump fails.

Hydraulic Performance Loss

Hydraulic performance loss is the most common condition for cavitation within a centrifugal pump. In this form of cavitation, bubbles are formed within the impeller, usually at the impeller blade inlet tips. As the bubbles form, they are carried downstream by the fluid flow. After a short period of growth, the bubbles will rapidly collapse and transfer energy to the surrounding fluid.

Since the specific volume of the vapor is much greater than that of the liquid, the bubbles occupy a very large amount of the available flow space within the impeller. The two-phase liquid has a lower density than the single phase liquid, so the rotational kinetic energy added to the fluid by the pump is reduced. As a result, the ability of the pump to effectively move the fluid is diminished. The system will exhibit a reduction in the generated head and/or the flowrate of the fluid. If the condition is mild, the only evident signs are the diminished head and the noise generated from the bubble collapses. If the condition is severe enough, the pump may be unable to deliver the demanded flow.

Insufficient NPSH is the common reason for hydraulic performance loss. Pump manufacturers typically state the minimum NPSH for operation to avoid this condition. However, the recommended value is normally for the best operating point of the pump; operation of the pump at other than ideal conditions will generally require more NPSH than at the best operating point. The additional NPSH compensates for the operation of a pump outside of nominal conditions. If a stock pump is selected for a particular application, an analysis of the operating conditions should be done to verify that sufficient NPSH is available.

Cavitation Erosion

Cavitation erosion is the condition where the collapse of cavitation bubbles near a surface causes material to be removed from the surface. Though the growth of a cavitation bubble takes a short time, the collapse of a cavitation bubble is nearly instantaneous. Normally, the collapse of the bubble simply transfers a brief, intense burst of mechanical energy to the surrounding fluid. The total energy involved in bubble collapse is small relative to flow energy, but it is very focused and intense. If the collapse occurs near a surface, the pressure gradient surrounding the bubble causes the bubble to collapse asymmetrically. As fluid from the high-pressure side “pushes” into the bubble, the collapse acts to

accelerate the fluid into a tiny jet stream. The jet can achieve velocities as high as 200 m/sec. This jet of fluid impinges upon the surface, and can cause pitting in the surface at the point of impact (Tillner, 1993). For water, the potential pressure exerted by the jet on a surface can be approximated as follows:

$$P_{jet} = P_{dynamic} = \frac{1}{2} \rho v^2$$

$$P_{jet} = \frac{1}{2} \left(1000 \frac{kg}{m^3} \right) \left(200 \frac{m}{sec} \right)^2$$

$$P_{jet} = 20 MPa$$

Equation 14 - Potential Pressure Exerted by Water Jet

Since mercury has a density approximately 13.6 times that of water, the jet pressure could theoretically reach a maximum localized pressure of over 250 MPa.

Unlike the first three forms of cavitation damage, cavitation erosion is not a specific evolution of vapor formation within the centrifugal pump. Instead, it is the result of the collapse of the bubble at the end of its life. Cavitation erosion is usually of greatest concern during hydraulic performance loss. Here, cavitation is often sustained for long periods of time, and erosion can cause very significant wear.

Erosion is often the principal concern associated with cavitation. Though hydraulic performance loss creates a cost in performance and operating expense, erosion can cause enough damage to an impeller to significantly limit or even prevent operation. Since the explicit detection of erosion during pump operation is usually impossible, a more suitable approach to protecting against erosion is to avoid the conditions where any form of cavitation may begin. Since cavitation – and cavitation erosion – may occur without noticeable performance loss,

operation of a pump near performance loss conditions is not recommended (Blevins, 1994).

Bubble Flow Characteristics

Error! Reference source not found., Error! Reference source not found., and Error! Reference source not found. illustrate the cavitation bubble flow patterns for hydraulic performance loss, hydrodynamic surging, and thermodynamic surging respectively. In hydraulic performance loss, the bubbles continue to travel in the direction of liquid flow, allowing for a stable condition where bubbles form at the low pressure point (typically the leading edge of the impeller blade) and collapse downstream. In hydrodynamic surging, the low flowrate of the liquid fails to provide sufficient drag force to maintain bubble flow in the direction of liquid flow. The bubbles may travel into the pump inlet and establish a cyclical pattern of growth and collapse, producing a “chugging” behavior in the liquid flow. In thermodynamic surging, the bubbles often grow enough to fill the impeller region, blocking the impeller and preventing fluid flow (Grist, 1999).

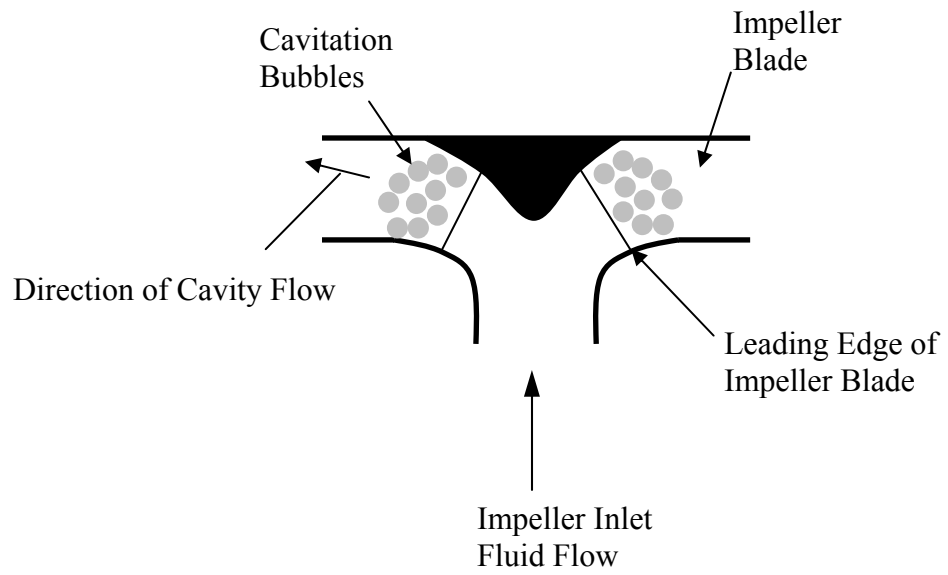


Figure 10 - Pattern of Cavitation Associated With Cavitation Erosion

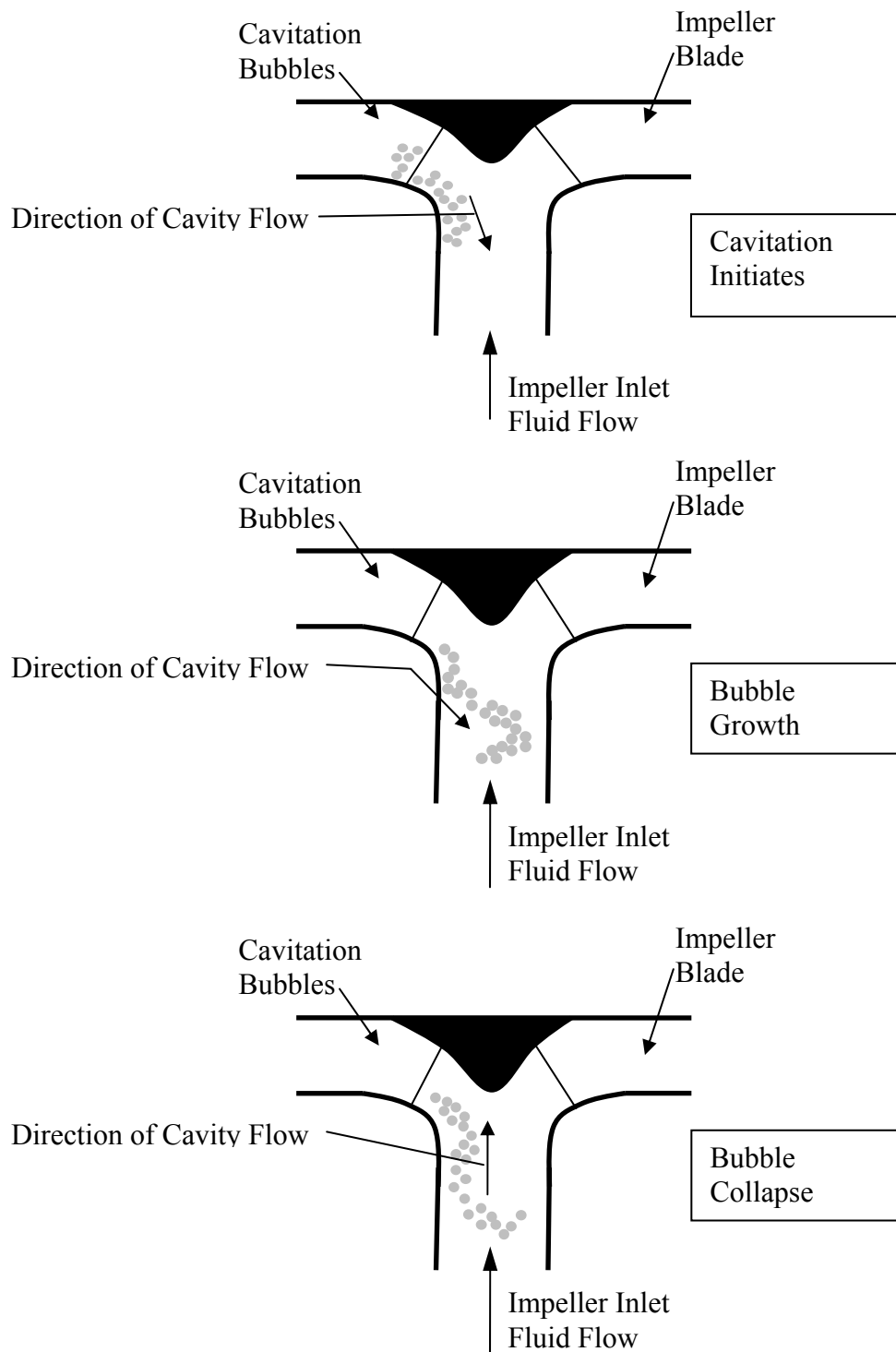


Figure 11 - Cavitation Pattern During Hydrodynamic Surging

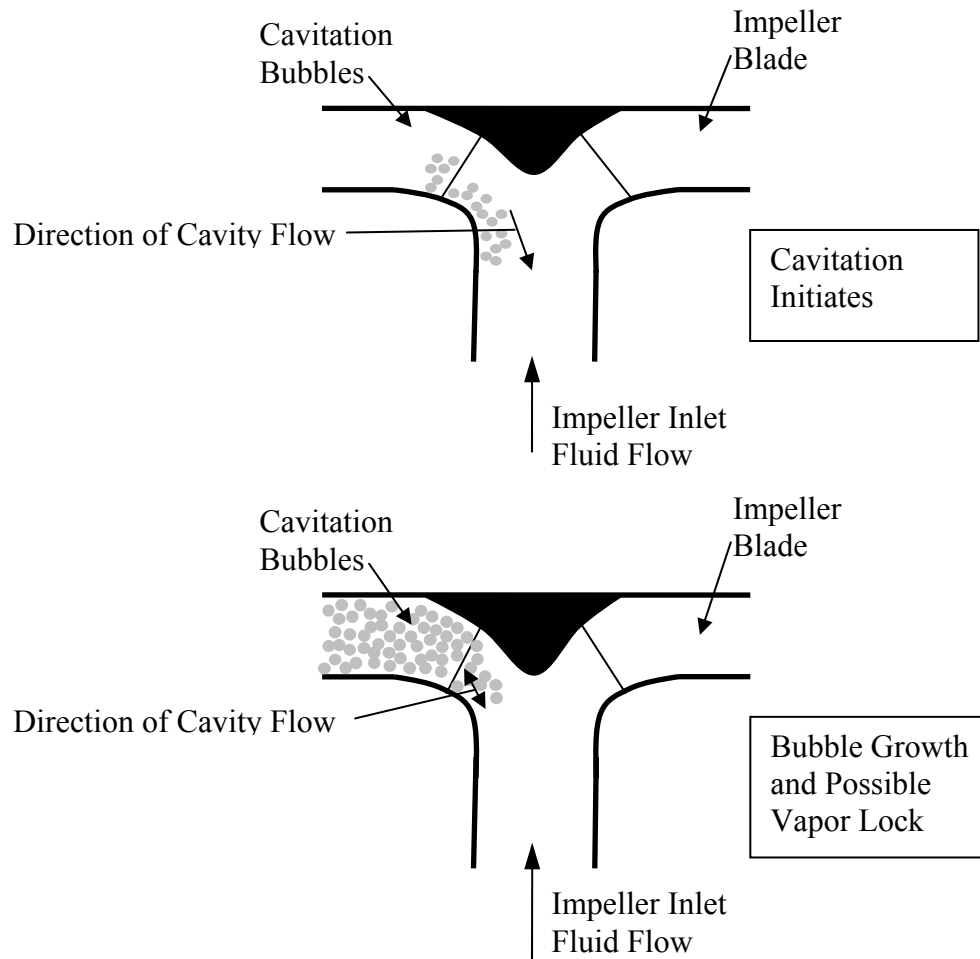


Figure 12 - Cavitation Pattern For Thermodynamic Surging

MATHEMATICAL ANALYSIS

The nature of cavitating flow is very complex. Though predictive modeling methods have been improving over the last several decades, most mathematical modeling of cavitation flow is simply parameterization; the physical metrics of flow are analyzed to determine when cavitation may start, and how much effect cavitation may have on flow in general. These metrics are typically tested and recorded for every pump. For most pumps, the required NPSH is determined by experimentally determining the conditions for cavitation inception. For a given pressure rise across a pump (Δp) and volumetric flowrate (V), the inlet pressure is gradually lowered until a 3 percent drop in flowrate is observed. The inlet pressure at which the flowrate drops by 3 percent is the point of cavitation inception. To determine pump performance curves, a similar procedure is used. For a given pressure rise across the pump, the volumetric flowrate is slowly increased by increasing pump speed. As the flowrate is increased, the pump will eventually no longer be able to supply sufficient energy to the fluid, and increasing the pump speed further will not increase the flowrate of the liquid. As cavitation occurs, the flowrate will begin to diminish. The point of cavitation inception is defined to be when the flowrate decreases by 3 percent of the maximum observed flowrate.

