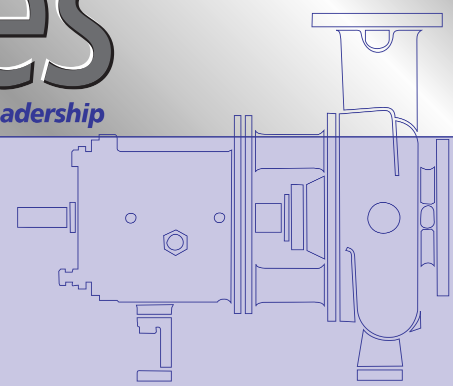
PumpLines
Innovation...Technology...Leadership
FALL 1999



Smart Technology for Pumps

Introducing PumpSmart™ The Future is NOW!

Jerry Connolly
Director, New Business Development
Goulds Pumps/ITT Industries

Over the past two years while doing research for a product that we were developing, we surveyed and interviewed dozens of customers regarding their vision, the future of industry, and what they will require from their vendor base in the future. Here's what you told us you wanted:

- Solutions, not just products.
- Increased versatility and functionality.
- Lower life cycle cost, not just initial cost.
- Quantitative measurement of results-reduced cost, higher efficiencies, increased productivity, payback, etc.

We've listened.

In November, in New York City, we will be introducing a new pumping system that we believe addresses these requirements, plus a few more. PumpSmart Process Systems, aggressively attacks all major portions of life cycle cost-initial capital cost, installation cost, operating cost and maintenance cost.

PumpSmart™ is a pumping system that utilizes a standard centrifugal pump in conjunction with a "smart controller" and ITT Industries proprietary Pump Control Software*. The software, which resides on the controller microprocessor chip, is the "brains" of the system, allowing the pump to monitor and REACT to any system condition, As a result, PumpSmart...

- Will not cavitate
- Will not run dry
- Will not run against closed suction or discharge valves
- Eliminates flowmeters, starters, and flow control valves
- Significantly lowers life cycle cost
- Significantly increases MTBF

Let's take a quick look at the four major areas of life cycle cost and see how PumpSmart addresses each.

Initial Capital Cost

Prior to shipment, PumpSmart is downloaded with all of the pump hydraulic information, the customer's specific control parameters (set points, alarms, shutoffs, trips, etc), and our Pump Control Software. The software allows the controller to continuously monitor both the system and pump conditions and will match the pump output to the exact system head required by the system. The automated flow control valve that was controlling the process is no longer required.

In addition, ITT Industries has developed a patented technique to measure flow internal to the pump casing. This eliminates the need for a flow measuring device, such as a mag meter. In addition, the PumpSmart controller has an integral starter, therefore, a separate starter is no longer required. PumpSmart also monitors and reacts to minimum flow conditions (slow down, shutoff, alarm, etc) so expensive recirculation lines and valves are not required.

Because PumpSmart integrates the functionality of this equipment, you can substantially decrease initial capital expenditures on a pumping system.

Installation Costs

By getting rid of excess equipment in your system, you are also decreasing maintenance costs.

- Flow control valve-piping and wiring (power and communications lines) costs are avoided. Costs to run and maintain plant air can also be avoided for pneumatic valves.

continued on page 5

*patent pending

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*View this issue of **PumpLines** on our website at www.goulds.com*

Goulds Pumps
ITT Industries
Engineered for life

New Products

IC Process Pumps
A New Generation of Process Pumps Which Minimize Life Cycle Costs

Stan Knecht
Product Line Manager
Vogel/ITT Industries

Goulds Pumps is proud to introduce its newest chemical process pump, the Vogel Model IC, ISO Pump (Figure 1). The IC pump is the first in a series of planned product developments which will expand and enhance ITT Industries portfolio of metric designed products to better meet the needs of our customers worldwide.

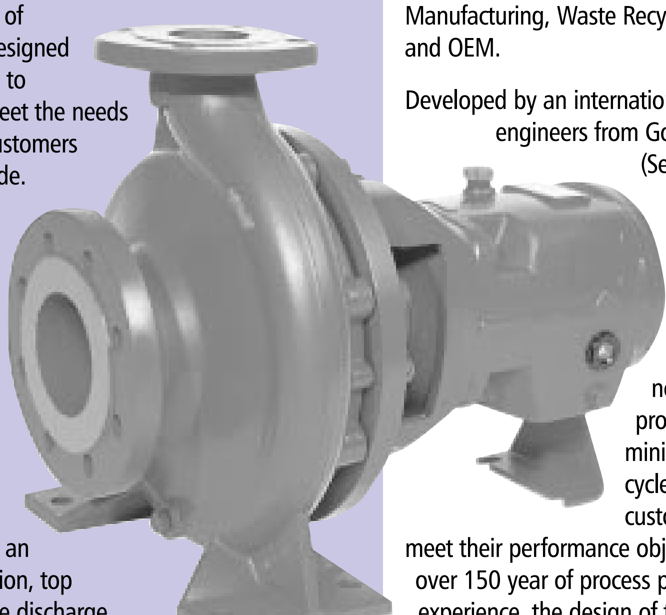


Figure 1

The IC is an end suction, top centerline discharge metal chemical process pump designed to meet or exceed the requirements of EN25199\ISO 5199 criteria for guaranteed reliability and installation flexibility. The Model IC features a back pull-out arrangement and utilizes a modular design to maximize component interchange-ability for decreased maintenance and inventory costs. As standard the Model IC pump is provided with flanges drilled in accordance with ISO 7005-2, PN16. Flange drilling in accordance with ANSI B16.5 Class 150 can also be provided upon request. Shaft sealing for the Model IC meets the dimensional standard of ISO 3069, supporting use of DIN 24960 L_1K mechanical seals, packing or cartridge seals.Spanning 30 different pump sizes, the IC series is capable of capacities to 450 m3/hr (1,980 gpm) and heads to 150m (492 ft). The design is suitable for operating temperature ranging from -40˚ C to 177˚ C (-40˚ F to 350˚ F) and for working pressures to 16 bar (235 psig). Pump constructions available include Ductile Iron, Stainless Steel, Duplex SS, and high alloys.

Designed as a Chemical Process pump, the Model IC is ideal for the pumping of corrosive solutions as well as solutions containing nominal solids typically found in Chemical, Petrochemical and Pharmaceutical industries. Yet the Model IC pump has the versatility to be applied to any process or utility application in industries such as Pulp & Paper, Food & Beverage, Water Treatment, Metal Manufacturing, Waste Recycling\Disposal and OEM.

Developed by an international team of engineers from Goulds Pumps (Seneca Falls, USA), Richter (Germany) and Vogel Pumpen (Austria), the design objective was to create a new generation process pump which minimized pumping life cycle costs to allow our customers to better meet their performance objectives. Leveraging over 150 year of process pumping experience, the design of the Model IC pump represents a collection of the "Best Available Technology" from all of ITT Industries, resulting in a new standard for process pumping performance and reliability.

World-Class Hydraulic Design

Based on decades of process pumping experience, thousands of ITT Industries' pump hydraulic designs were evaluated to provide the Model IC pump with hydraulic performance with maximum efficiency and stable operating performance. Featuring a key driven, enclosed impeller design, the Model IC offers benchmarked hydraulic performance while at the same time minimzing hydraulic loading for extended pump and seal life.

Patented Cyclone Seal Chamber

As I explained in my Cyclone Seal Chamber article in the July issue, critical to reducing pumping Life Cycle cost in process services is the need to improve reliability of the shaft seal. This can only be accomplished by providing the **best shaft sealing solution in the best sealing environment.** Based on years of experience in process industries and extensive development testing the IC pump has been fitted with the Cyclone Seal Chamber, a seal chamber proven to provide the optimum sealing environment for extended seal life.

