48TH TURBOMACHINERY & 35TH PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 9-12, 2019
GEORGE R. BROWN CONVENTION CENTER

CAVITATION SENSING AND OPTIMIZATION IN ROTARY POSITIVE DISPLACEMENT PUMPS FOR TRANSFER DUTIES

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ABSTRACT

Pumps used in terminal transfer duties have a unique set of demands placed on them. They must be able to accomplish all of the following reliably and consistently:

- Transfer as fast as piping network allows (minimize operation time & demurrage charges).
- Unload as much fluid as possible (tank stripping).
- Handle varying suction and fluid conditions including varying viscosities, densities and vapor pressures.
- Transfer to various locations within their facility.
- Clear bottoms/residuals without pump damage.

Increasingly pumps with variable speed drives are employed to allow the transfer rate to be varied according to the fluid and tank conditions as these vary. In order achieve optimum transfer rates, the typically pump needs to be operated as fast as possible consistent with maintaining sufficient margin between NPSHa and NPSHr. In doing so pump cavitation is held to a level that prevents a negative impact on pump reliability.

While variable speed drives offer the possibility of achieving optimum transfer rates, the difficulty presented is determining the optimum speed for reliable pump operation at each fluid condition. Depending on the type of pump utilized several different sensor schemes have been employed to accomplish this. The challenge has been to develop scheme which has wide applicability and reliability with different fluid types.

This paper will discuss the unique constraints around transfer service and the types of pumps utilized together with a summary of the range of application for each type.

It will then present the development of a new cavitation sensing scheme developed for use in rotary positive displacement pumps. The theory behind the scheme will be discussed and complemented with the results of a physical testing program of a PD pump in a loop that allows the fluid viscosity to be varied. Finally, a proposal for how the algorithm can be applied to variable speed service will be discussed.

INTRODUCTION

Globally there are over 4900 tank terminals worldwide in 2300 cities with a combined storage capacity of around 282 billion US gallons (1.1 billion m^3). Tank terminals are a key facilitator of the global trade in fluid products and the role of terminals will continue to evolve in response to trends such as increased tight oil & gas production, growing energy demand in developing countries, sanctions and increases in refining costs. Given its continued importance, it follows that the pumps deployed into these terminals are also important since without them the tank terminal would be unable to function.

PUMPS FOR TANK TERMINAL & TRANSFER OPERATION

Basic Pump types

There are two basic pump types that are deployed into these services – Centrifugal or Positive Displacement. It is important for the reader to understand the physics underlying the operation of each type as they are completely different. Refer also to **Figure 1**.