Classical Approach

The most common approach to analyzing the risk of cavitation in a centrifugal pump is to compare the available NPSH of the system to the required NPSH of the pump. A safety margin is usually applied to the required NPSH (e.g. 1.3) to account for any pressure fluctuations or uncertainties within the flow. So long as the operating NPSH stays above this threshold, the risk of cavitation is low.

Sometimes, the required NPSH for a pump may not be explicitly known. In this case, the most common method of determining the required NPSH for a pump is to use a test facility. There are several possible layouts for testing a pump, but the most common feature is a method of controlling the pressure, or head, at the inlet of the pump. Usually, the head is controlled by the fluid level in the inlet vessel (feed tank). By slowly decreasing the level of fluid in the feed tank, the NPSH can be slowly reduced and the inception of cavitation can be easily determined. This provides the reference point required to assess safe pump operation limits.

Nondimensional Analysis and Scaling Factors

Many attempts have been made to find nondimensional parameters and scaling factors for pump performance and the onset of cavitation. However, a complete canon of analytical tools continues to elude researchers. The complexity of the fluid flow through a pump impeller has been sufficiently difficult to model that finding approximation metrics is based almost solely on application-specific testing and data acquisition.

The Thoma coefficient is the most common relationship applied due to its simplicity. In it, the NPSH is compared to the delivery head of the pump. Larger Thoma coefficients are less prone to cavitation than lower values; a common recommendation is to maintain a Thoma coefficient of 10 or greater (Grist, 1999).

$$\sigma_{Thoma} = \frac{NPSH}{H}$$

Equation 15 - Thoma Coefficient

The coefficient of cavitation is a relationship between the NPSH and the flow speed of the pump. The basic theory is that a faster fluid requires a greater

amount of static pressure to prevent bubble formation due to the increased kinetic energy of the flow. Due to the popularity of the Thoma coefficient, this parameter has not seen extensive use.

$$\kappa = \frac{NPSH}{U^2 / 2g}$$

Equation 16 - Coefficient of Cavitation

The suction specific speed is becoming as popular as the Thoma coefficient for pump parameterization. In it, the NPSH is related to the volumetric flowrate and the angular velocity of the impeller. This parameter has proven to be rather consistent for water, and can be broadly applied across a wide range of impeller designs. The greatest use of the suction specific speed has been to avoid the surge types of cavitation, whose onsets are related to an insufficient flowrate.

$$N_{ss} = \frac{\omega \sqrt{Q_{opt}}}{(NPSH)^{3/4}}$$

Equation 17 - Suction Specific Speed

Concluding Remarks on Cavitation Theory

A thorough, explicit analysis of pump cavitation continues to elude researchers. Extensive research has been applied to water cavitation due to the overwhelming need to pump water compared to other fluids; still, even water analysis is mostly empirical and is based on the avoidance of observed cavitation onset parameters.

Adapting the data from water cavitation experiments to other fluids is still beyond the state of the art. Fortunately, insufficient NPSH drives cavitation for all fluids. As research continues in this area, scalable factors may yet emerge.

Though explicit prediction of cavitation is still difficult, the *detection* of cavitation is much easier. Significant cavitation produces characteristic pressure variations, particularly at the pump inlet. The variations can be readily observed with dynamic pressure instrumentation or accelerometers. If the pump is in an accessible location, a nearby observer can often hear the noise produced by the collapsing bubbles.

The most common form of cavitation in a centrifugal pump is hydraulic performance loss, and is often coupled with cavitation erosion. Here, the two most common solutions are to either slow the pump down, or to increase the inlet pressure to generate more NPSH. If cavitation surging (either hydrodynamic or thermodynamic) is observed, the most common cure is to increase the flowrate and to increase the NPSH.

Dimensionless Analysis Parameters for Scaling Mercury and Water Cavitation Inception

Dimensionless analysis is a very common approach to solving problems of fluid flow. Rather than attempting to directly model the conditions of a flow through mechanistic equations, dimensionless analysis attempts to find ratios of flow and material properties. Often these property groups are extracted from physical models, as the Reynolds number is extracted from the Navier-Stokes equations for fluid flow. Much like benchmark tests in software design, these ratios can be compared to known fluid flow conditions to find similarities and provide predictions of the flow conditions. This chapter first introduces

nondimensional and dimensional performance factors, and then compares these factors for water and mercury pumping applications.

Dimensionless Analysis in Fluid-Structure Interactions

Extensive study on the use of dimensionless parameters for the study of flow-induced vibrations was performed by Dr. Robert D. Blevins (Blevins, 1977). While his work focuses primarily on the vibrations of structural objects within a free stream flow, it does provide some useful insight on the interaction between fluids and structures as occurs inside a pump. He proposed the use of the following parameters for analysis:

$$\text{Geometry} = \frac{l}{D}$$

Equation 18 - Geometry Factor

The geometry ratio is useful for scaling dimension, such as length to width and surface roughness to width.

$$\text{Reduced Velocity} = \frac{U}{fD} = \frac{1}{\text{Strouhal}}$$

Equation 19 - Reduced Velocity

Reduced velocity is useful in analyzing the frequency of vortex shedding in wakes as a fluid passes a structure. Here, “U” represents the mean flow of the fluid, and “D” represents the width of the structure normal to the direction of fluid flow. “f” is the frequency of the vibration. For small reduced velocities, Blevins suggests that the interaction between the structure and the fluid wake is very strong.

$$\text{Mass Ratio} = \frac{m}{\rho D^2}$$

Equation 20 - Mass Ratio

The mass ratio “provides a measure of buoyancy effects and the inertia of the model relative to that of the fluid”. The “m” is the mass of the model per unit length, and the “D” is a characteristic dimension of the model. With a decreasing mass ratio, the structure becomes increasingly prone to vibration. Since Mercury’s density is roughly 13 times greater than water, the likelihood of flow-induced vibrations is significantly greater for mercury than for water.

$$\text{Reynolds Number} = \frac{UD}{\nu} = \frac{\rho UD}{\mu}$$

Equation 21 - Reynolds Number

The Reynolds number is very common in fluid analysis and provides a gauge of the turbulence of the fluid flow and scales inertial effects to viscous effects. The manner of separation of fluid flow from the structure as the fluid passes by is a function of the Reynolds number. Comparing mercury to water, we see that the higher density of mercury will serve to increase the Reynolds number, indicating greater flow turbulence. The dynamic viscosity is also typically lower for mercury, which will further increase the risk of vortex shedding.

$$\text{Mach Number} = \frac{U}{c}$$

Equation 22 - Mach Number

The Mach number is a measure of the importance of compressibility in a fluid flow. Compressibility is usually negligible for Mach numbers less than about 0.3.

$$\text{Damping Factor} = \zeta$$

Equation 23 - Damping Factor

The damping factor is a measure of the fraction of vibration energy that is dissipated by a structure per cycle. If the energy input is less than the dissipated energy, vibration amplitudes will diminish and resonance buildup will not be a problem. In a liquid, acoustic damping is generally proportional to the liquid density times the speed of sound of the liquid. However, the flow structure will also play a part in damping. The influence of the piping and flow components on damping can also be important. Measurement of the damping factor requires dynamic testing of the structure, and is not available for many centrifugal pumps.

The damping factor can play a key role in structural wear. In sump-type centrifugal pumps, the vibrations of the impeller are transmitted to the shaft bearings by the impeller shaft. In a long-shaft design, the moment arm of the shaft can create very large forces in the bearings due to impeller vibrations. If the damping factor of the pump system is low (i.e. little vibrational energy is dissipated per cycle). The flow forces will do work on the impeller and energy will integrate over many cycles. The integral forces transmitted to the shaft bearings may prematurely wear the bearings, and may result in premature bearing failure.

Other Nondimensional Numbers and Performance Measures

There are several other nondimensional factors used in the analysis of bubble dynamics. These are discussed by Christopher Brennen (Brennen, 1995):

$$\text{Reduced Temperature} = \frac{T}{T_{crit}}$$

Equation 24 - Reduced Temperature

Use of the reduced temperature allowed Brennen to correlate the bubble behavior of various fluids – including mercury and water – with a similar set of parameters. The reduced temperature ratio has been used in many problems in fluid flow, particularly when liquid and vapor phase changes are involved.

The critical temperature is the temperature of the critical point of the fluid. The critical point is the pressure and temperature pair at which the latent energy difference between a fluid's liquid and vapor phases ceases to exist. The use of the critical point for reduced properties is a relatively old concept in thermodynamics. Many older fluid and gas tables were based on reduced properties.

One previous attempt to correlate cavitation inception between water and a liquid metal in a centrifugal pump was performed by Albert C. Grindell in 1957. Grindell attempted to predict the pump suction static head (H_{ci}) of Sodium-Potassium by adding the difference between the pump suction static head and the liquid vapor pressure (H_{vp}) of water to the vapor pressure of the Sodium-Potassium liquid (Grindell, 1957), as seen in Equation 25. Grindell's correlation is not nondimensional, but is rather a direct correlation between the vapor pressures of water and a liquid metal.

$$H_{ci,NaK} = (H_{ci,H_2O} - H_{vp,H_2O}) + H_{vp,NaK}$$

Equation 25 - NaK Cavitation Inception Head

The data in Table 1 represents the average values of a series of tests; each series of tests is considered a “run” in Grindell's terminology. Each “run” of water at a given pump speed was compared to at least one similar “run” of NaK at the same pump speed. At about 3380 rpm, the pump produce a flow of about 430

Table 1 - Grindell Data for Water-NaK Cavitation Correlations

Water Tests						Sodium-Potassium Alloy							
	N _{av}	Q _{av}	T	H _{ci}	H _{vp}	N _{av}	Q _{av}	T	H _{vp}	Estimated H _{ci}	Alloy Test H _{ci}		
	(rpm)	(gpm)	(°F)	(ft abs)	(ft)	(rpm)	(gpm)	(°F)	(ft)	(ft abs)	(ft abs)	(ft)	(%)
Run 1	3375	306	188	45.1	21.3	3390	308	1490	40.5	64.3	63.3	+1.0	1.6
Run 2	3374	436	138	39.5	6.4	3383	435	1501	43.0	76.1	79.8	-3.7	-4.7
Run 3A	3003	306	188	43.2	21.3	3018	310	1502	43.2	65.1	68.0	-2.9	-4.3
Run 3B						3030	305	1497	42.1	64.0	65.5	-1.5	-2.3
Run 3C						3047	307	1500	42.7	64.6	65.0	-0.4	-0.6
Run 4A	2009	436	138	37.0	6.4	3000	432	1493	41.2	71.8	75.4	-3.6	-4.8
Run 4B						3000	432	1503	43.5	74.1	79.0	-4.9	-6.2
Run 4C						2985	435	1503	43.5	74.1	77.6	-3.5	-4.5
Run 5	2603	304	188	40.2	21.3	2601	303	1481	38.5	57.4	59.0	-1.6	-2.7

gpm, and at 2600 rpm, the flow was about 300 gpm. The water temperature was varied to test the dependence of the water temperature on the correlation. The final two columns are a comparison of the predicted and measured H_{ci} values for the NaK alloy. Also noteworthy is that the tests were never concerned with correlation of *damage* from cavitation, but only the *inception* of cavitation. The results are given graphically in Figure 13.

At first inspection, there seems to be a strong correlation between the vapor pressures and the pump suction static head at cavitation inception. Despite the dramatic differences in temperature and the disregard for state properties such as viscosity and density, the maximum deviation in a series of tests was found to be 6.2%. However, the total range of variation in the test series is less than 35%. Grindell himself noted, however, that difficulty was encountered when making temperature measurements for the fluids, particularly the 1500-°F NaK alloy. With the uncertainties in the alloy temperature, the error encountered by the correlation cannot be conclusively attributed to any single source. The dependence of other state variables is unknown, and cannot be conclusively dismissed. Nevertheless, the data presents a reasonable argument for a strong correlation between the suction head and vapor pressure.

Smithsonian Physical Tables

The Smithsonian Physical Tables gives a partial list of vapor pressures for various fluids across a range of temperatures. Table 2 displays the Smithsonian data for vapor pressures for Mercury, where the columns represent the units digit of the temperature. For example, the first entry under column “5” is for 275 °C.