The Cyclone Seal Chamber is fundamentally an oversized, taped bore seal chamber which features two helical grooves cast into the seal chamber bore engineered to improve shaft sealing reliability. The helical grooves are key to the design as they act to maintain flow patterns within the seal chamber which both cool and lubricated the mechanical seal while preventing solids and vapor from collecting in the seal environment.

The Cyclone Seal Chamber (Figure 2) has been rigorously tested and proven to maintain the optimum sealing environment even in the presence of solids and vapors for extended seal life and reduced cost. As an added benefit the Cyclone seal chamber can eliminate the need for auxiliary seal flush for liquids containing up to 10% solids by weight, further reducing installation cost and simplifying maintenance activities.

Proprietary Mechanical Seal Design

Complimenting the Cyclone Seal Chamber is a proprietary mechanical seal which has been designed for exclusive use with the IC pump to further optimize shaft sealing performance. This seal features a stationary spring design utilizing balanced seal faces and Hastelloy C springs. This arrangement extends seal life by eliminating shaft sleeve fretting, reducing seal face loading and by eliminating seal clogging and corrosion as spring are located external to the pumpage. This seal is available as a single seal in either conventional or cartridge design in addition to a double cartridge configuration.

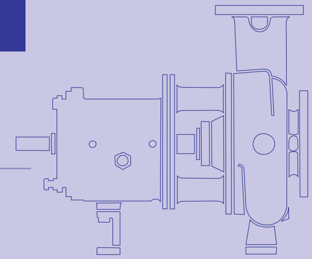
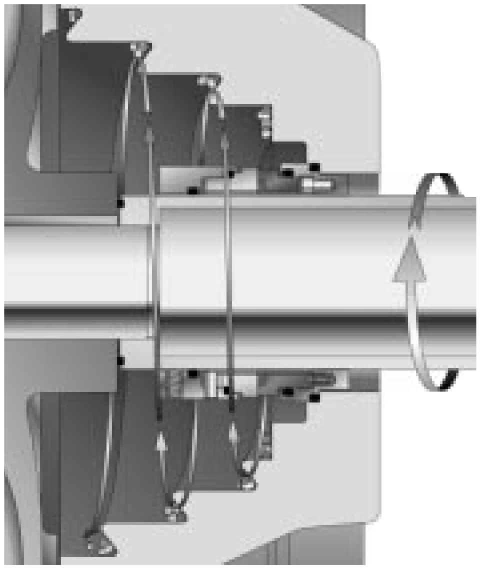


Figure 2



Heavy Duty Power End with Large Capacity Oil Sump

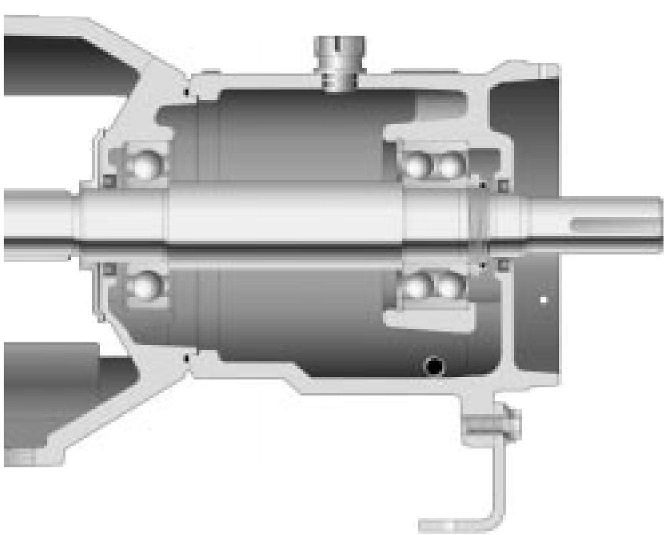
The power frame of the Model IC (Figure 3) pump provides a rigid and durable shaft support suitable for demanding applications. Heavy-duty double row, angular contact thrust bearings and single row radial ball bearings have been designed to provide L_{10} bearing life in excess of 17,500 hours. Standard supply for the Model IC pump is a rigid, stainless shaft designed for reliable, corrosion-resistant power transmission while limiting shaft deflection to less than 0.05mm.

The bearing frame for the Model IC pump has been designed with a large oil sump for it's class, 2 – 3 times larger than most competitor's design. The large oil sump maximizes radiating surface area for increased oil cooling. Cooler oil temperatures result in extended bearing life and reduced Life Cycle cost.

A host of engineered options are also available for the Model IC pump including Hydrovar™ pumping system controller, suctions inducers and heating jackets to provide our customers with engineered pumping solutions to meet the needs for their most difficult process pumping applications.

Please contact your local sales representative to find out more details about the New Model IC series, the latest innovation in ISO pumping process solutions.

Figure 3



New Products

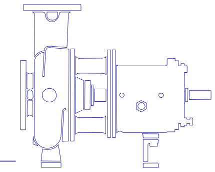
Vertical Product Line Expansion

Jim Johnston
Assistant Product Manager
Goulds Pumps/ITT Industries

With our Vertical Pump Operation's high degree of flexibility, our products continue to be attractive to customers in the heavy industrial market segments (i.e. Power, Hydrocarbon Processing, Mining, etc.). As plant sizes increase and the demand for more cost competitive products increases, we feel the need to further expand our product range. One area of potential involves a new hydraulic line of pumps with 6200 Ns (Specific Speed). This high flow, low head product would find attractive usage in large cooling towers or small to medium size circulating water applications. As our major competitors already have pumps in this Ns range, it is necessary for us to develop this product in order to offer competitive products. The markets we serve require VPO to offer the highest speed pump possible, which results in the most competitive price. The 6200 Ns will allow us to continue to do just that.

Initial development will be in a 20" model, the 20NMO. This model size would allow future line expansion up to a 60" pump. Hydraulic performance from this base model is highly predictable, which allows VPO to quote projects prior to completion of specific pattern equipment.

VPO's past success with new model development and product line expansion should continue with the 6200 Ns "N-Line" product. We have every confidence that performance expectations will meet or exceed our predictions. Since Goulds Pumps/ ITT Industries employs the best Sales and Distribution network in the world, it is only fitting that VPO offer the most competitive Vertical Pump Products in the world.



Cavitation in Centrifugal Pumps

Allan R. Budris
Director of Product Development
Goulds Pumps/ITT Industries

Thinking you have sufficient system suction pressure to a pump, just because you exceed the net positive suction head required by the pump curve, is no guarantee of acceptable pump performance. You may need a margin to suppress the cavitation that exists in a pump substantially above the published NPSHR value. Cavitation does exist above the NPSHR. Then again, the pump might run fine with almost no margin above the NPSHR of the pump. This article clarifies this situation, and is based on the new Hydraulic Institute Standard on NPSH Margin (ANSI/HI 9.6.1-1998). It provides a method for identifying the likelihood of a pump experiencing cavitation problems, by providing a simple method of calculating the "Suction Energy" of the pump, and recommending NPSH Margins for levels of suction energy.

The noise, the vibration and possibly the reliability of a centrifugal pump and mechanical seal may be significantly affected if an appropriate Net Positive Suction Head (NPSH) margin is not provided above the published Net Positive Suction Head Required (NPSHR) of the pump. The NPSHR Margin is defined as the NPSH Available at the pump inlet, minus the NPSH required by the pump. The NPSH Margin Ratio is the NPSHA divided by the NPSHR.