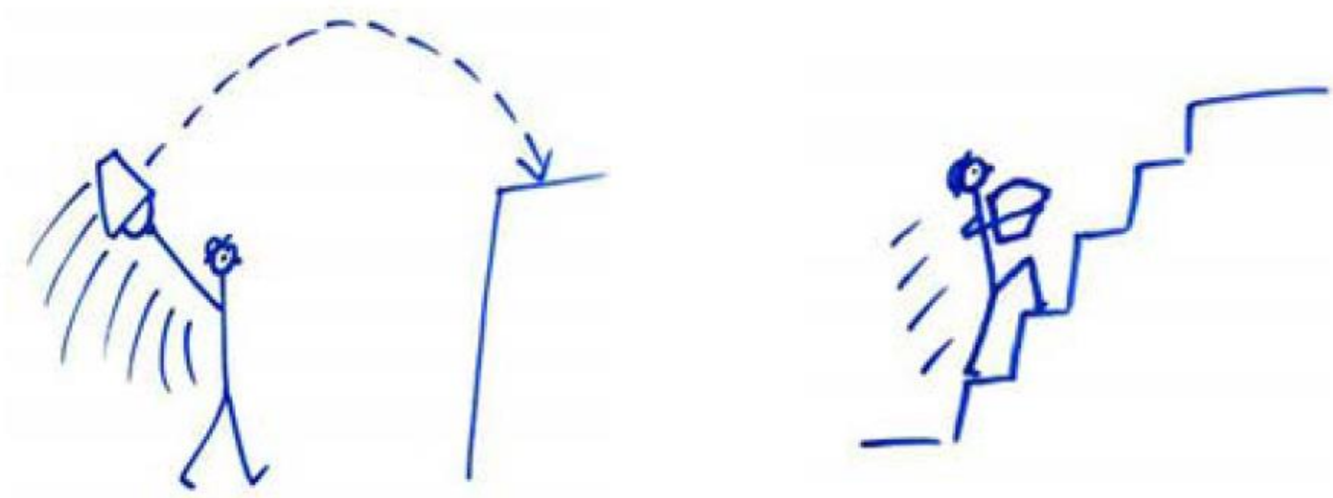

Figure 1.
Centrifugal Pump Operating Principal **Positive Displacement Pump Operating Principal**
Sketch courtesy of Axel Jaeske

(Note that in this paper for the sake of brevity we will focus on the main types of Positive Displacement pumps used for transfer service, these being Twin-Screw and 3-Screw. Some lower flow and/or lower pressure services sometimes utilize a progressing cavity or gear pump.).

A centrifugal pump transmits velocity to a fluid by use of an impeller. As the fluid passes then through the pump's volute casing or diffuser, it converts the velocity into pressure (basically, a kinetic energy machine). Most centrifugal pumps are designed for low viscosity fluid (typically water) as a performance basis.

For higher viscosities standard correction factors (such as Hydraulic Institute Standard 9.6.7 are applied to account for the change in performance. The typical characteristics for a single stage centrifugal pump type BB1 with specific speed 1250 US (24 metric) with a 19.2" (488mm) impeller diameter operating at 1785 RPM are shown in **Figure 2**.

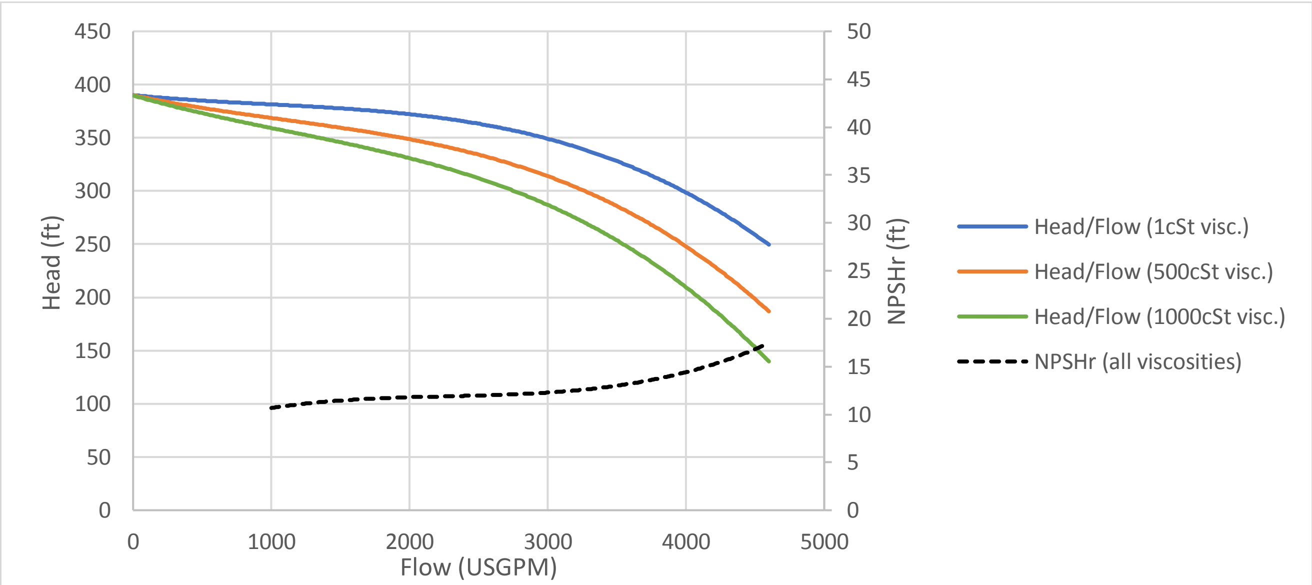

Figure 2. Typical performance for a Ns = 1250 (nq = 24) BB1 pump @ 1785 RPM with a 19.2" (488mm) impeller