Since the Smithsonian data does not directly extend to the temperature range of the SNS facility, further data is needed. A useful correlation was presented by Alan Menzies:

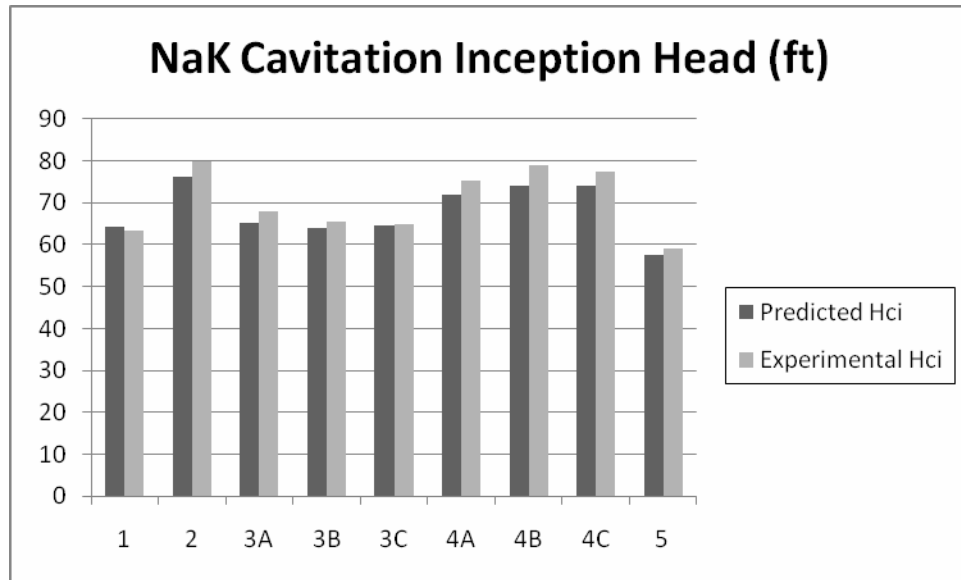


Figure 13 - Grindell Data for Water-NaK Cavitation Correlation

$$\log_{10}(p) = 9.9073436 - \frac{3276.628}{\theta} - 0.6519904 \log_{10}(\theta)$$

Equation 26 - Menzies Correlation

The Menzies correlation (Menzies, 1917) gives the vapor pressure of Mercury in mmHg for temperatures in Kelvin. The correlation provides results within 0.5% accuracy for measured values of mercury vapor pressure near 120 °C.

As is expected, the vapor pressure for Mercury is rather low in the relatively cool regions of this table. Extrapolating this data to 60 °C (140 °F) – the inlet temperature for the SNS pump is 60 °C – suggests that the vapor pressure for Mercury is very small. If, as Grindell attempted to show, a strong correlation exists between vapor pressure and pump suction static head for cavitation inception, then Equation 25 would simplify to the following.

$$H_{ci,Hg} \cong (H_{ci,H_2O} - H_{vp,H_2O})$$

Equation 27 - Simplified Menzies Correlation

Again, the data presented above is not sufficient to verify the correlation. However, it does provide a reasonable starting point for correlations between Mercury and water cavitation within centrifugal pumps.

Further Nondimensional Analysis

Significant emphasis has been placed by Brennen on the use of the temperature / critical temperature ratio $\left(\frac{T}{T_{cr}}\right)$ in nondimensional analysis. For example, the ratio of densities of saturated liquid and saturated vapor $\left(\frac{\rho_l}{\rho_v}\right)$ as a

Table 2 - Mercury Vapor Pressures (Smithsonian, 2003)

Temp (°C)	Vapor Pressure in Mercury (mmHg)									
	0	1	2	3	4	5	6	7	8	9
270	123.92	126.97	130.08	133.26	136.5	139.81	143.18	146.61	150.12	153.7
280	157.35	161.07	164.86	168.73	172.67	176.79	180.88	185.05	189.3	193.63
290	198.04	202.53	207.1	211.76	216.5	221.33	226.25	231.25	236.34	241.53
300	246.81	252.18	257.65	263.21	268.87	274.63	280.48	286.43	292.49	298.66
310	304.93	311.31	317.78	324.37	331.08	337.89	344.81	351.85	359	366.28
320	373.67	381.18	388.81	396.56	404.43	412.44	420.58	428.83	437.22	445.75
330	454.41	463.2	472.12	481.19	490.4	499.74	509.22	518.85	528.63	538.56
340	548.64	558.87	569.25	579.78	590.48	601.33	612.34	623.51	634.85	646.36
350	658.03	669.86	681.86	694.04	706.4	718.94	731.65	744.54	757.61	770.87
360	784.31									

function of the critical temperature ratio exhibits the same trend for a variety of Newtonian fluids (Brennen, 1995). Of particular note, while the shapes of the ratio functions are similar, the magnitudes tend to vary according to the degree of electrostatic interactions between molecules within the fluid. For example, Helium-4 exhibits the lowest density ratio at any given critical temperature ratio, and water exhibits the highest density ratio. Presumably, the effects of the Van der Waals forces within the water, enhanced by the hydrogen atoms, serve to hold the molecules closer together, thereby increasing the density of the liquid. The electrostatic forces would have minimal effect in the vapor phase, where the molecules tend to be spread further apart. Interestingly, Mercury bears roughly the same density ratio as Hydrogen (H_2). Since the density ratio for Mercury is among the lowest for the fluids compared by Brennen, there may be a weaker electrostatic tendency to resist cavitation than in water.

Evidence of the reduced electrostatic cavitation resistance of mercury can be found in mercury's nonwetting nature. The high self-affinity of mercury reduces mercury's tendency to "wet", or adhere to surfaces, as shown in Figure 14.

Since mercury does not strongly adhere to most surfaces, less energy is required to separate the mercury from the surface during bubble formation. The reduced energy requirement may promote bubble formation at higher pressures than if the mercury was strongly adhered to the surface.

Brennen uses the critical temperature ratio as a benchmark for the *thermodynamic parameter*, Σ . The thermodynamic parameter, in equation form, is given in Equation 28 (Brennen, 1995).

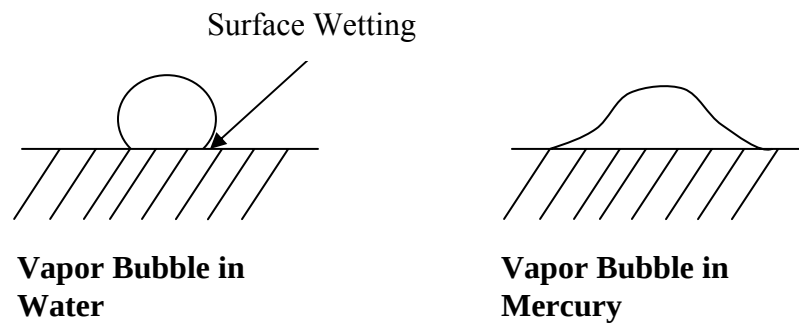


Figure 14 - Vapor Bubbles against a Surface in Water and Mercury

$$\Sigma(T) = \frac{L^2 \rho_v^2}{\rho_L^2 c_{PL} T_\infty \alpha_L^{\frac{1}{2}}}$$

Equation 28 - Thermodynamic Parameter

The thermodynamic parameter is the result of a derivation of the Rayleigh-Plesset equation for bubble dynamics with a first-order Taylor expansion as an approximation for small temperature differences between the bubble and the surrounding fluid.

With the thermodynamic parameter, the temperature of the fluid can be related to the vapor pressure of the bubble and to the “critical time” of bubble growth – the time required for the thermal term in the Rayleigh-Plesset equation to gain equal magnitude to the inertial terms (Brennen, 1995). This correlation is important in determining whether the bubble dynamics are “inertially controlled” or “thermally controlled”. Generally, low critical temperature ratios correspond to inertially controlled growth, where the growth of the bubble is not measurably dampened by thermal considerations.

COMPARISON OF MERCURY AND WATER PROPERTIES

In order to compare the physical behavior of mercury and water in a flow, a solid understanding of the differences in fluid properties between the two fluids must be reached. Many numerical methods of representing the properties of water exist, but few are as convenient as the *XSteam* tables for MATLAB and Excel written by Magnus Holmgren (Holmgren, 2006). The tables provide water properties as a function of temperature and pressure, whereby a user input of pressure gives a series of properties ranging from 0 C to 300 C. The water tables were then compared against mercury properties.

An extensive survey of literature on mercury fluid properties was performed by H. Cords (Cords, 1998) for the European Spallation Source (ESS). His efforts produced a concise report of numerical approximations of mercury properties based on the available literature. These findings give a series of mercury properties in the range of 0 C to 300 C based on the user input of pressure. Due to the paucity of data on mercury flow properties, most mercury properties are compared below at 1 bar, which is the most reliable pressure for the equations.

In the comparative graphs, water properties are calculated at 1000 bar, and mercury properties are calculated at 1 bar. The different pressures are used to allow an illustration of the behavior of water as a fluid from 0 C to 300 C, and to utilize the mercury equations at their most universally accurate pressure. The temperature trends of the fluid properties can then be compared to determine how mercury may differ from water, and how those differences may influence the characteristics of cavitation in mercury.

In **Error! Reference source not found.**, the density of water is compared to the density of mercury. Since water is generally considered incompressible, and mercury is approximately 13 times less compressible than water, the difference in pressure between the water and mercury does not significantly affect the comparison of densities. This will influence any inertial effects in the flow.

The thermal conductivity of mercury, shown in **Error! Reference source not found.**, is significantly greater than water, and increases sharply with temperature. The ability of mercury to efficiently dissipate thermal energy will decrease the likelihood of thermodynamic cavitation since it will be more difficult to establish the relatively large thermal gradients that often result in localized boiling in subcooled fluids. However, evaluation of thermodynamic cavitation also requires evaluation of density and specific heat, which will be discussed later in this section.

Shown in **Error! Reference source not found.**, the speed of sound serves as a measure of the compressibility of a fluid. Mercury and water are relatively incompressible fluids. They have similar magnitudes of sound speed, but mercury is a much less compressible fluid due to its greater density (Equation 29). However, the temperature does play a significant role in determining the degree of difference between the fluids.

$$c^2 = \frac{\partial P}{\partial \rho}$$

Equation 29 - Speed of Sound Related to Density and Compressibility

Mercury has an extremely low specific heat capacity value compared to the specific heat capacity of water, which is typical for a liquid metal. The specific heat capacities are shown in **Error! Reference source not found.** The low specific heat of mercury will increase the likelihood of thermal cavitation

since less energy is required to significantly change the local temperature of the fluid. This effect is counter to the difference in thermal conductivity.

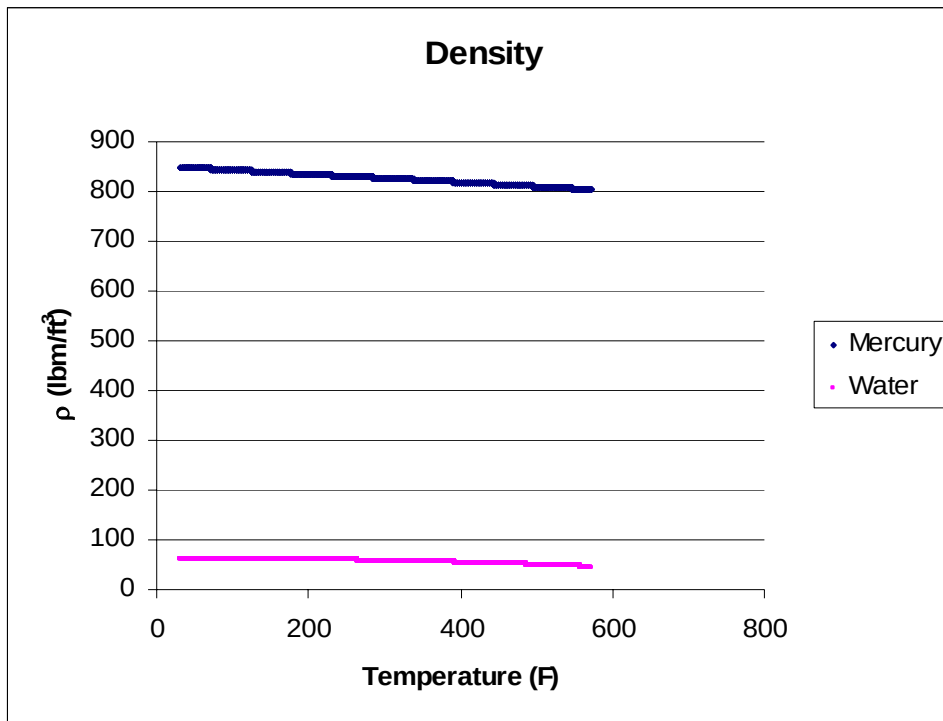


Figure 15 - Density (Water at 100 Bar, Mercury at 1 Bar)