By Hydraulic Institute definition, the NPSHR of a pump is the NPSHA that will cause the total head to be reduced by 3%, due to flow blockage from cavitation vapor in the impeller inlet. NPSHR is by no means the point at which cavitation starts; that level is referred to as incipient cavitation. The NPSHA at incipient cavitation can be from 2 to 20 times the 3% NPSHR value, depending on pump design and suction energy level. It can take from 1.05 to 2.5 times the NPSHR value just to achieve the 100 percent head point (NPSH "Required"-0%).

Suction Energy

Due to the very high NPSH Margins required to completely suppress cavitation, we know that cavitation must exist in a high percentage of pump applications. However, we also know that acceptable life is achieved in most installations, despite this cavitation. So how can we predict when cavitation is likely to cause problems? The amount of energy in a pumped fluid which flashes into vapor and then collapses back to a liquid, in the high pressure areas of the impeller, determines the extent of the noise and/or damage from cavitation. Suction Energy is another term for the liquid momentum in the suction eye of a pump, which means that it is a function of the mass and velocity of the liquid in the inlet.

The following formulas, which are based on the "Suction Energy" graph presented in the Hydraulic Institute standard, can be used to approximate the Suction Energy in a pump:

Suction Energy (SE) = (De x n x S x s.g.)
- De = Impeller Eye Diameter (inches)
- n = Pump Speed (RPM)
- S = Suction Specific Speed RPM x (GPM).5/(NPSHR).75
- s.g. = Specific Gravity of Liquid

De = Suction Nozzle Diameter x 0.9 (is a good approximation for End Suction Pumps)
De = Suction Nozzle Diameter x 0.75 (is a good approximation for Side/Double Suction Pumps)

Suction Energy Levels

The Hydraulic Institute has divided Suction Energy into three Regions:

- LOW SUCTION ENERGY – NPSH Margin is not critical, except for the effect on the head generated by the pump at very low margins.
- HIGH SUCTION ENERGY – Pumps with low NPSH Margins, especially when operated in the suction recirculation flow range, may experience noise, vibration and/or minor cavitation erosion damage with impeller materials that have low cavitation resistance.
- VERY HIGH SUCTION ENERGY – Pumps with low NPSH Margins, especially when operated in the suction recirculation flow range, may experience erosion damage, even with cavitation resistant materials such as stainless steel.

The following suction energy milestones, from the Hydraulic Institute graph, and field experience gained by ITT Industries, approximate the values of High and Very High Suction Energy.

Start of High Suction Energy (De x n x S x s.g.)
- End Suction Pumps: (SE) = 160 x 10^6
- Horizontal Split Case Pumps/Radial Inlet: (SE) = 120 x 10^6

Start of Very High Suction Energy (De x n x S x s.g.)
- End Suction Pumps: (SE) = 240 x 10^6
- Horizontal Split Case Pumps/Radial Inlet: (SE) = 180 x 10^6

NPSH Margin Recommendations:

Table 1 summarizes the Hydraulic Institute minimum NPSH margin Ratio guidelines (NPSHA/NPSHR), which are applicable within the Allowable Operating Region of the pump.

Table 1
NPSH Margin Ratio Guidelines (NPSHA/NPSHR)

Suction Energy Level	NPSH Margin Ratio
Low	1.1 to 1.3
High	1.3 to 2.0
Very High	2.0 to 2.5

High and Very High Suction Energy pumps that operate with the minimum NPSH Margin values recommended in Table 1 will normally have what is considered "acceptable" seal and bearing life, (but not necessarily optimal). They may still be susceptible to elevated noise levels and erosive damage to the impeller. This can require more frequent impeller replacement than would otherwise be experienced, had the cavitation been totally eliminated.

It will typically take a NPSHA of 4 to 5 times the 3% NPSHR of the pump to totally eliminate cavitation. This ratio can reach 20 for Very High Suction Energy pumps, and a low of 2 for some pumps with Low Suction Energy levels.

Additional NPSH Margin may be needed to cover uncertainties in the NPSHA (available) to the pump or operating flow point. If a pump runs further out on the curve than expected (which is very common), the NPSHA of the system will be lower than expected,

New Products...

continued from page 1

and the NPSHR for the pump will be higher, thus giving a smaller (or possibly negative) actual NPSH Margin. All pumping systems must be designed to have a positive margin throughout the full range of operation.

Optimum pump performance also requires that proper suction/inlet piping practices are followed to ensure a steady uniform flow to the pump suction at the requred suction head. Poor suction piping can result in separation, swirl and turbulence at the pump inlet, which decreases the NPSHA to the pump and causes added cavitation.

NPSHA Margins of two to five feet are normally required (above those shown in Table 1) to account for these uncertainties in the actual NPSHR and NPSHA values. This added margin requirement could be even greater depending upon the severity of the conditions, especially if the pump is operating in suction recirculation. If the application is critical, a factory NPSH test should be requested.

Summary

In summary, the following key points should be understood about cavitation in a centrifugal pump, NPSH Margin requirements, and how they are affected by the Suction Energy level of the pump:

- Cavitation exists at and substantially above the NPSHR of a pump.
- The Suction Energy level of a pump (as installed in a system) determines if the cavitation that frequently exists in a pump will cause noise, vibration and/or damage to the pump.
- High Suction Energy pumps are likely to be noisy with higher vibration and will possibly experience less than optimum pump life, if sufficient NPSH Margin is not provided.
- High Suction Energy pumps are more susceptible to problems from poor suction inlet piping, especially if they also operate in suction recirculation.
- Very High Suction Energy pumps will be noisy, will have high vibration and are likely to experience reduced pump life if sufficient NPSH Margin is not provided. Very High Suction Energy pumps are very susceptible to problems from poor suction inlet piping.

The Future is NOW! Smart Technology for Pumps

(continued from page 1)

- Flowmeter-similar piping and wiring (power and communication lines) costs are avoided.
- Starter-The installation cost of the starter can be replaced by the cost to install PumpSmart.
- Recirculation line-costs associated with piping the line and installing valving is avoided.

Operating Costs

Since PumpSmart utilizes a unique variable frequency controller with our Pump Control Software, it will automatically match pump operation to the system head requirements. Energy consuming control valves are no longer required. Our most recent installation of PumpSmart is on a 100 horsepower cooling tower pump (Model 3196 XLT). The system was designed with two duplicate pumps and control valves. The pump running with the valve in operation is consuming 98 horsepower, while the PumpSmart System is consuming only 63 horsepower and is running over 300 RPM slower. At 0.60/kW-hr, this represents over <u>$12,000 in energy savings per year</u>. And because PumpSmart continuously calculates savings (see Figure 1), the running total, in dollars, will constantly be in view in the DCS control room or on the PumpSmart keypad.