A positive displacement pump separates a fluid from its suction passage, encloses that volume in a chamber, and displaces the fluid to the pump's discharge. The key operating principle of a positive displacement pump is the minimization of the fluid's internal slip, generally achieved by the appropriate clearances between rotor and stator. The resulting flow characteristic is insensitive to pressure especially as the fluid viscosity increases. The typical characteristics for a Twin-Screw positive displacement pump with a 10.8" (275mm) rotor and 4" (102mm) pitch operating at 1785 RPM are shown in **Figure 3** (both **Figures 2** and **3** are for an assumed fluid Specific Gravity of 1.0 for simplicity of comparison).

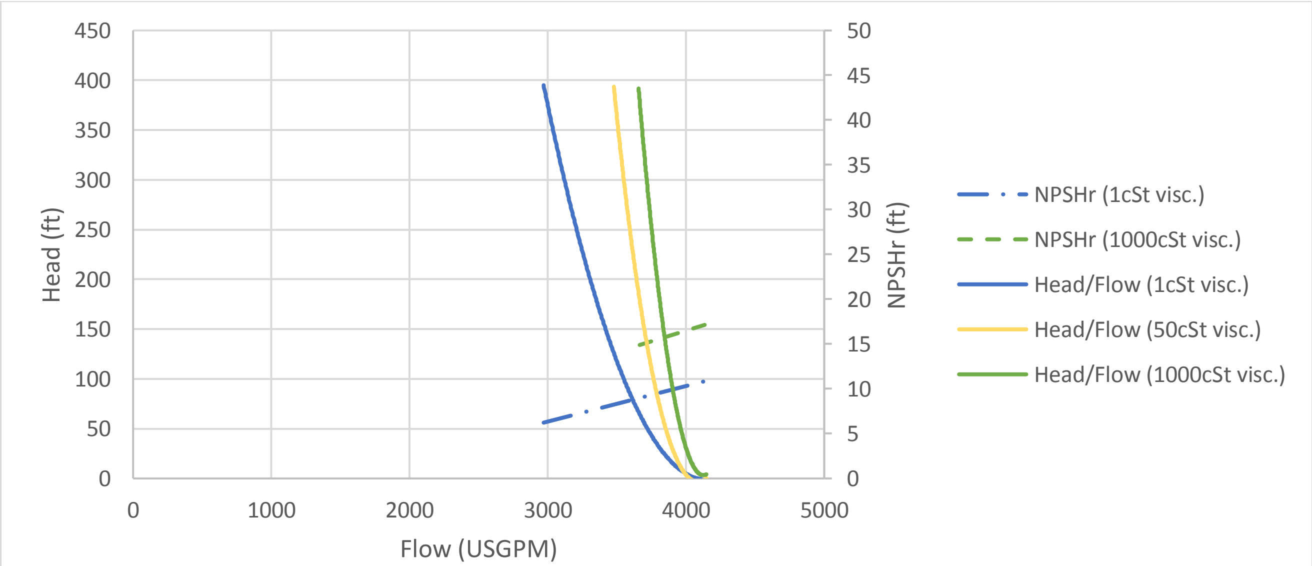

Figure 3. Typical performance for a Twin-Screw with a 10.8" (275mm) rotor, 4" (102mm) pitch @ 1785 RPM

*Note to readers on the use of Head in Y axis of **Figures 3**, **5** and **11**. PD pumps generate Pressure and not Head in response to system resistance. However, for the purposes of facilitating easier comparisons with centrifugal pumps (that generate Head), these figures have been converted to Head using an assumed Specific Gravity of 1.0.*

Examination of pump pressure-flow characteristics and system curve interaction

When a given system head curve is applied, its intersection with the pump curve provides pump performance for a single operating design point. It should be clear by inspection of the two pump characteristics that changes to the fluid properties or system curve will be very different depending on the type of pump deployed.