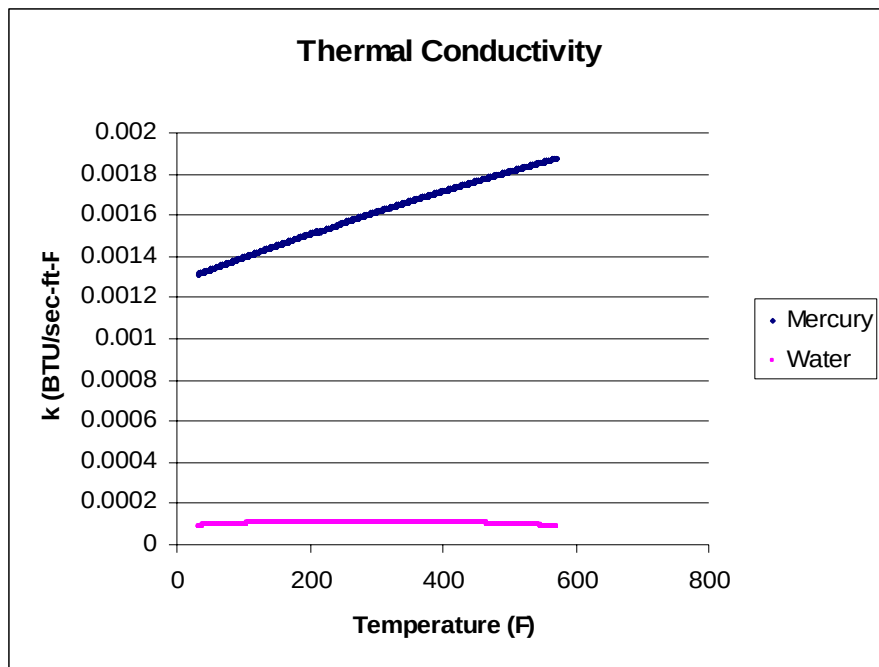


Figure 16 – Thermal Conductivity (Water at 100 Bar, Mercury at 1 Bar)

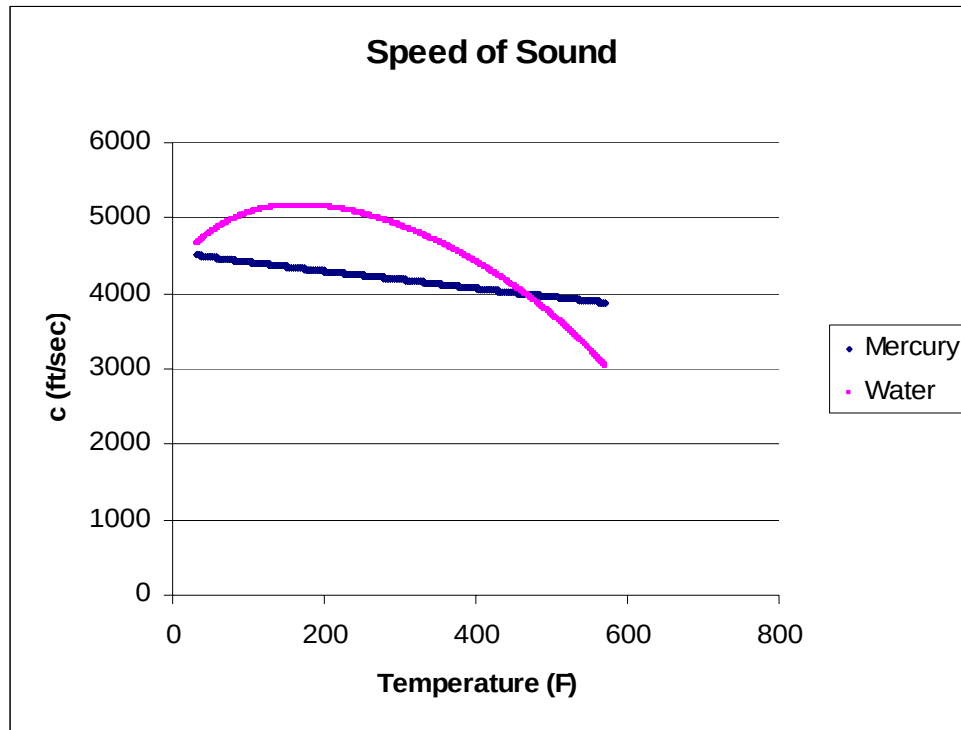


Figure 17 - Speed of Sound (Water at 100 Bar, Mercury at 1 Bar)

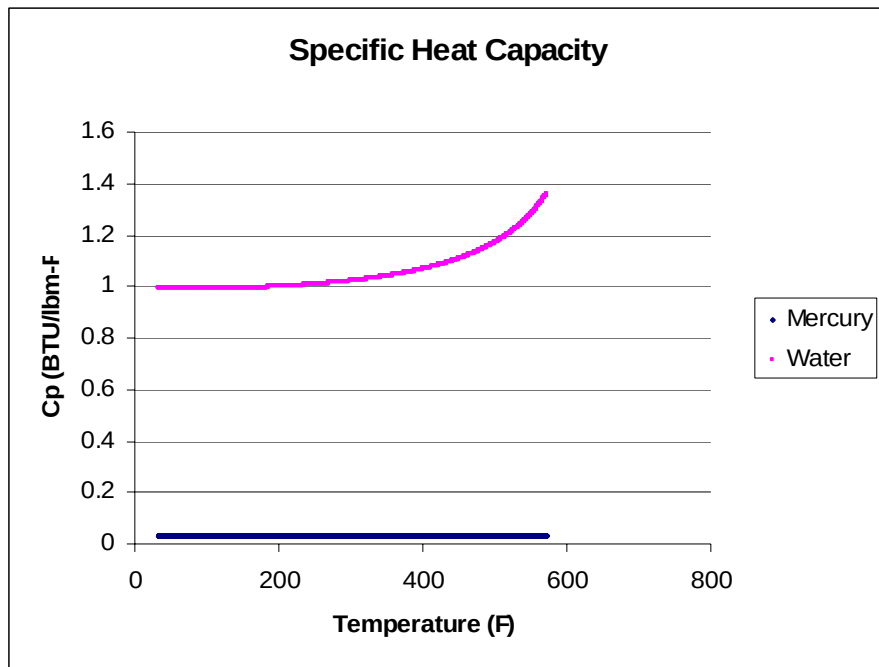


Figure 18 - Specific Heat Capacity (Water at 100 Bar, Mercury at 1 Bar)

In **Error! Reference source not found.**, the viscosity of water appears to be far more dependent on temperature than mercury. As the water temperature approaches the freezing point, the water viscosity exhibits asymptotically increasing behavior. When both liquids are sufficiently beyond their freezing points, they exhibit similar trends in viscosity. Since viscosity plays a key role in the Reynolds number – this temperature sensitivity will require particular attention.

Thermal diffusivity, shown in **Error! Reference source not found.**, is much greater for mercury than for water. The greater thermal diffusivity indicates that thermal energy is transferred by diffusion through mercury more efficiently than through water. The relatively strong diffusivity of mercury helps minimize thermal gradients within the fluid, lowering the probability of thermodynamic cavitation.

The Prandtl number is useful for comparing viscous effects to thermal effects in flow. Shown in **Error! Reference source not found.**, the very low Prandtl number values for mercury suggest that mercury is far more efficient at thermal diffusion than momentum diffusion; this correlates well with the high density and high thermal conductivity of mercury. This strongly suggests that, for mercury, Hydraulic Performance Loss (Grist, 1999), is more likely to be a concern than thermal-based cavitation.

The turbulent Prandtl number accounts for both momentum transfer enhancement due to turbulence, and thermal conductivity enhancement due to turbulence. An assessment of turbulent Prandtl number values and models is provided in Appendix A.

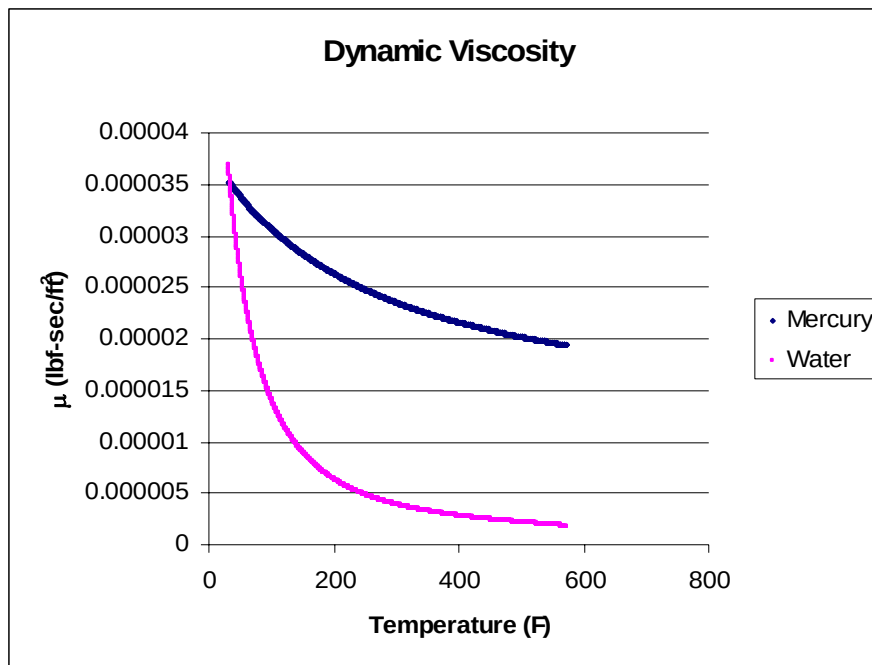


Figure 19 - Dynamic Viscosity (Water at 100 Bar, Mercury at 1 Bar)

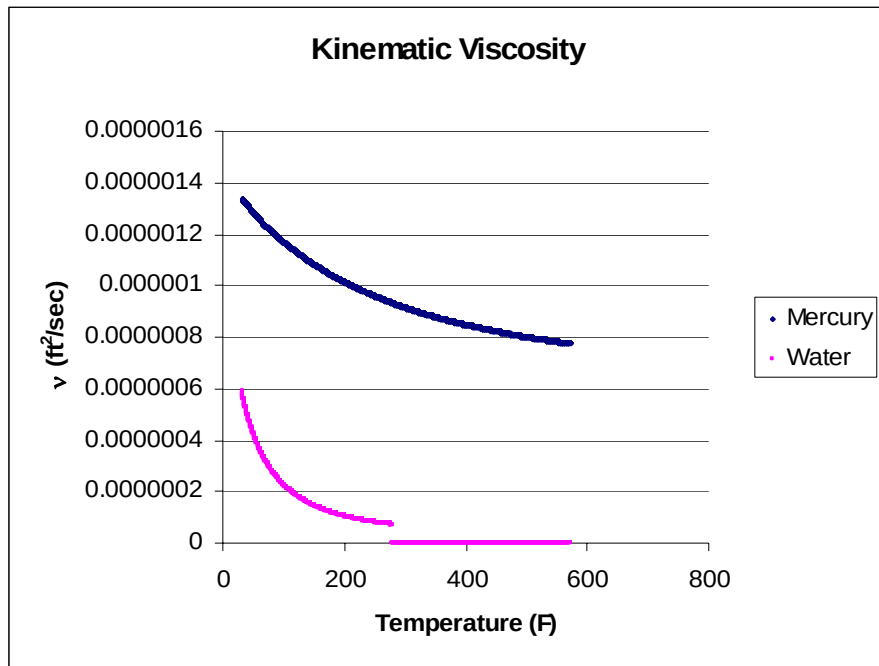


Figure 20 - Kinematic Viscosity (Water at 100 Bar, Mercury at 1 Bar)

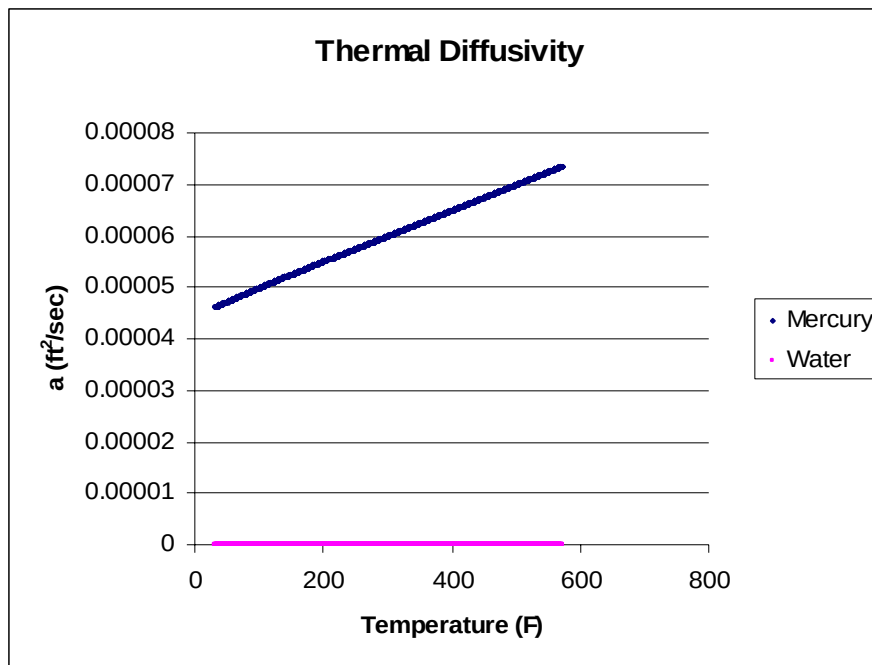


Figure 21 - Thermal Diffusivity (Water at 100 Bar, Mercury at 1 Bar)