Maintenance Costs

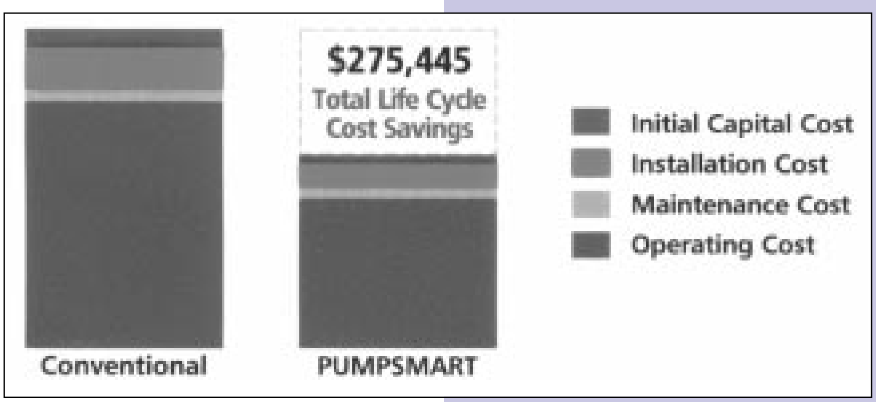
Designing a pump that is heavier, with bigger bearings and a larger shaft does not automatically mean longer life. The primary components in pump failures are mechanical seals and anti-friction bearings. These are brought on not by general fatigue, but by excessive vibration, excessive loads and poor lubrication. These failures are caused primarily by the following upset conditions:

- Dry running-caused primarily by closed suction valves.
- Continuous operation below minimum flow.
- Cavitation due to insufficient NPSH available.
- Heat build-up and subsequent liquid vaporization due to a closed discharge valve.

PumpSmart detects all of these prior to the upset condition occurring and prevents the pump from operating during these transient conditions. The pump will react by stopping, slowing down, alarming or any combination of these actions, depending upon how you want PumpSmart to be programmed.
By utilizing the pump Reliability Factors seen earlier in this edition of Pumplines, we can quantitatively measure the anticipated increase in mean time between failure (MTBF) of PumpSmart as compared to a traditional pumping system. By running a pump at a slower speed, at or close to best efficiency and at a reduced impeller diameter, we will be able to calculate, with your help and input, the expected increase in MTBF for any given ANSI pump currently running in a process application.

PumpSmart is the next level of technology for our industry. This introduction to PumpSmart provides just a glimpse of the product's potential. Initially, this PumpSmart technology will be available on our ANSI models 3196 and 3298. Look for more information in the coming months.

Figure 1



Tech Talk

Flushless Mechanical Seals For Paper Stock Applications

Mark F. Brown
Product Manager
Goulds Pumps/ITT Industries

Introduction

Shaft sealing of centrifugal paper stock pumps is an important consideration for pulp and paper mills. Studies show shaft sealing components such as packing and mechanical seals are the primary cause of maintenance and downtime of paper stock pumps.

Traditionally, paper stock pumps have been sealed with packing (Fig.1). Packing is an effective and forgiving sealing method. However, it requires the constant attention of maintenance personnel. It must be frequently adjusted or packed to maintain the proper leakage rate that is necessary for cooling. Packing also requires a constant supply of flush water that needs to be collected and treated. This adds additional cost and environmental considerations.

Mechanical Seals

Mechanical seals have become much more common and accepted in pulp and paper mills (Fig. 2). While mechanical seals also require flush water, maintenance is much less frequent. The downside is that when a mechanical seal fails, pump disassembly is required to replace it. To minimize unnecessary mill outages, pulp and paper mills are intent on maximizing mechanical seal life. To accomplish this, it is necessary to focus on the primary cause of mechanical seal failures, which is the seal environment. Paper stock applications are flushed with mill water and one may assume that this would lead to very reliable seal operation. However, the reliability and quality of flush water itself often leads to mechanical seal failures. Flush water is the life blood of pulp and paper mechanical seals. If seals could be operated with an uncontaminated and an uninterrupted stream of flush water, the vast majority of seal failures would be eliminated.

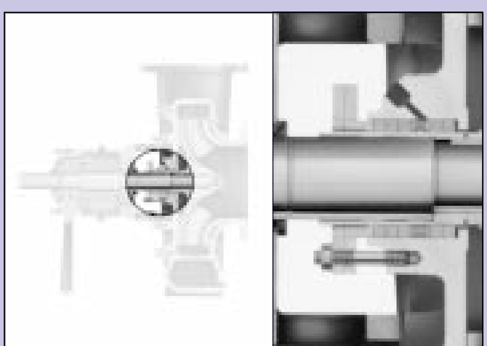


Figure 1 – *Packing can be used to seal paper stock pumps. However, this method requires the constant attention of mill personnel and a constant supply of flush water.*

With existing mechanical seals and seal chambers this is wishful thinking as particulate matter often present in the flush water can accelerate seal face wear. Secondly, air bubbles in the flush water will gravitate to the seal faces and will cause dry running. Lastly, interruption of the seal flush water will allow the paper stock to enter the seal chamber, pack around the seal and cause an imminent failure. Seal water interruption is caused by the inadvertent closing of a valve or plugging of a flush-line. Plugging often occurs after an outage when the paper stock is allowed to flow back into the seal chamber and flush-line. At start-up the flush-line valve may be turned on, but plugging impedes the flow of flush water into the seal chamber resulting in a seal failure.

Mechanical seals are housed in a pump seal chamber which provides the environment for the seal. Prior to discussion on paper stock pump sealing it is important to understand the evolution of seal chambers over the last 15 years.

Seal Chamber Evolution

Since vapor and solids are unavoidable with most liquids being pumped and with flush water, pump designers set out to design seal chambers that would be more forgiving to these environmental certainties. Their mission was to find a way to use the product being pumped to lubricate the mechanical seal, but, keep harmful solids and vapor away from the seal faces.

For this reason Taper Bore seal chambers were developed in the mid 1980s (Fig.3). With the taper bore design it was anticipated that solids and vapor would be centrifuged to the bore and then axially move towards the impeller where they would exit the seal chamber. However, once the solids made it into the box they could not escape. High pressure at the impeller vane tips and low pressure at the shaft made the flow of solids and vapor out of the box impossible. Accumulation of solids resulted in severe erosion of seal and pump parts. In addition, dry running mechanical seal failures caused be vapors trapped in the seal chamber also occurred.

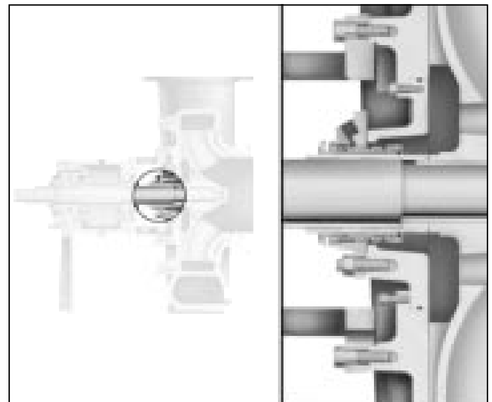
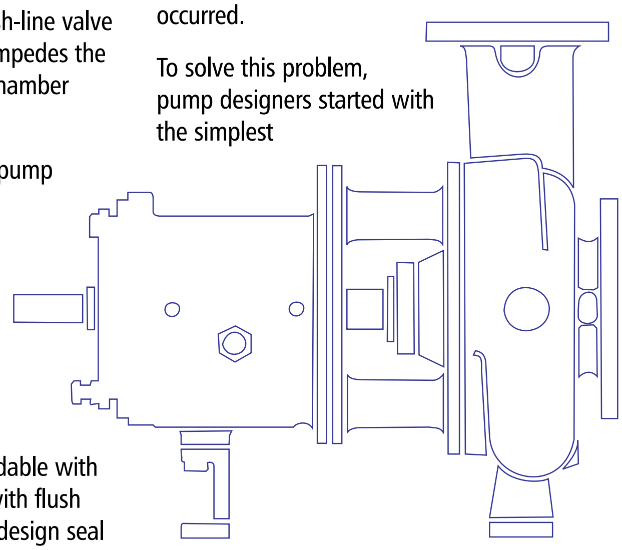


Figure 2 – *Mechanical seals require less maintenance, but, also requires a constant supply of flush water. Reliability and quality problems with flush water are the primary causes of mechanical seal failures.*

To solve this problem, pump designers started with the simplest approach which was to add axial ribs to the bore to disrupt the existing flow profiles. This design does dramatically change the flow profile and is effective at removing solids from the seal chamber. Unfortunately, the new flow profile that allows the solids to escape also deflects solids towards the seal faces which can cause failures. It is a good solution for vapor removal but has a solids handling limitation of approximately 1%.