In the case of the centrifugal pump:

- Changes to the fluid viscosity will change the slope of the pump characteristic curve resulting in a different intersection with the system curve and hence a different operating flow.
- Changes to the system curve will similarly result in a different intersection with the pump characteristic curve and hence a different operating flow.
- Because large variations in operating flow are possible, care needs to be taken to ensure the pump operates within the Preferred Operating Region (POR), which is typically 70% to 120% of BEP (per Hydraulic Institute standard 9.6.3 and API 610). (**The reader is further cautioned that simply selecting a pump to operate in the POR is itself insufficient to assure pump reliability**. This is because neither Hydraulic Institute 9.6.3 or API 610 consider the effect of impeller trimming on apparent BEP. Refer to *Bradshaw 2018* for more discussion on this.)

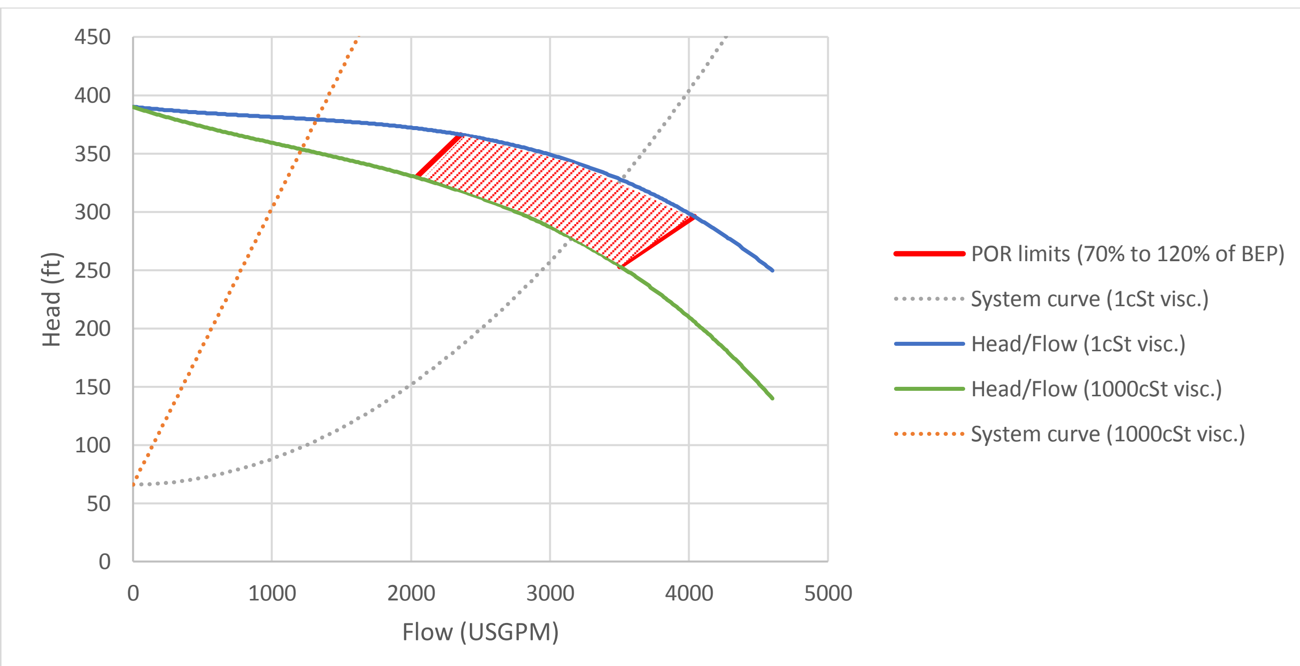

Figure 4. Centrifugal pump POR with system resistance curves of varying viscosity

In the case of the positive displacement pump:

- Changes to the fluid viscosity will slightly change the slope of the pump characteristic curve. The resulting intersection with the system curve and hence the operating flow will remain relatively unchanged.
- Changes to the system curve will result in a different intersection with the pump characteristic curve. While the operating flow will remain relatively unchanged, the pump absorbed power and developed pressure can vary significantly.
- Because large variations in absorbed power and developed pressure are possible, adequate protection is required to prevent overpressure or overpower from occurring in event of a system fault or human error (such as closing a discharge isolation valve). *In the example in **Figure 5** it can be seen that there is no practical overlap of the pump characteristic curve with the 1000cSt system curve, requiring that the system either operate with an open bypass valve or variable speed.*

Note that because of the completely different pumping mechanism used for PD pumps there is no direct equivalent of the centrifugal pump POR for these machines. They can be operated reliably at any combination of flow and pressure within their defined allowable envelope of operation.