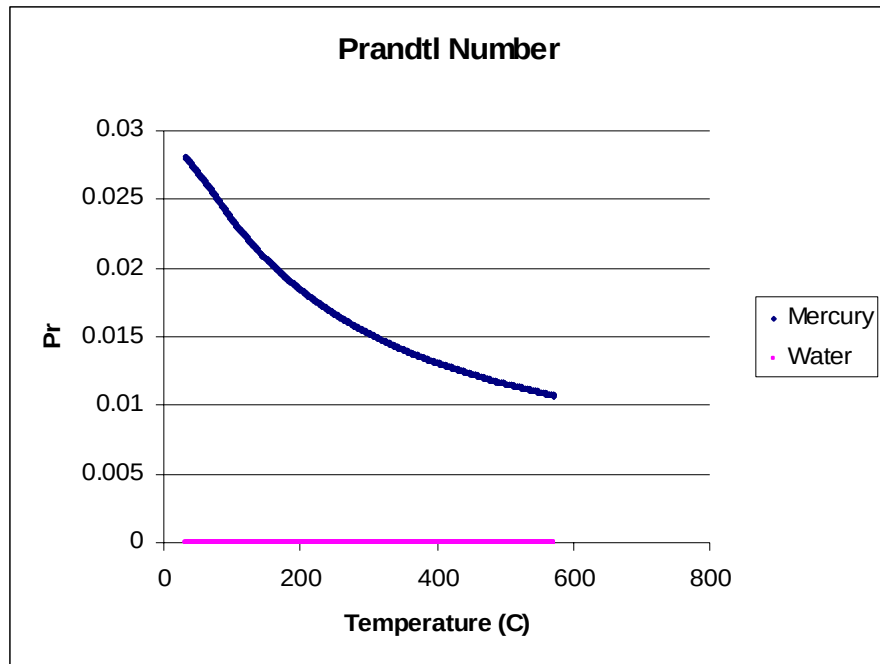


Figure 22 - Prandtl Number (Water at 100 Bar, Mercury at 1 Bar)

SNS PUMP NOISE EVALUATION

In vibrational frequency analysis, the vibrations transmitted through system components are recorded and evaluated to determine their sources. Common sources of “noise” in a pump include shaft rotation, impeller blades passing by the cutwater, flow turbulence (and cavitation, if present), bearing motion, and electrical signals from AC waveforms. A common practice is to evaluate the noise frequencies in *orders*. Rather than producing the frequency responses as functions of time, the recorded responses are plotted as multiples of the shaft rotation frequency. For example, if the shaft is rotating at 10 cycles per second (10 Hz), then the 10 Hz frequency is the first order, the 20 Hz frequency is the second order, and so on. Plotting the frequencies in orders readily allows the reader to determine which frequencies are present as a function of the shaft speed. Since many sources of vibration in a centrifugal pump produce frequencies that are multiples of the shaft rotation speed, a plot in orders provides a straightforward presentation of the frequencies that can be interpreted without intensive calculation.

A spectral analysis of the SNS pump noise was conducted previously by Benjamin Rothrock (2006). Here, the tests of the pump at SNS at 150 revolutions per minute are reviewed. This is decidedly slower than the design operating speed of nearly 400 rpm. For reference, this spectral signal was recorded by an accelerometer inboard of the shaft in a horizontal orientation.

Figure 23 (Rothrock, 2006) is a comparison of two operational times – July 10, 2006 and June 23, 2006. As noted on the chart, the July 10th test was performed at approximately 150 rpm. The June 23rd test was performed at the full speed of 400 rpm. In Figure 24 (Rothrock, 2006), the July 10th test is expanded for more convenient analysis.

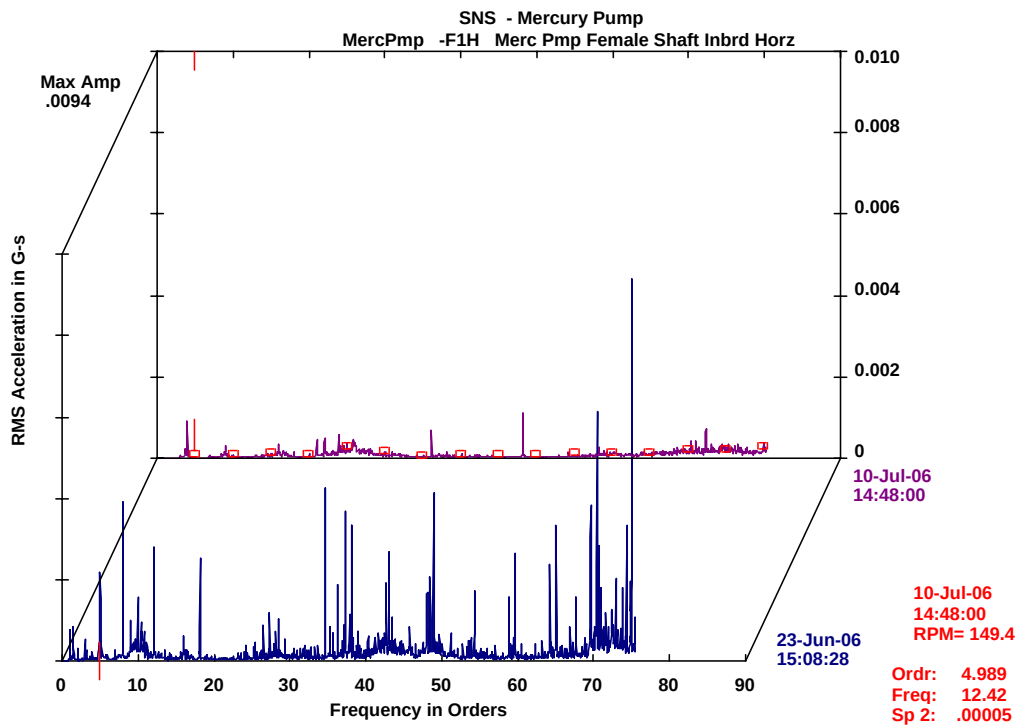


Figure 23 - Frequency Spectrum of the SNS Pump (Rothrock, 2006)

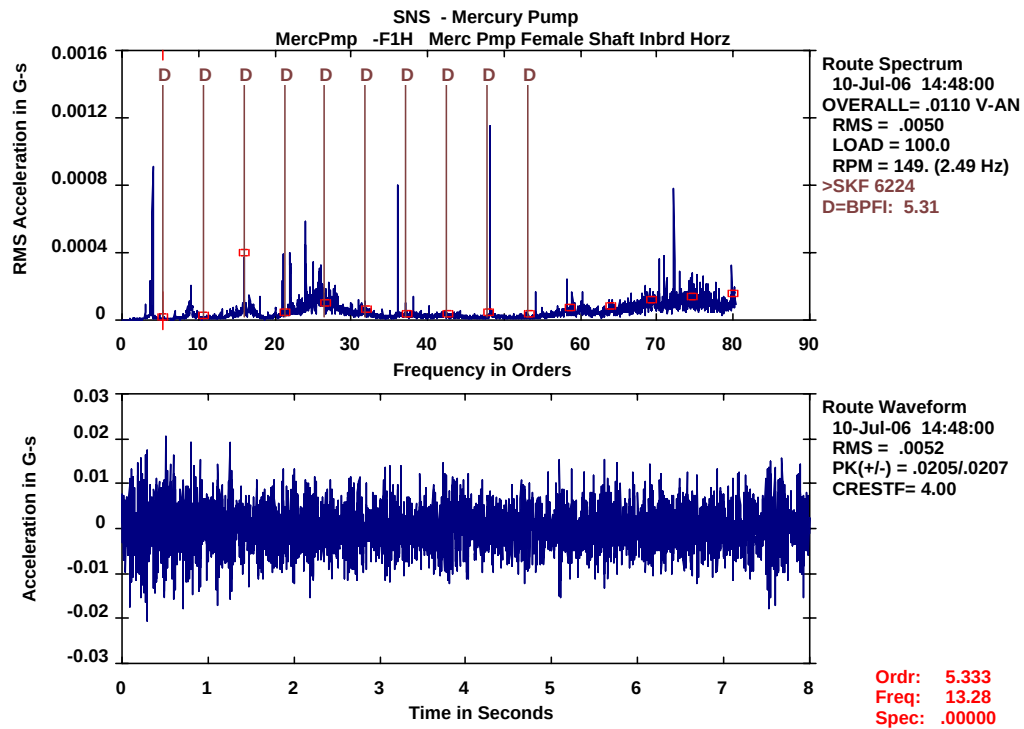


Figure 24 - 150 RPM Detail of SNS Pump (Rothrock, 2006)

The upper plot describes the frequency in orders, and the lower plot describes the frequency in time. The following is an analysis of the frequencies in the upper plot in Figure 24 (Rothrock, 2006).

Frequency Analysis

1. Shaft Rotation

The shaft rotation produces vibrations at the 1X order due to shaft misalignments and eccentricities. The virtual absence of any noise at the 1X order indicates that there is no difficulty with shaft alignment or eccentricity.

2. Vane Pass Frequency

The SNS pump is a five-vane impeller with a single volute design. Therefore, the vane pass frequency would appear at the 5X order. There is a substantial spike at the 4X order, but not at the 5X order. This indicates that the vane pass frequency is not significantly transmitting to the detector, and would ordinarily imply that the vane pass is not a significant concern.

One possibility remains, however for the vane pass frequency to cause problems in the shaft at frequencies other than 5X. If the vane pass frequency and the shaft frequency are harmonically related (that is, one frequency is a harmonic of the other), then it is possible for the vane pass frequency to cause vibrations at frequencies *other* than 5X. As an example, suppose the vane pass frequency happens to exist at five times the resonance frequency of the shaft at a particular speed:

$$f_{\text{vanepass}} = 5 * f_{\text{resonance,shaft}}$$

Then, the vane pass frequency would likely cause the shaft to vibrate at 1X. It would even be possible for the 1X frequency to build up to larger amplitudes than the 5X, which would then appear to be a shaft imbalance problem.

Another possibility is the excitation of bearing frequencies. The rollers in the bearings travel at frequencies that are multiples of the shaft speed. For example, a radial bearing may be listed as a 4X bearing. In this case, the rollers in the bearing travel at four times the shaft speed, and are sensitive to 4X vibrations and any harmonic interactions. There are possibilities that the vane pass frequency at 5X could cause sympathetic vibrations that build up a 4X vibration in the bearings. These possibilities are not extremely likely; however, a pump under gratuitous conditions and geometry may experience such effects. Due to the long shaft in the SNS pump, the bearings will be more susceptible to vibration damage. Further, the high mercury density increases variation in bearing load due to flow pulses caused by pump vane pass, which may reduce bearing life if not considered during pump design and bearing selection.

3. Shaft Misalignment

A shaft that is not properly aligned with its seals can create frequencies at any multiple of the 1X order. However, only the lowest few orders are likely, since energy transmission along high-order frequencies is difficult to sustain. A minor spike is observed at the 3X order, and a major spike is noted at the 4X order. The 4X order spike is normally indicative of alignment problems; since the pump is turning at the sub-optimal speed of 150 rpm in this plot, an alignment vibration is not unrealistic, even if the shaft is correctly aligned for full-speed operation.

At low speeds, the effect of shaft misalignment may sometimes be seen due to asymmetric loads from the fluid flow. As the fluid flows through the impeller from the inlet to the volute, the fluid will experience pressurization/depressurization cycles as the blades pass by the volute and the cutwater. At low speeds, the pressure forces in the fluid may have sufficient time to exert cyclical forces on the impeller, which are then transmitted to the shaft. These forces may appear to be an imbalance if present.

4. Turbulence

In Figure 24, the presence of many high-frequency vibrations is apparent. Most of this is attributable to turbulence, and is ordinary for pump operation. Due to the random nature of turbulence, the presence of turbulence noise is generally not problematic, as sustained excitation of particular frequencies is highly unlikely. Some of the larger spikes may be indicators of cavitation, as will be discussed next.

5. Cavitation

Cavitation, as noted earlier, produces random noise across the high-frequency end of the spectrum. The primary concern of cavitation in frequency analysis is to identify large-scale vibrations of high frequencies that might be due to cavitation. Since the cavitation frequency response occurs primarily through the violent implosion of the cavitation bubbles, cavitation noise is generally noted by the very strong frequencies generated in the high-frequency range.

In Figure 24, there are a few such spikes in the 20X range, one in the 30X range, one near 50X, and one above 70X. Looking at Figure 24 alone, the temptation is to identify these spikes as cavitation. However, when the magnitudes of these spikes are compared to 400 rpm operation in Figure

23, the scale is placed in better perspective. While these frequencies are identifiable, they do not necessarily represent cavitation. At this point, cavitation does not appear to be a significant problem at 150 rpm.

6. Electrical

Electrical noise is produced by the influence of the AC waveform on the mechanical operation of the pump motor. The source of the electrical noise is therefore the electrical input rather than the shaft rotation. So, electrical noise is dependent on the AC frequency, not the shaft rotation frequency.

To determine the extent of electrical noise in the signal, the electrical order must first be determined. For a shaft operating at 150 rpm, the shaft is turning at about 2.5 revolutions per second, or 2.5 Hz. The electrical frequency is 60 Hz, which is approximately 24 times greater than the shaft speed, or 24X order. Inspecting Figure 24, we see that the most significant spike in the 20X-30X range occurs at 24X. This particular spike can be at least partly attributed to electrical noise from the pump motor. Conservatively speaking, this frequency is not reliable as an indicator of any other noise.