The ultimate solution to this problem involved creating a flow profile that removes solids

Tech Talk

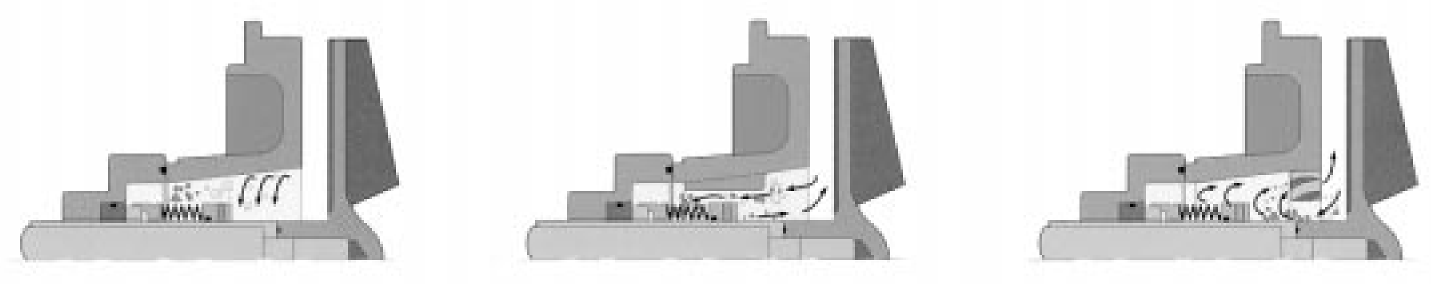


***Figure 3** – The bore of the seal chamber has had many changes since the mid-1980s. On the left, the first tapered-seal chamber failed to adequately handle liquids with solids or entrained vapors. Shown in the center, a tapered bore with axial ribs was able to deal with up to 1% solids content in liquids. The seal chamber on the right uses a vane particle ejector to handle up to 10% solids content.*

without deflecting them at the mechanical seal. Recently, a seal chamber design that meets this objective was introduced. This design uses a "vane particle ejector" or VPE at the seal chamber entrance. A low pressure zone created by the VPE creates a unique flow path that directs solids and vapor away from the mechanical seal. Testing has also confirmed that the back pump out impeller vanes, in conjunction with the VPE, create a turbulent zone which helps to minimize the amounts of solids entering the seal chamber bore. This VPE design extends the solids handling limit to 10% by ensuring that the seal faces are continuously flushed with clean liquid.

Paper Stock Applications

With the success of this seal chamber design in chemical and mining applications, it was decided to test this design on paper stock applications. A prominent seal manufacturer developed a seal configuration that removed the springs from the chamber which is critical to the operation (Fig.4). Then a 500 hour endurance test was conducted including five 60 hour outages to simulate typical mill outages. After the testing was completed the seal was extensively analyzed and was found to show no abnormal wear.

This was followed up by extensive field testing in mills throughout North America with excellent results. This unique seal chamber received a patent for technological improvement. It has proven that it can be operated on 6% oven dry paper stock applications without the need for external flush water and the problems that it can create. The seal chamber and mechanical seal combination allows the seal to extract uncontaminated water from the paper stock without causing de-watering. This clean extracted water ensures that the mechanical seal will operate in a clean friendly environment. In addition, by eliminating the need for flush water mill operating costs can be substantially reduced.

Summary

The development of this new mechanical seal chamber now enables mills to operate pumps without the need for expensive flush water which provides an environmentally safe shaft sealing solution. In addition, mills should see a dramatic increase in the reliability of mechanical seals that will lead to substantial reductions in maintenance costs of paper stock pumps.

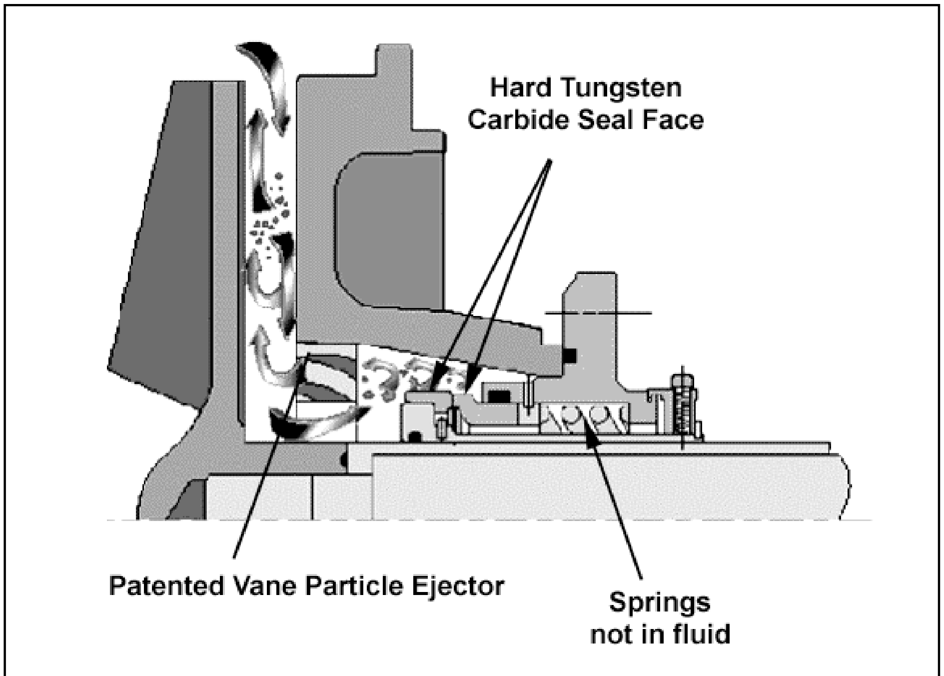
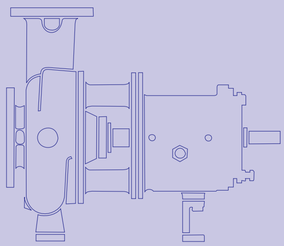


***Figure 4** – This new seal chamber and mechanical seal design will operate on 6% OD paper stock without flush water.*



Tech Talk

Effects of Centrifugal Pump Hydraulics on Reliability

Anthony E. Stavale
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Goulds Pumps/ITT Industries

Introduction

Reliability of mechanical equipment has received considerable attention over the past ten years. Because of the large number of centrifugal pumps, and their relatively low reliability, they have been the focus of much of this attention in the process industries. Numerous articles have been written that address the application, installation, operation and maintenance aspects of reliability. We will discuss some critical application aspects from the point of view of the manufacturer. Relationships between pump design and application will be reviewed and a quantitative approach to selecting the most reliable pump for a given application will be proposed.

A properly selected pump is a necessary, but not sufficient condition for reliable pump operation. Critical installation parameters have been discussed extensively by others. It has been demonstrated that pump reliability can be improved when these parameters are followed, both for pumps that are well selected for the duty, as well as for pumps that are misapplied. Proper pump selection is the key to enhanced reliability.