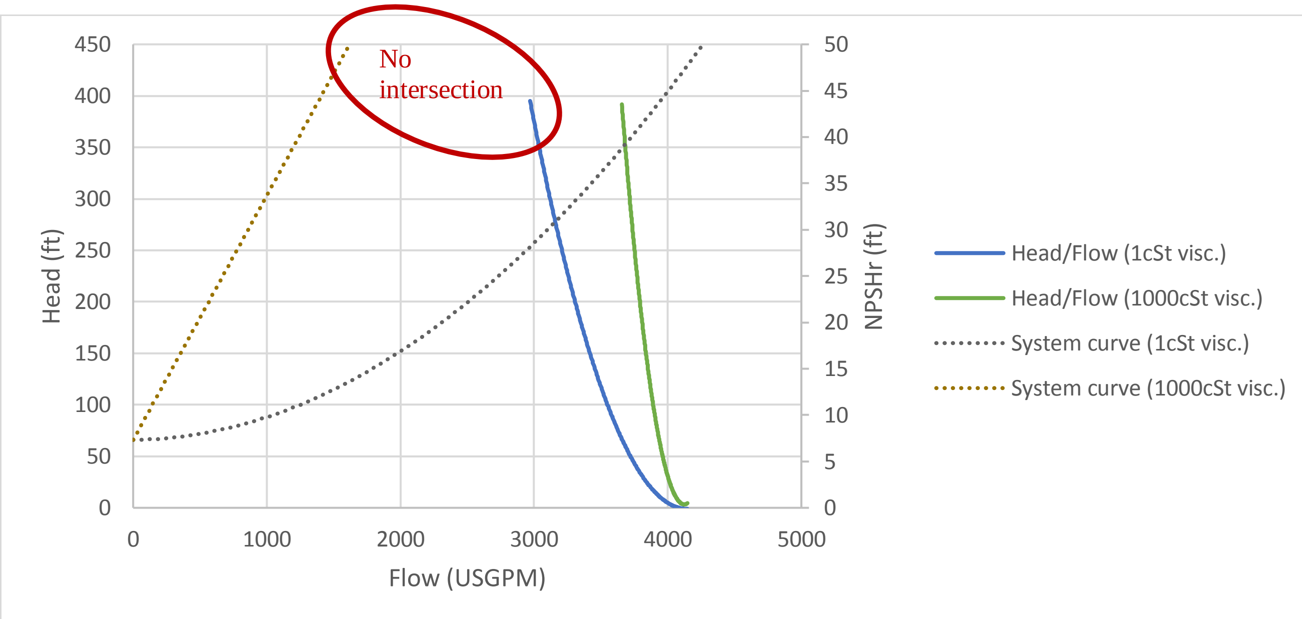

Figure 5. Twin-Screw pump interaction with system resistance curves of varying viscosity

Examination of centrifugal pump NPSH characteristics

NPSHr for a centrifugal pump is a function of the flowrate when speed is constant. At flows below shockless flow (Q_{SF}) and above the onset of suction recirculation flow, the impeller component of NPSHr according to *Gülich J.F. 2010, Equations 6.10 and 6.21* is defined as follows. (The equation numbers used are the same as those in the book for clarity).

$$NPSHr = \lambda_C \frac{c_{1m}^2}{2g} + \lambda_W \frac{w_1^2}{2g} \qquad (6.10)$$

$$\lambda_W = \lambda_{W,opt} \left(\frac{tan\beta_1}{tan\beta_{1,opt}}\right)^{0.57} \qquad (6.21)$$

The author’s experience is that NPSHr in this region can conservatively assumed to have close to a linear relationship with changing flow. At flowrates below the onset of suction recirculation NPSHr will typically remain constant or rise slightly for most of the pump specific speeds utilized for transfer services.