In most high-power pumps, including the SNS pump, the AC source is three-phase. The use of three-phase power has two effects on the electrical noise; first, the distribution of electrical power among three waveforms reduces the magnitude of the 60 Hz frequency. Second, the three 60 Hz waveforms are set at 120 phase intervals, which results in pulses at three times the 60 Hz, or 180 Hz. In the example above, this would cause electrical noise at 72X. It appears in Figure 24 that there may be noise at 72X that is attributable to electrical sources, but analysis of individual frequencies at such high orders comes with relatively low

certainty; there are simply too many possible sources – like harmonics or cavitation – of high-order vibrations to conclude that a spike at 72X is certainly from electrical noise.

Another confounding issue with electrical sources of noise is the drive of the motor itself. The motor drive contains many complexities, including rotor/stator interactions, electromagnetic interactions, and mechanical interactions (including torque pulses) that can cause additional vibrations in the system. Further, the behavior of these vibrations can have speed-dependence that is not known, and may be the driving source of significant vibrations in the shaft.

However, electrical noise can serve one useful purpose in this analysis; since electrical noise is often not a significant concern in an operating pump, its magnitude can be used as a quick estimate of the extent of concern of other vibration noises. At 150 rpm, this detector indicates that the electrical noise is one of the greater sources of noise in the system. If this is so, then other noise signals might not be above tolerance limits.

Frequency Noise Comparison of 150 RPM and 400 RPM

Figure 23 contains data from two operational points of the SNS pump; 150 rpm at July 10th, and 400 rpm at June 23rd. The plots are very different, with the 400 rpm data containing much larger vibration amplitudes. Since the plot is in orders, the spectrum can be compared directly; that is, the shaft rotational vibrations will exist at 1X in both plots, the vane pass frequency will exist at 5X in both plots, and so on. The only data point with a significant shift is the electrical signal, which occurs at 60 Hz in both plots. For the 400 rpm test, the electrical signal is located at approximately 9 Hz.

Inspecting the 400 rpm signal, we see increased vibration amplitudes at 1X, 2X, 3X, 5X, 8X, 9X (electrical), 10X, 12X, 18X, and across the entire spectrum above about 25X. Some increase is expected due to the faster, more energetic condition of operation at 400 rpm. The high-frequency noise is the portion of the signal that is most immediately a concern. Here, the spread of very large amplitudes of high-frequency vibrations is indicative of cavitation within the pump. At 400 rpm, the noise points very strongly to a significant cavitation problem. Also notable is the now-significant vane pass frequency at 5X, and a lesser vane pass frequency harmonic at 10X. This indicates that the process of the vane tips passing by the cutwater is generating significant vibrations in the system. The vibrations are caused by the rapid stressing and de-stressing of the fluid that becomes entrained within the space between the vane tip and the cutwater. This stress cycle in the fluid is a very likely source of the cavitation as well; as the liquid mercury is stressed by the passing blade, the local pressure may be reduced below vapor pressure. If so, then the subsequent collapse of the cavitation bubbles is a possible source of the cavitation noise in the system.

The other notable change from 150 rpm to 400 rpm is the diminishment of the 4X order signal. This indicates that the perceived alignment noise from slow-speed operation does not occur at full speed. During startup and shutdown sequences, we may then expect alignment vibrations from the shaft. Since the SNS pump is nominally designed to operate at full speed for extended durations, this is probably not a major issue.

Determining the likelihood of cavitation in the pump is significant; cavitation erosion damage may be possible at the vane tip and at the cutwater. Since the system fluid is Mercury, the possibility of cavitation presents greater concern than is normal in a water system; prior studies have indicated that erosion from Mercury cavitation occurs at roughly 10 to 13 times the rate of erosion from water cavitation under equivalent conditions (Tillner, 1993). In other words, the

rate of damage seems to scale roughly with the density of the fluid. If data is then available for water cavitation damage and the erosion rates of the vane tip and cutwater, then an estimate of the erosion rate due to Mercury can be made with reasonable certainty.

Significance of the Location of Detectors

The accelerometer analysis performed by B. Rothrock involved more than one accelerometer location. Multiple accelerometer locations are used to verify signals and to detect vibrations that may be relatively mute at particular locations (Pokrovskii, 1972). For example, Figure 25 and Figure 26 illustrate the vibrations recorded by an accelerometer located outboard the female shaft (outside the sump tank near the outboard bearings in **Error! Reference source not found.**) with the same horizontal orientation.

In Figure 25, the 150 rpm data is located in the rear plot and was recorded 5 minutes after the data in Figure 23 and Figure 24. The operating conditions were similar, and the slight difference in time was likely due to the limitation of signal recording. Comparing the plots, we see:

1. Shaft Rotation

The 1X frequency is virtually absent in both plots, indicating a lack of eccentricity to the shaft and impeller.

2. Vane Pass Frequency

The 5X frequency is similarly minimal in both plots (in Figure 26, the marker is located at 4X rather than 5x). This also indicates that the vane pass is not a significant source of vibration at 150 rpm. However, Figure 25 shows a noticeable increase in the vane pass frequency.

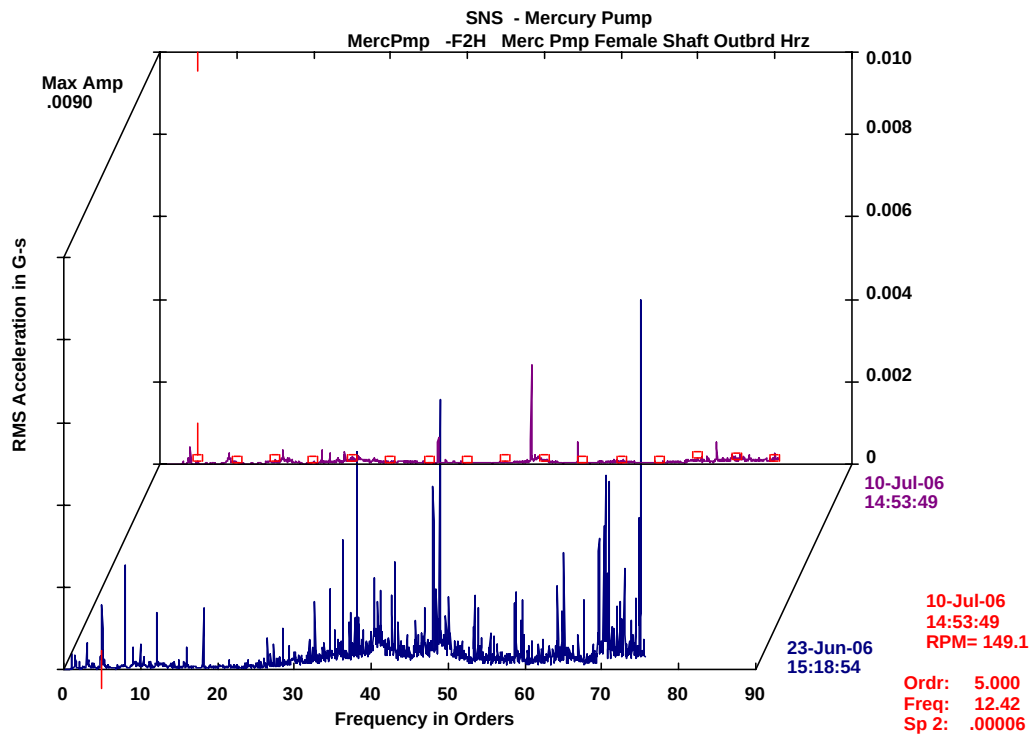


Figure 25 - Frequency Spectrum of the SNS Pump – Outboard Accelerometer (Rothrock, 2006)

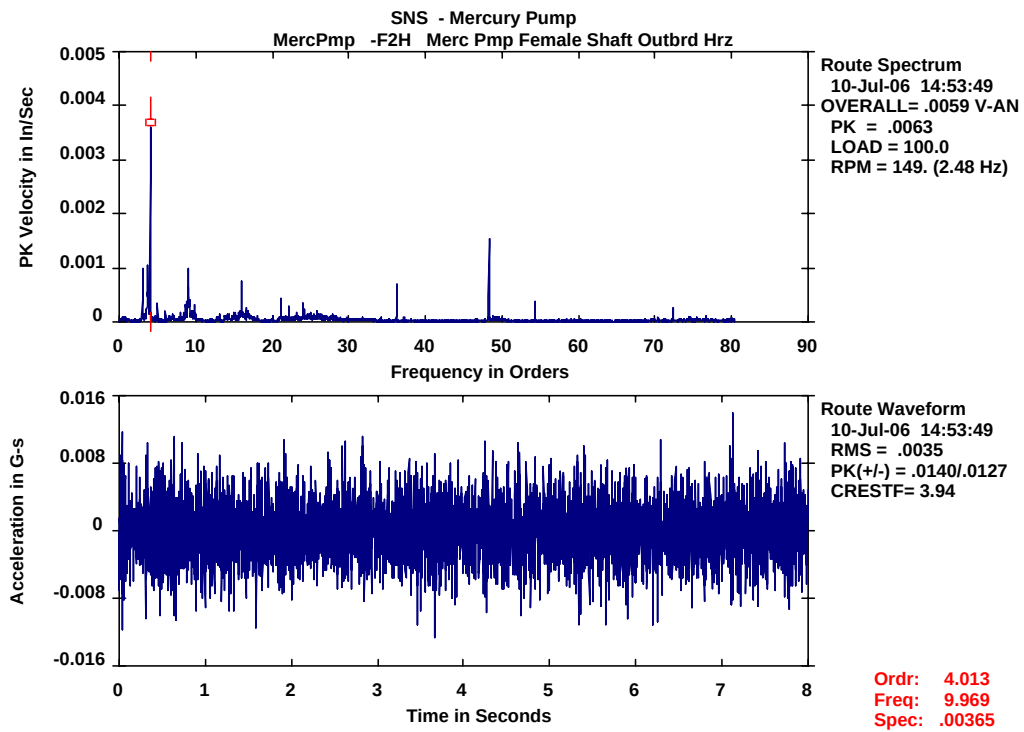


Figure 26 - 150 RPM Detail – Outboard Accelerometer (Rothrock, 2006)

This agrees with the inboard accelerometer, and supports the theory of vane pass frequency as a significant source of vibration at high speeds.

3. Shaft Misalignment

The 3X and 4X frequencies show relatively significant noise in Figure 26, particularly the 4X frequency. This is most likely due to the aforementioned alignment problem at slow speeds.

4. Turbulence

Here, the outboard detector records less noise among the high-order frequencies, indicating a lack of sensitivity to the turbulence in comparison to the inboard detector. The difference in high-order frequency response is the greatest difference between the two signals. The inboard detector will be more useful at detecting cavitation.

5. Cavitation

The 150 rpm signal of the outboard location does not indicate cavitation, as is evidenced in the turbulence analysis. However, in Figure 24, there is a very clear increase in the high-range frequencies when the pump is operated at 400 rpm. This increase overwhelms the increases in vibrations at all other orders. Since the outboard accelerometer is perceived to be less sensitive to high frequencies, this increase must be regarded as significant. Along with the verified increase in the vane pass frequency, the pump is most likely experiencing significant cavitation as the vane tips pass the cutwater when running at full speed.

At 400 rpm, the outer edge of the blade passes by the cutwater at approximately 8 m/sec. The change in dynamic pressure in the fluid based on this event can be approximated:

$$\Delta P_{dynamic} \cong \frac{1}{2} \rho v^2 = \frac{1}{2} \left(13,600 \frac{kg}{m^3} \right) \left(8 \frac{m}{sec} \right)^2$$

$$\Delta P_{dynamic} \cong 4.35 * 10^5 Pa = 63.1 psi$$

Equation 30 - Change in Dynamic Pressure at Cutwater

So, the vane pass may account for a localized pressure drop of about 60 psi when the pump is operating at full speed. This significant drop in pressure significantly increases the possibility of cavitation.

6. Electrical

The electrical signal at the 24X and 72X orders are visible in Figure 26, but not significant. This simply indicates that the outboard location may be better isolated from the electrical AC waveform than the inboard location.

As seen by Figure 25 and Figure 26, the location of the outboard accelerometers on the equipment will affect their ability to detect certain vibrations. For example, the most sensitive location to place an accelerometer to detect the vane pass frequency is on the pump outlet pipe near the volute. Likewise, accelerometers will be most sensitive to shaft rotation vibrations near the radial bearings, where the forces of the vibrations are transmitted to the structural materials.

Effect of the Orientation of the Accelerometers

The orientation of the accelerometers (e.g. horizontal vs. vertical) can also affect their ability to detect sounds. Here, Figure 27 and Figure 28 show the recordings of vertical accelerometers inboard, and Figure 29 and Figure 30 show the recordings of vertical accelerometers outboard.