Pump Selection

Application issues may be divided into two categories; optimum selection of a pump size, and optimum selection of auxiliary equipment (i.e. mechanical seals, lubrication methods, bearings, couplings, etc.). In this article, we will focus on size selection when applied to pumps of a given design (i.e. from a single manufacturer) operating on a given service. We will not consider the effects of different services nor will we discuss selection of auxiliary equipment.

When selecting a pump one of the first things a user does is to determine the head and capacity required. After deciding on a supplier, and a product line, there is still a choice of a number of different size pumps that will handle the duty. As an example Figure 1 lists five ANSI pump sizes from a single manufacturer that could be selected to handle a duty of 300 GPM at 150 FT. Review of the options indicate that the first pump, a 2x3-8 would probably be the least expensive due to its smaller size. Because it draws the least HP (18.2) it would have the lowest operating cost. Figure 1 does not provide any information regarding the relative reliability of the five selections.

Figure 1

SELECTED PUMPS						
PUMP	RPM	IMPELLER DIAMETER (INCHES)	TDH (FT)	HP	NPSHR (FT)	EFF %
2 x 3 - 8	3550	7.00	158	18.2	11.1	66
2 x 3 - 13	1750	12.63	153	18.5	5.6	62
2 x 3 - 10	3550	7.13	157	19.8	15.0	60
3 x 4 - 13	1750	12.50	153	19.8	4.6	58

This is an interesting situation since maintenance expense can be one of the major cost items in the life cycle cost of a pump. A number of surveys have shown that the average mean time between repair for an ANSI pump is 15 months. The average repair cost cited by users is $2500 per repair. Since the average cost of a small ANSI pump is in the $4000 range, <u>the repair costs will exceed the initial cost in less than three years.</u>

Pump Selection Reliability Factors

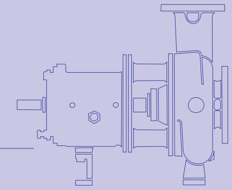
From a reliability point of view there are three major factors that affect the selection; operating speed, impeller diameter, and flow rate. In this article a method of assigning a numerical value for each factor is proposed. This value allows ranking the relative reliability of alternate pumps on each factor. The numerical values range between zero and one, higher values indicating more reliable selections. Since a poor ranking on any one factor can significantly affect the reliability of the pump, an overall reliability index is formed by taking the product of the three individual factors. This product will be referred to as the Reliability Index (RI).

RPM (F_R) The operating speed affects reliability directly through; wear in rubbing contact surfaces (mechanical seals, and shaft seals), bearing life, heat generated by the bearings and lubricants and wear caused by abrasive in the pumpage. For most of these items the wear has a linear relationship to the pump RPM. Thus the RPM factor is taken as a linear function of operating speed. The starting and ending points for the relationship are set as zero RPM and the maximum RPM for which the pump is designed since reliability is a function of the basic design.

A value of 0.2 is assigned to the RPM factor when application is at maximum design RPM. This value was arrived at by recognizing that the index is a comparative rating. The maximum and minimum values of the parameter affect how each parameter is weighted relative to the other parameters. Although the value of 0.2 is somewhat arbitrary, it does insure the RPM is weighted equally with the other parameters. It was also found that the final index values are not very sensitive to this value.

As an example, if a given pump was designed to operate at a maximum speed of 3500 RPM, an application at 3500 RPM would be assigned FR = 0.2. If the same pump was applied at 1750 RPM the value of FR would be assigned a value half way between 0.2 and 1.0, or 0.6 (the speed is one half the maximum).

Impeller Diameter (F_D) The impeller affects reliability through the loads it imposes on the shaft and bearings. Impellers produce two types of loads; one that is relatively steady in both magnitude and direction, and a second which is variable in both magnitude and direction. The first is a result of non-uniform pressure distribution in the casing. It produces a shaft deflection in one direction (at a given capacity) which causes the mechanical seal faces to run off-center, but not wipe radially (for most seal designs). The second load is a result of the interaction between the impeller vanes and casing discharge, and produces a deflection as each vane passes the discharge. This second effect can be very damaging because it causes the seal faces to move radially relative to each other many times per



revolution. The magnitude of this movement may be greater than the steady deflection. Both loads are related to the impeller diameter in a cubic manner. Thus they decrease rapidly as the impeller diameter is reduced, and reliability increases equally rapidly. But, as the diameter is further reduced the possibility of encountering suction recirculation and resulting random loads increases. Since suction recirculation occurs at the pump inlet where fluid energy levels are lower than at the exit, the loads produced by recirculation are not as great as those produced by the impeller/discharge interaction. Consequently there is an optimum diameter which is closer to the maximum diameter than to the minimum which maximizes reliability. Since the loads produced by recirculation are less severe at lower RPM, FD is made a function of RPM. The optimum diameter is taken as 75% of the trim range (25% from maximum).

Thus a pump with an impeller diameter trim range of 10″ to 6″ would be assigned a value of FD = 1.0 when trimmed to 9″ at any speed. When trimmed to a maximum diameter (10″) FD would be assigned a value of 0.0 if operation was at the maximum design RPM, and 0.5 when operated at one half of maximum RPM.

Flow Rate (F_Q) A centrifugal pump is designed to operate most reliably at one capacity for a given RPM and impeller diameter. This flow rate is called the best efficiency point (BEP). At this flow hydraulic loads imposed on the impeller are minimized and are steady. At flows greater than or less than the BEP the hydraulic loads increase in intensity and become unsteady due to turbulence in the casing and impeller. These unsteady loads have the same effect on reliability as the impeller/discharge loads discussed above. In order to measure the effect of these loads a series of tests were conducted on a pump. The tests involved varying the following parameters:

- RPM
- Impeller Diameter
- Flow Rate
- Pump shaft to motor shaft alignment
- NPSH Margin

Vibration at the bearings was selected as a convenient direct indication of the relative shaft motion. Vibration levels are averaged over the range of parameters as a function of flow rate. Results show that the vibration at BEP is 60% of the level at 10% of BEP, and is 45% of the level at 120% of BEP. Thus if a reliability factor for flow is assigned a value of 1.0 at BEP, then values of .60 at 10% BEP and .45 at 120% of BEP are appropriate for this pump.

Experience with pumps of a variety of sizes has shown that smaller pumps vibrate less when throttled back on their curves than do larger pumps. This is probably a result of smaller pumps being more rugged relative to the imposed loads than larger pumps. Thus the reliability factor for flow rate was made dependent on BEP capacity.

A pump selected at BEP capacity is assigned FQ = 1. A small pump (BEP $<$ 50 GPM) is assigned FQ = 0.5 when operated near shutoff. A large pump (BEP $>$ 3000 GPM) is assigned FQ = 0 when operated near shutoff. For all pumps FQ is assigned a value of zero when applied at flows greater than 125% of BEP. This is done in recognition of rapidly increasing NPSHR as well as high impeller loading.

Reliability Index (R_I) The Reliability Index is formed as a product of the three individual factors:

RI = FR x FD x FQ

Values will range from zero to one with higher values indicating greater reliability. Because this factor does not take into account design characteristics it cannot be used to compare pumps of different designs. These design characteristics can include single or double volute, number of impeller vanes, shaft overhang, relative frame loading, etc. The value of the Reliability Index is in assisting the selection of the most reliable pump of a given design in the same service.

An example of the use of the Reliability Index is given in Figure 2. This Figure lists the five pumps identified in Figure 1 which could be selected for a duty of 300 GPM at 150 FT. In Figure 2 these pumps are ranked using the Reliability Index. The fourth column lists RI for each pump. The 2x3-13 has the highest index value, primarily due to operation at less than its maximum design speed. The 2x3-10 has an RI of zero due to selection at 125% BEP. It can be seen in the next to last column that the NPSHR for this pump is considerably higher than the others.