For flowrates exceeding the impeller shockless flow, NPSHr typically increases by an exponent of 2 to 3 in accordance with *Equation 6.22*, making it highly undersirable to operate a centrifugal pump at these flowrates since the NPSHr can quickly become higher than NPSHa causing a large drop in performance and resulting in cavitation of sufficient magnitude to negatively affect reliability.

$$NPSHr = NPSHrSF \left(\frac{Q}{Q_{SF}}\right)^{2\ to\ 3} \qquad (6.22)$$

(*with some simplification by setting* $Q_{sa} \approx Q_{SF}$)

These characteristics of a centrifugal pump are summarized in **Figure 6** below. It should also be noted that while increasing fluid viscosity will also increase NPSHr in a centrifugal pump (refer to Hydraulic Institute standard 9.6.7), it is seldom the limiting factor in their application.

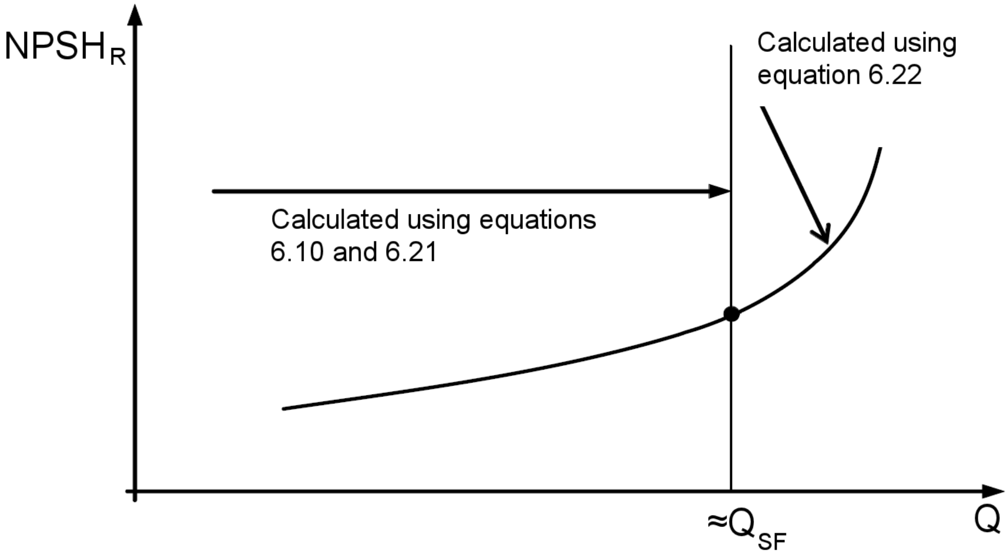

Figure 6. NPSHr curve shape definition for centrifugal pumps

Examination of positive displacement pump NPSH characteristics
For Twin-Screw or 3-Screw Positive displacement pumps, NPSHr is a function of three variables:

1. The axial velocity of fluid entering the pumping screws.
2. The viscosity of the pumped fluid.
3. The pressure developed by the pump.

Items 2 and 3 have a similar mechanism of action in that a higher fluid viscosity or a low developed pressure result in a higher pump volumetric efficiency. A high volumetric efficiency means there is less slip flow (leakage) back to the suction end of the screws which in turn means more of the fluid filling of the screw chambers must be done from the casing suction passage. Refer to **Figure 7**.