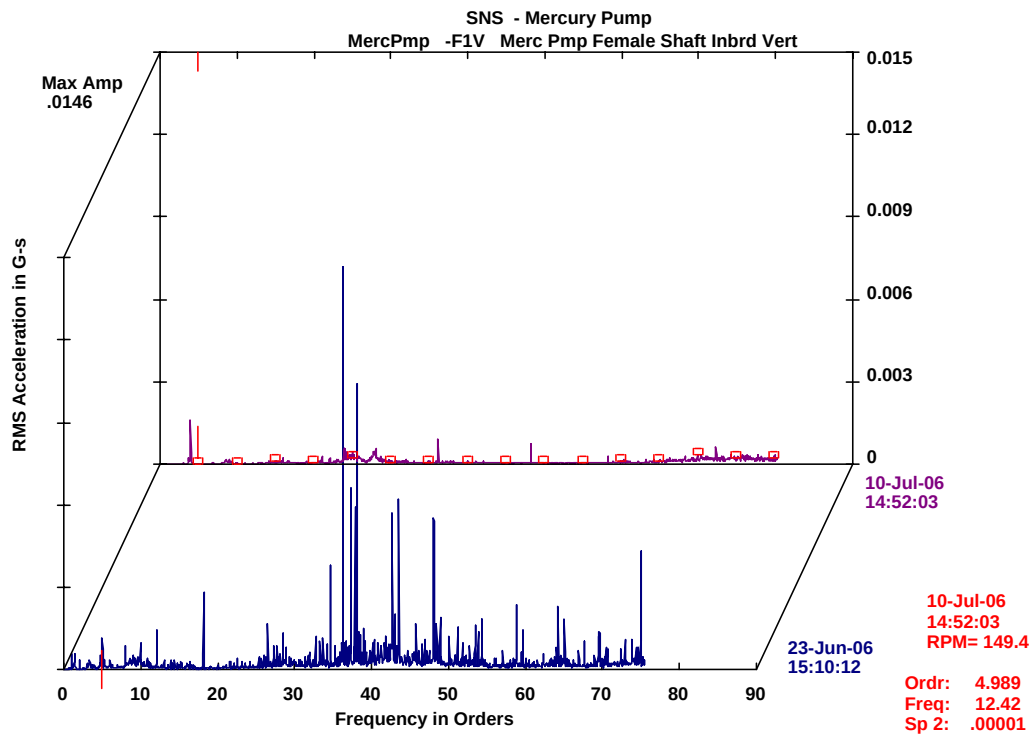


Figure 27 - Inboard Vertical Accelerometer (Rothrock, 2006)

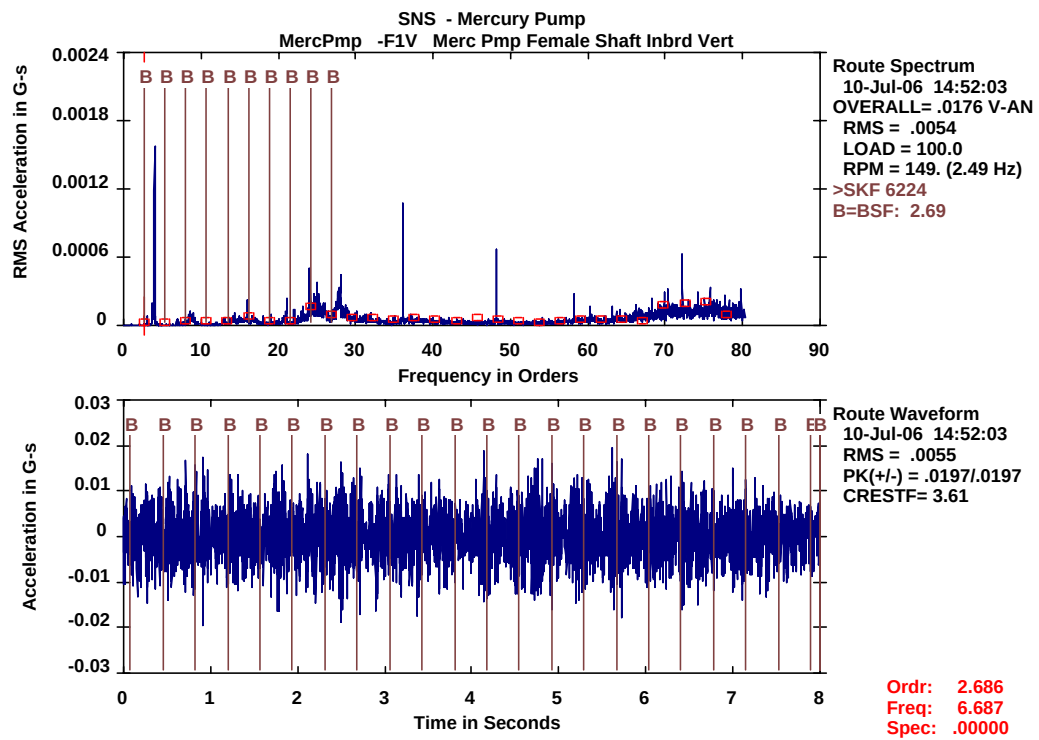


Figure 28 - 150 RPM Detail Inboard Vertical Accelerometer (Rothrock, 2006)

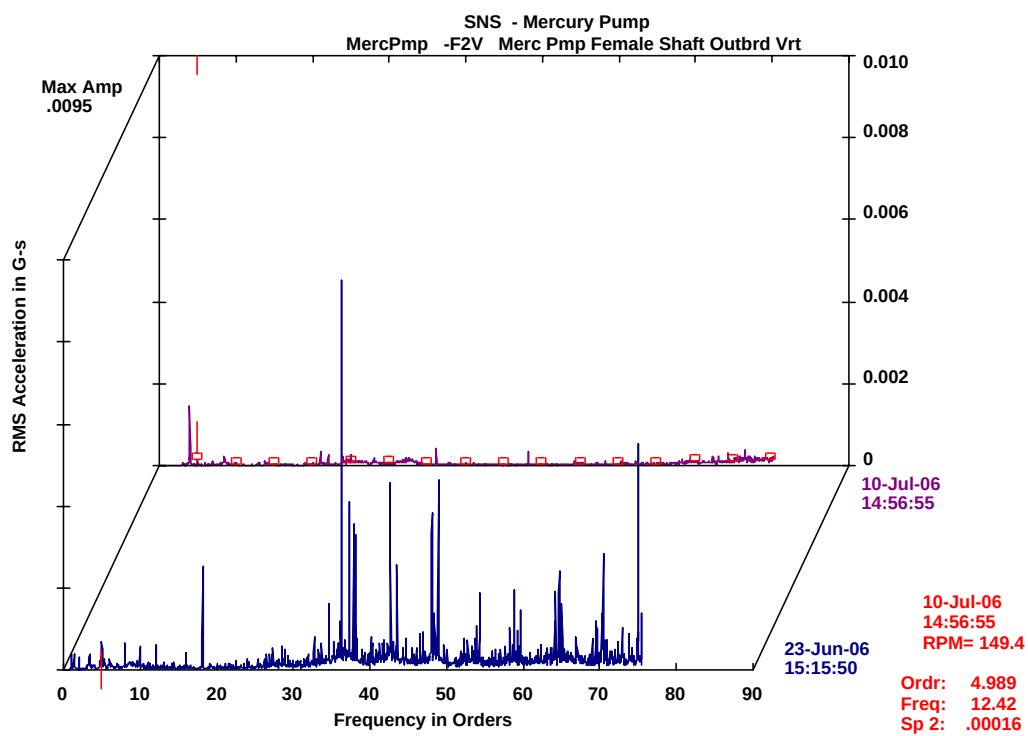


Figure 29 - Outboard Vertical Accelerometer (Rothrock, 2006)

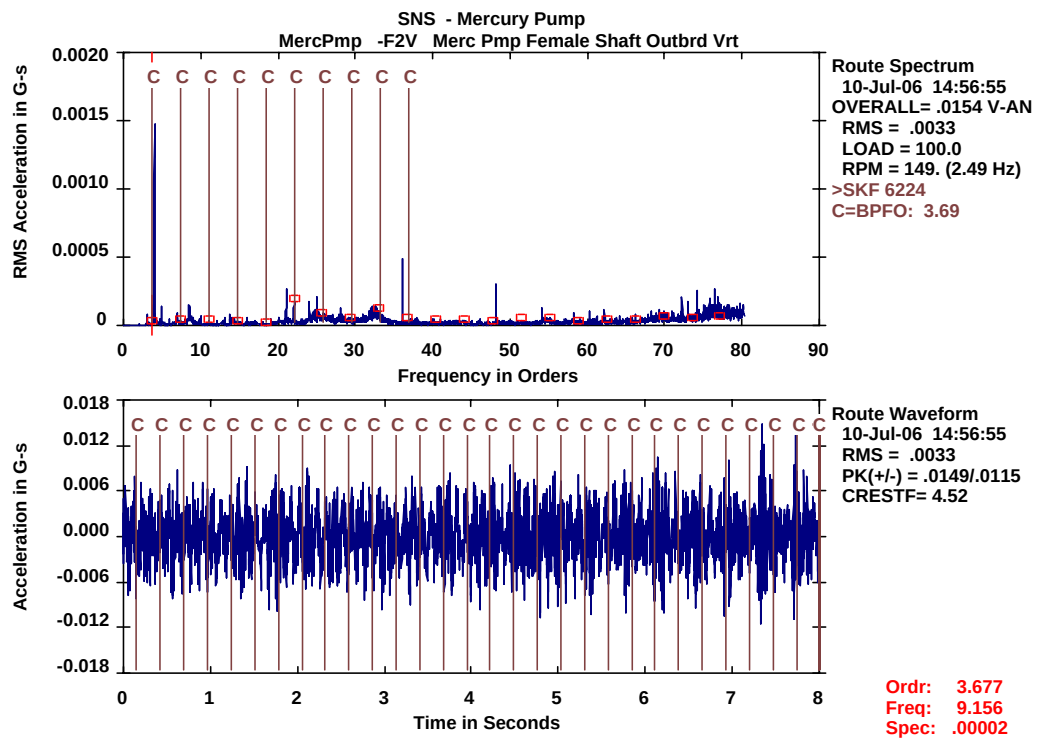


Figure 30 - RPM Detail – Outboard Vertical Accelerometer (Rothrock, 2006)

The vertical accelerometers display some variations in the signal, particularly in the upper frequencies. However, they display the same basic information that is found in the horizontal accelerometers; the 4X frequency is very significant, indicating a likely problem of bearing wear at 150 rpm operation. Likewise, the 5X frequency and the cavitation frequency range become significant at full speed operation, indicating possible cavitation due to the vane pass frequency.

Possible Remedies for the Observed Vibrations

The acoustic signals of the SNS pump indicate two general potential problems: bearing wear and cavitation. As noted by B. Rothrock, the bearing wear is most likely due to bearing over lubrication. This diagnosis is supported by the elevated bearing temperatures observed during operation. A re-evaluation of the bearing lubrication schedule is normally recommended as the first corrective course of action.

The cavitation problem is a significant concern, particularly as it occurs at the desired operational speed of nearly 400 rpm. The prime suspect is the overstressing of the mercury that becomes “trapped” between the vane tip and the cutwater. As the vane tip pass by the cutwater, this fluid must rapidly accelerate out of the way and may be vaporizing in the process. Without completely redesigning the impeller and volute, the only two viable solutions are to either slow down the pump or to increase the clearance between the impeller and the cutwater.

CONCLUDING REMARKS

In the SNS pump, evidence exists that cavitation may be a limiting factor of operation. The vibration analysis indicates noise that is commonly associated with flow cavitation when the pump is operated at 400 rpm. Flow cavitation at design speed may increase the rate of impeller wear and pump failure. The increased degradation of pump components due to cavitation may limit the time of operation of the mercury flow loop, as the pump may require an increased maintenance to overcome the wear.

The problem of cavitation in liquid metal flow is not unique to the SNS pump. In the nuclear power industry alone, the current interest in liquid metal reactor technology will likely result in an increase in the number of liquid metal flow loops used for power generation. As the use of liquid metals increases, an increased understanding of flow limitations like cavitation inception will be required for these liquids. Some research has shown that the known cavitation parameters of centrifugal pumps in water may extend to other liquids, such as mercury or liquid sodium (Grindell, 1957). However, the data in this field is insufficient to generate reliable conclusions, and additional testing must be performed to validate theoretical models of liquid metal cavitation.

Several options exist to address the concern of cavitation within a centrifugal pump. Alternative impeller designs, such as double volute impeller or multistage impellers may be able to provide the necessary pump performance without inducing cavitation within the flow. Other pump designs, such as permanent magnet pumps (P-M pumps), may be able to circumvent impeller cavitation since there is no impeller. As the understanding of pumping liquid metals increases, the viability of these options will be better understood.

BIBLIOGRAPHY

1. “Smithsonian Physical Tables.” Prepared by Forsythe, W., Knovel, Norwich, New York, 2003.
2. Blevins, R. D., “Flow-Induced Vibration.” Van Nostrand Reinhold Company, 1977.
3. Brennen, C., “Cavitation and Bubble Dynamics.” Oxford University Press, 1995.
4. Churchill, S., “A Reinterpretation of the Turbulent Prandtl Number.” Ind. Eng. Chem. Res. 2002, Vol. 41 pp. 6393-6401.
5. Cords, H., “A Literature Survey on Fluid Flow Data for Mercury – Constitutive Equation.” Forschungszentrum Julich GmbH, Julich, Germany, December 1998.
6. Daris, T., Bézard, H., “Analysis of Adverse Pressure Gradient Thermal Turbulent Boundary Layers and Consequence on Turbulence Modeling.” IUTAM Symposium on One Hundred Years of Boundary Layer Research, pp. 395-404. Springer, 2006.
7. Dong, Y. H., Ly, X. Y., Zhuang, L. X., “An Investigation of the Prandtl Number Effect on Turbulent Heat Transfer in Channel Flows by Large Eddy Simulation.” Acta Mechanica Vol. 159, pp. 39-51, Springer-Verlag 2002.
8. Gol'dshtik, M. A., Kutateladze, S. S., Lifshits, A. M., “Determination of the Turbulent Prandtl Number in an Asymptotic Boundary Layer with Suction.” Translated from Izvestiya Akademii Nauk SSR, Mekhanika Zhidkosti i Gaza, No. 1, pp. 74-79. Novosibirsk 1981.
9. Grindell, A., “Correlation of Centrifugal Pump Cavitation Inception Data for Water and Elevated Temperature Sodium-Potassium Alloy.” University of Tennessee, 1957.
10. Grist, E., “Cavitation and the Centrifugal Pump.” Taylor and Francis, 1999.
11. Holmgren, M., “Water and Steam Properties According to IAPWS IF-97.” XSteam Macros for Excel, www.x-eng.com., 1996.