This approach enables the user to have significant additional information available on which to base a selection. Reliability Index can be a reasonable indicator for judging the merits of a range of pump selections of a given manufacturer's design operating in the same service.

Figure 2

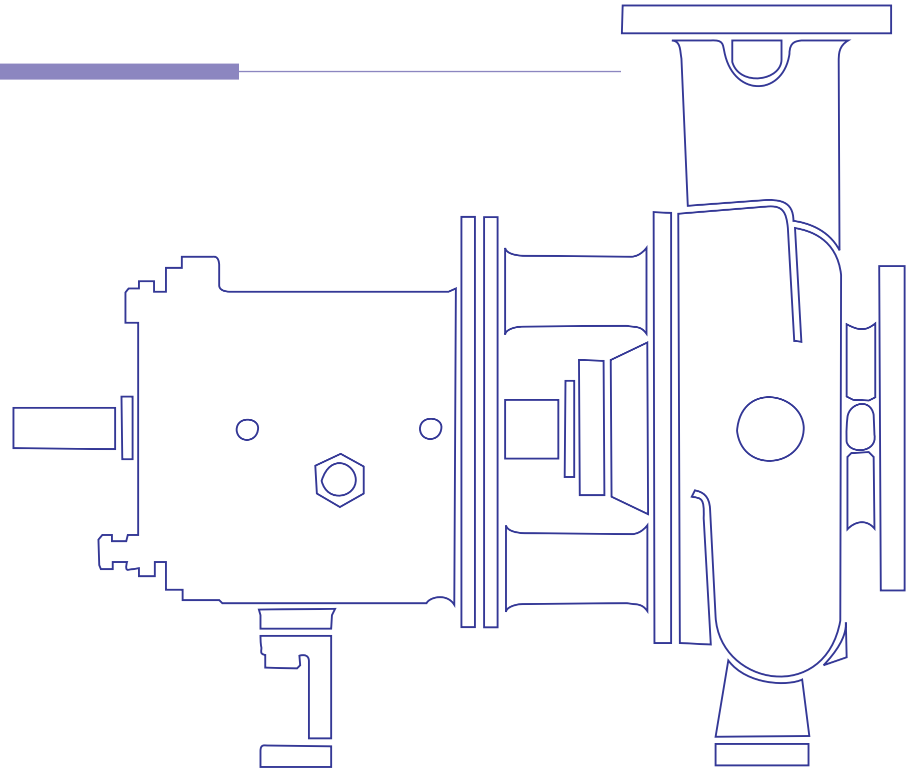
RELIABILITY OPTIMIZED PUMP SELECTION – #1

PUMP SIZE	IMPELLER DIAMETER	RPM	RI	FRPM	FD	FQ
2 x 3 - 8	7.0	3550	0.12	0.2	0.96	0.63
2 x 3 - 13	12.6	1750	0.47	0.6	0.79	1.00
2 x 3 - 10	7.1	3550	0.00	0.2	0.66	0.00
3 x 4 - 13	12.1	1750	0.17	0.2	0.96	0.87

RELIABILITY OPTIMIZED PUMP SELECTION – #2

PUMP SIZE	DIAMETER RATIO	Q/QBEP	COST FACTOR	ENERGY FACTOR	NPSHR FT	NPSHA FT
2 x 3 - 8	0.63	1.15	1.00	1.00	11.1	17
2 x 3 - 13	0.91	1.00	1.49	1.02	5.6	17
2 x 3 - 10	0.28	1.25	1.11	1.09	15.0	17
3 x 4 - 13	0.78	0.58	1.97	1.05	4.0	17

Material Matters



Problems With Copper In Cast High Alloy Austenitic Stainless Steels

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High Alloy Austenitic Stainless Steel

The high alloy austenitic stainless steels are an important group of engineering alloys. They have been used successfully in both cast and wrought forms in a variety of aggressive chemical environments. These alloys are attractive because of their cost advantage over competing nickel-base alloys, and customers often specify them.

Two commonly specified alloys, Alloy 904L (no cast version available in ASTM) and Alloy CN-7M (cast version of Alloy 20) are fully austenitic stainless steels designed for use in various reducing acids such as sulfuric acid, and other aggressive chemical environments. Both alloys contain significant levels of copper intentionally added as an alloying element.

The Problem With Copper in Castings

Alloys like Alloy 20 and 904L were never intended to be cast alloys. It should be recognized that castings are metallurgically more complex than the wrought equivalents, and differences should be expected due to processing.

In castings, segregation during solidification leads to local variations in both chemical composition and microstructure. This results in corrosion resistance and mechanical property differences. During solidification and grain growth, the low melting point elements segregate, or concentrate at inter-dendritic regions and grain boundaries. The wrought products are more homogeneous due to secondary hot working processing, and do not suffer from these segregation effects.

The two major alloying elements, nickel and chromium, exhibit only a slight tendency to segregate. On the other hand, copper, molybdenum, manganese, silicon, sulfur, and phosphorous all significantly segregate to the inter-dendritic and grain boundary regions in cast stainless steels. The large grain size typically found in castings and susceptibility to low melting constituent grain boundary films complicates matters, making these alloys troublesome to cast. Castings with thick sections and complex geometry are particularly susceptible to hot tearing during casting, and weld cracking during repair.

While CN-7M is similar (note chemistry differences) to Alloy 20, there is currently no cast version of 904L available in ASTM. As a wrought product the copper is homogeneously distributed. However, in the cast alloy, the last metal to solidify is rich in copper, which forms grain boundary films.

Recommendations

Be cautious when selecting or specifying high copper containing alloys such as cast 904L or similar alloys, which can produce grain boundary films rich in copper. In certain environments such as electrowinning or electrogalvanizing systems, the copper rich grain boundaries can act as sacrificial anodic regions against the metal grains acting as larger cathode surfaces; resulting in grain boundary loss from being selectively or preferentially attacked. In pumps, this often appears as through wall leakage or corrosion assisted fatigue type cracking failures.

When specifying fully austenitic alloys with low melting point elements such as sulfur, phosphorus, and copper, these elements should be kept to a minimum to eliminate grain boundary weaknesses, and cracking tendencies. Similarly, cast alloys selected for electrolytic services, should also keep these elements to a minimum. Cast alloys such as ASTM A743 Grades CN-3M (Jessop 700 cast equivalent), CN-3MN (AL6XN cast equivalent), or Grade CK-3MCuN (254SMO cast equivalent) are better choices, and recommended for aggressive electrolytic services, since these are low or free of copper additions.

Finally, chemistry control and AOD (Argon-oxygen-decarburization) refining is key to successful production of high copper containing austenitic stainless alloys. This capability is currently being added to the IPG high alloy foundry at our Ashland Operations.

Manufacturing Processes

Continuous Improvement Process

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Implementation of "Best Practices" is playing a key role in the Goulds' Continuous Improvement Process.

Supplier Approval and Selection Procedure is considered a Critical Process that strongly impacts Quality, On Time Performance and more importantly the bottom line: Profit.

The output of this process is to ensure that the selected supplier(s) delivers quality parts on time at a total competitive cost. In appearance this process may sound simple but in reality is a lengthy and complex process and if it is not properly applied can have a negative overall and long-term impact.

The main steps of this process are: define the job requirements, select potential suppliers, screen the "Best in Class" suppliers, benchmark supplier's Quality System and based on the Total Assessment, select the Preferred Supplier(s).