12. Jischa, M., Rieke, H. B., "About the Prediction of Turbulent Prandtl and Schmidt Numbers from Modeled Transport Equation.", International Journal of Heat and Mass Transfer, Vol. 22, p. 1547, 1979.
13. Kays, W. M., "Turbulent Prandtl Number – Where Are We?", Journal of Heat Transfer, Vol. 116, p. 284, ASME 1994.
14. Leishear, R. A., and Stefanko, D. B., "Modifications to, and Vibration Analysis of, Tank 7 Slurry Pumps, F and H Tank Farms." Westinghouse Savannah River Company, Aiken, SC 2001.
15. Menzies, A., "New Measurements of the Vapor Pressure of Mercury." *Proceedings of the National Academy of Sciences of the United States of America*, Vol. 15, No. 12., pp. 558-562., 1919.
16. Mercer, C., "Frequency, Hertz, and Orders." Prosig Ltd., www.prosig.com 2007.
17. Pokrovskii, B. V., and Rubinov, V. Y., "Acoustic Determination of Cavitation Phenomena in Centrifugal Pump Elements." *Khimicheskoe i Neftyanoe Mashinostroenie*, USSR, July 1972 - translated 1973 by Plenum Publishing Corp., New York 1973.
18. Rothrock, B., *Report to SNS Group of 150 rpm Behavior of SNS Centrifugal Pump*, 2006.
19. Shah, Y. et. al., "Cavitation Reaction Engineering." Kluwer Academic / Plenum Publishers, 1999.
20. Tillner, W., "The Avoidance of Cavitation Damage." MEP, London, 1993.
21. Wnek, T. F., "Pressure Pulsations Generated by Centrifugal Pumps." Warren Pumps, Inc., www.warrenpumps.com, 2007.
22. Yahkot, V., Orszag, S. A., Yahkot, A., "Heat Transfer in Turbulent Fields.", International Journal of Heat and Mass Transfer, Vol. 30, p. 15, 1987
23. Zhaoshun, Z., Guixiang, C., Chunxiao, X., "Modern Turbulence and New Challenges." *Acta Mechanica Sinica (English Series)*, Vol. 18, No. 4. Allerton Press, Inc., New York, USA 2002.

24. Scheper, t. et. al., "Process Integration in Biochemical Engineering."
Advances in Biochemical Engineering / Biotechnology, Vol. 80. Springer-Verlag, Berlin, Germany 2003
25. "Summary Report of Target Test Facility R&D." SNS Doc. – 101010000-TR0001-R00, Sep. 2001.

APPENDIX

Appendix A :

Literature Review Of Turbulent Prandtl Numbers

Overview

The turbulent Prandtl number is an extension of the concept of the Prandtl number:

$$\text{Pr} = \frac{\mu C_p}{k} = \frac{\nu}{\alpha}$$

Equation A 1 - Prandtl Number

Where the Prandtl number compares diffusion of momentum and diffusion of thermal energy for a static or laminar system, the turbulent Prandtl number attempts the same ratio for a turbulent fluid flow. For example, mercury's Prandtl number of about 0.01 to 0.02 suggests that thermal energy diffuses primarily by conduction rather than convection. The turbulent Prandtl number is, in concept, analogous to the standard Prandtl, only the influence of turbulent eddies is included.

To date, there is no convenient relationship to define the turbulent Prandtl number. Mathematically, it can be written in the same form as the Prandtl number (Gol'dshtik, 1981; Daris, 2006):

$$\text{Pr}_t = \frac{\nu_t}{\alpha_t}$$

Equation A 2 - Turbulent Prandtl Number

However, the problem lies in adequately defining the turbulent kinematic viscosity and turbulent thermal diffusivity. Rather than attempting to solve the problem directly, most research on the turbulent Prandtl number has been cast toward either relating the turbulent Prandtl number to boundary layers (or mixing lengths in the case of free flows) or defining the turbulent Prandtl number through simulations and modeling.

Early Treatment of Turbulent Prandtl Numbers

Because of the difficulty in accurately calculating the turbulent Prandtl number, many early uses of the turbulent Prandtl concept simply assumed that the dynamic and thermal effects of turbulence were roughly equal, resulting in $Pr_t = 1$ (Gol'dshtik, 1981). Studies have indicated this to be a grossly inaccurate oversimplification for many flow conditions, as increased turbulence tends to strengthen the viscous (convective) effects with respect to conductive heat transfer.

Another approximation of the turbulent Prandtl number was suggested by Jischa (Jischa, 1979). This correlation is offered for fully developed turbulent flow and not for near-wall boundary layers.

$$Pr_t = 0.85 + \frac{0.015}{Pr}$$

Equation A 3 - Jischa Approximation

This correlation is noteworthy for its convenience and for its somewhat surprisingly good correlation for fluids with Prandtl numbers above about 0.7 (Churchill, 2002). A notable observation is the limiting value of $Pr_t = 0.85$ for high Prandtl number fluids. However, this bears little value in the analysis of liquid metal flows, which typically have very low Prandtl numbers.

Zhaoshun notes that the use of a constant value for the turbulent Prandtl number is tolerable for large Prandtl numbers, but not so for fluids with Prandtl numbers less than one (Zhaoshun, 2002). The reason for this is that increased turbulence causes the convective mode of transfer to gain importance relative to the conductive mode of transfer. In the case of liquid metal flow (e.g. mercury) there seems little reason to believe that a single turbulent Prandtl number can be defined for the fluid. Instead, Pr_t must be recalculated for every new flow condition.

Separation of Turbulent and Static Effects

One novel approach to the problem of turbulence and the Prandtl number was to treat the static and turbulent effects on the Prandtl number as independent and additive (Churchill, 2002). In his paper, Churchill defines a *total* Prandtl number, Pr_T , which is the sum of the Prandtl number, Pr , and a turbulent Prandtl number Pr_t . Here the use of the term “turbulent Prandtl number” is unique; it refers only to the *change* in the Prandtl number due to turbulence. That is, in Churchill’s approach, the total Prandtl number correlates to the turbulent Prandtl number of other approaches.

The final relationship that Churchill proposed for the total Prandtl number was:

$$\frac{1}{Pr_T} = \frac{(\overline{u'v'})^{++}}{Pr_t} + \frac{1 - (\overline{u'v'})^{++}}{Pr}$$

Equation A 4 - Churchill Approximation

The term $(\overline{u'v'})^{++}$ is defined as a dimensionless parameter relating the relative amount of shear stress attributable to variations in local velocity. The terms u' and v' represent the turbulent components of the u and v velocity

components. For example, the velocity term u may be seen as the sum of its average and time-varying components, $u = \bar{u} + u'$, and u' represents the deviation of the u velocity term from its mean. Churchill uses the relationships in Equation A 5 to capture the turbulent and total Prandtl numbers.

$$\begin{aligned} \overline{(u'v')}^{++} &= -\frac{\overline{\rho u'v'}}{\tau_w} \\ j &= -k \left[1 + \left(\frac{\text{Pr}}{\text{Pr}_t} \right) \left(\frac{\mu_t}{\mu} \right) \left(\frac{\partial T}{\partial y} \right) \right] \end{aligned}$$

Equation A 5 - Relationships in the Churchill Approximation

Here, u' and v' are the time-varying components x and y direction velocities respectively. The heat flux density is j . With these relationships and a thorough understanding of the flow properties, a reasonable value for the total Prandtl number can be determined.

Equation A 4 also provides insight by analyzing the extreme flow cases. In stagnant flow, the total Prandtls reduces to equal the Prandtl number, while for highly turbulent flow, the turbulent Prandtl term dominates the contribution to the total Prandtl number. This separation of turbulent and static effects is not unusual in fluid mechanics; the same approach is often used in energy and momentum balance equations to define quantities like shear stress and heat transfer.

Turbulent Prandtl in Numerical Modeling

In an age of easily accessible computing, accurate modeling of turbulent flow in numerical simulations is naturally of great interest. A successful model allows engineers to study the behavior of fluid flow without necessarily running physical experiments, and can be useful in testing new theories in fluid flow. Unfortunately, turbulent flow is very expensive to model in terms of

computational time and resources due to the sheer volume of information that must be accounted for. To simplify the problem of computational resource limits, several techniques have been developed to simplify the problem.

LES

Large Eddy Simulation (LES) is a numerical method of solving turbulent flow that focuses on modeling only the small-scale turbulence (Zhaoshun, 2002) and to solve the large eddies based on flow physics and geometry (Dong, 2002). This allows the use of a larger grid than conventional numerical simulations and significantly reduces the computational load on a computer. LES simulation, due to its reasonable balance of solution detail and computational demands, is one of the most popular methods of turbulent flow simulation today.

LES simulation has highlighted perhaps the greatest difficulty in accurately modeling turbulent flow – resolving the thermal and viscous boundary layers near flow boundaries (e.g. pipe walls). In order to accurately portray the rapid transitions in flow characteristics like Reynolds number, Prandtl number, and gradients like the temperature and velocity gradients, LES simulations require a very high density of mesh points in and around the boundary layers. These high concentrations of mesh points often cause the simulations to require an unacceptable amount of computational space and time.

One common workaround to the boundary layer problem is to provide averaging correlations in the boundary layers. For example, velocity and temperature profiles are normalized by known profiles for similar flows. These simplifications can provide good agreement between experimental data and calculated data, but can produce disagreement on turbulent Prandtl numbers (Dong, 2002).

The turbulent Prandtl number can also be used to help optimize the LES simulation. Dong notes that large Prandtl numbers require greater mesh densities in and around the boundary layers to model the convective effects properly. By understanding the turbulent Prandtl number prior to numerical simulation, the model can be streamlined for adequate efficiency and accuracy.

Recent Developments in Analytical Forms of the Turbulent Prandtl Number

Many attempts to provide a simpler model of the turbulent Prandtl number have been made. A few of the more notable efforts are given here.

Yahkot

Yahkot approached the problem of turbulent Prandtl numbers by applying Renormalization Group Theory (RGT) instead of the conservation equations. He proposed the following correlation (Yahkot, 1987):

$$\left(\frac{\frac{1}{\text{Pr}_T} - \alpha}{\frac{1}{\text{Pr}} - \alpha} \right)^{\frac{1+\alpha}{1+2\alpha}} \left(\frac{\frac{1}{\text{Pr}_T} + 1 + \alpha}{\frac{1}{\text{Pr}} + 1 + \alpha} \right)^{\frac{\alpha}{1+2\alpha}} = \frac{\mu}{\mu + \mu_t}$$

Equation A 6 - Yahkot Approximation

In this correlation, α is defined as:

$$\alpha = \frac{\left(\left[1 + 8 \left(1 + \frac{2}{d} \right) \right]^{\frac{1}{2}} - 1 \right)}{2}$$

Equation A 7 - Definition of α in Yahkot Approximation

The quantity “d” in this correlation is a topic of some discussion in the literature. Yahkot proposed a value of d=7 with a possibility that d=3 might be better. Churchill suggested that d=8 appears to better fit experimental data (Churchill, 2002).

Kays

Kays (Kays, 1994) proposed a simpler correlation for the turbulent Prandtl number:

$$\text{Pr}_t = 0.85 + \frac{0.7}{\text{Pr}} \left(\frac{1 - (\overline{u'v'})^{++}}{(\overline{u'v'})^{++}} \right)$$

Equation A 8 - Kays Approximation

This correlation bears resemblance to Jischa (Eq. 15) in the use of an additive term of 0.85. The difference between the two correlations is the use of the local turbulent shear stress as a modifier to the Prandtl number. Kays suggested that, for liquid metals, the coefficient of 0.7 should be replaced by 2.

Summary

The issue of solving turbulent Prandtl numbers in fluid flow is not yet resolved. In particular, further refinement of numerical models and increased computational power of computers may bring new insights on turbulent behavior.

Still, much effort has been expended on mathematically modeling turbulent Prandtl numbers, and much has been accomplished.

A quick and rough estimate of Pr_t can be determined by Jischa's correlation (Equation A 3). This provides the benefit of a ballpark estimate of Pr_t without having to consider the actual flow conditions of a problem. This method is not useful for liquid metals and other cases where $Pr \ll 1$.

Other methods of calculating Pr_t include Yahkot (Equation A 6 and Equation A 7) and Kays (Equation A 8). Currently, the best marriage between convenience and accuracy appears to be the Kays correlation. Independent studies of the Kays correlation by Kays and by Churchill show good agreement with the correlation and average experimental data.

VITA

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