Defining the job requirements is a critical step in identifying a potential supplier. The Procurement Team is a Multi-Disciplinary Team that includes highly skilled personnel from R&D, Engineering, Quality, Marketing, Purchasing and in many cases the Customers. The initial team objectives are to spell out the job requirements, select potential suppliers and to support the concurrence process once the supplier is selected.

The screening process consist in a self-survey, in order to identify and select the "Best in Class" Suppliers.

The On Site Survey benchmarks Supplier's Quality System and Process Capabilities. The survey results play a major role in the Preferred Supplier selection.

On Site Survey covers eleven modules of the Quality System:

1. Manufacturing Facility
2. Quality Organization
3. Purchasing Material Control
4. Document/Design Control
5. Measurement Equipment
6. Process Control
7. Final Verification
8. Non Conforming Material
9. Quality Assurance Records
10. Material Handling/Storage and Packaging
11. Life Test and Reliability

Each module of the quality survey consists of numerous questions, which are scored for 1 to 5 (5 is the best). The percentage of each module is calculated based on the maximum score applicable and the total supplier's score for that particular quality element.
A minimum score of 70% is required in order to pass each of the Quality System modules.

Below is the quality survey rating scale:

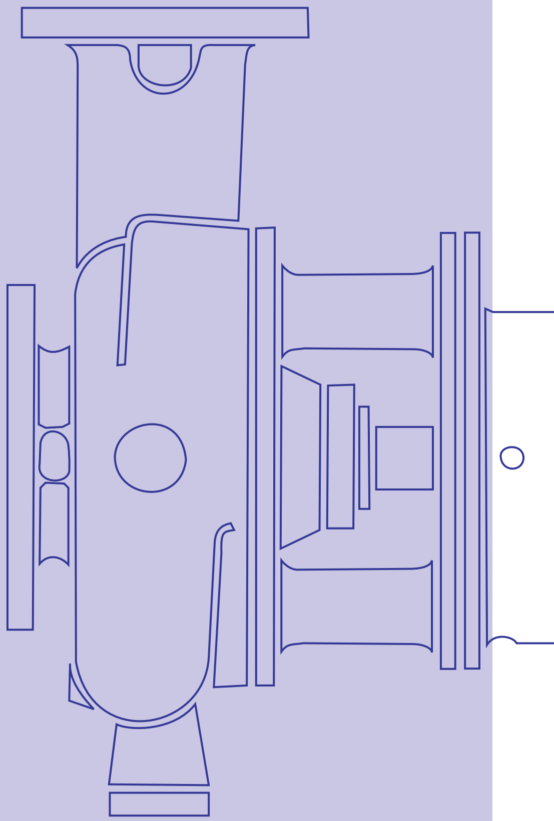
APPROVED (A)
- Each section rated 70% or above.
- Fully approved for all classes of work.
- Must have continual process improvement plan.

CONDITIONAL APPROVAL (CA)
- Each Section rated 60% or above.
- Approved with restrictions for a specific time frame.
- Must have corrective action plan.

LIMITED APPROVAL (LA)
- 7 of the Sections rated 60% or above.
- Must have corrective action plan.
- Approval restricted to specific parts.

APPROVAL WITHHELD (AW)
- 4 or more of the sections rated under 60%.
- Not acceptable as IPG/Goulds supplier at this time.

The implementation of the new Supplier Approval and Selection Process has improved overall Supplier Performance.



Service Solutions

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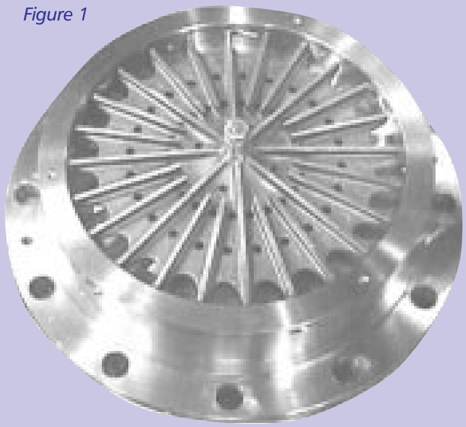
PRO Project Engineers work with Customers and our PRO Service Centers to solve our customers pumping problems by analyzing pumping systems and providing turnkey solutions that make commercial sense.

Figure 1



Problem: A gulf coast chemical plant was experiencing excessive maintenance costs on three 5th Edition, API process pumps due to extreme off peak performance requirements. Vibrations averaged .3 to .5 in/sec at all times. When the pumps were purchased the pumping requirements were 415 GPM at 932 TDH and now the requirements had changed to 35-60 GPM at 932 TDH.

Due Diligence: To properly address this pump modification we had to first review the exact requirements of the pumping system as it currently existed to better decide to either replace or modify the existing pump. The pump was clearly oversized and the pump that this system needed was not readily available in an API design. The solution options were a new ANSI pump, a new vertical can pump, or a re-rate of the existing pump. The customer did not want to have an ANSI pump in this service due to insurance concerns. The vertical can pump was not the preferred option due to the turnkey costs associated with the removal of the existing system and the construction of the new sump and piping required for the vertical pump. The re-rate was preferred if it was possible.

To completely examine the hydraulic issues we went to our family of curves for the API overhung process pumps and did not find one that adequately covered this application. As was stated earlier, we did find an adequate hydraulic fit in our low flow ANSI product line. The first step in this review was to find the basic head and capacity coverage. We always look at the range of flow as few pumps are run at only one point, most are run over a range Low, Design, and High. Then we cover NPSH issues; we never want unexpected cavitation. Typically the NPSH required by the pump should provide an adequate margin of 1.5' to 2' below the customers NPSH available at the maximum flow required. Once, the hydraulic characteristics of the pump are confirmed to be adequate the physical issues of making the existing pump act like the new pump are examined.

All process pumps have two components that develop the head and maintain the capacity, the impeller and the volute (case). Energy is transferred from the driver through the coupling into the shaft and then into the fluid along the working side of the impeller vanes. This is the end of energy transfer from the driver. While energy is being transferred into the fluid, the impeller is also slowing the product down through a diffusion process (smaller area to larger area as the vanes develop from inlet to exit). This reduction in velocity head increases the pressure head as defined by Bernoulli. When the product is discharged into the volute it also is diffused to complete the head required of the pump. The product capacity is regulated by the impeller eye area and also by the volute opening area at the discharge of the impeller. These two areas are sized to define a certain capacity typically known as the Best Efficiency Point.

Solution: For our re-rate to work we had to take all these issues into account and then mechanically conform the existing pump to their requirements. In this case our impeller turned out to be a radial vane open face low flow type (see Figure 1). Typically an API process pump utilizes a closed impeller (see Figure 2). We designed in a case adapter to accommodate the open impeller. The volute area was too large on the existing volute to reduce it by extending the existing volute tongue so we utilized a diffuser designed into the seal chamber cover (see Figure 1). This is similar to a diffuser style pump or barrel type pump. Commercially this allowed the customer to spend about 27 cents on the dollar versus the total replacement of the pump with a new unit. The 27 cents included the removal of the pump by their personnel, the modification of the pump with new components including an upgraded power frame in our PRO shop, new mechanical seal, reinstallation of the pump by their plant personnel and all software associated with the job. The pumps were out of service for 4 to 5 days. Vibration readings averaged .07 in./sec. after this upgrade.

For more information on this re-rate or to explore a re-rate opportunity on another pump please contact your local PRO Service Center or Service Engineer.

Figure 2

Send your comments or suggestions to:
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