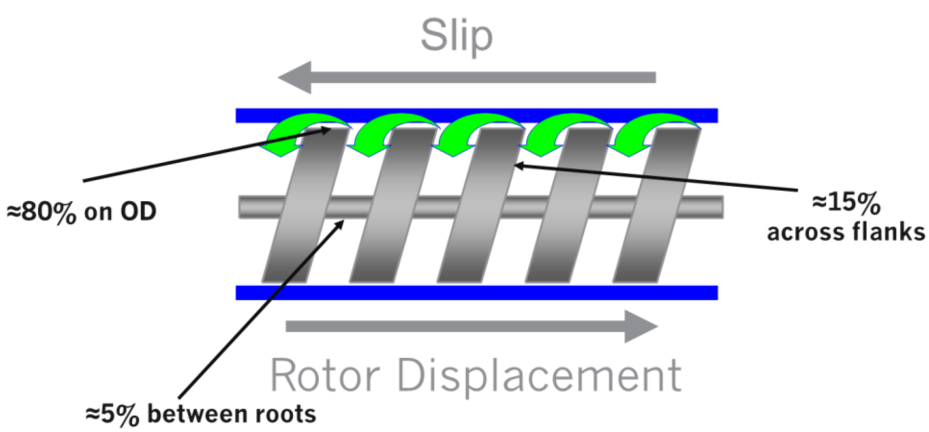

Figure 7. Leakage flow distribution in a typical screw pump

In some designs the internal clearance in the pump is increased in order to reduce both volumetric efficiency and NPSHr. However, since this results in a lower overall pump efficiency it must be applied carefully at the system design stage in full understanding of the overall lifecycle costs.

A typical overall NPSHr characteristic is shown in **Figure 8** for different fluid viscosities and developed pressures (converted to head @ SG = 1.0 for comparison).

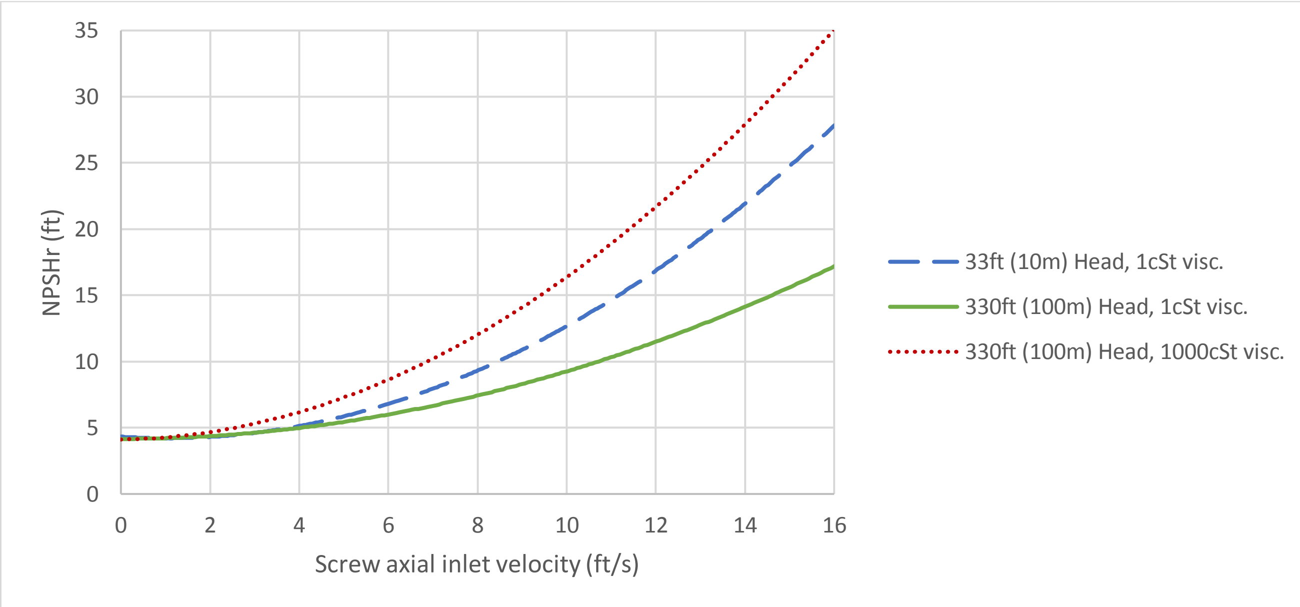

Figure 8. Typical NPSHr characteristic for a screw pump

Examination of pump gas/vapor handling and dry run characteristics
Centrifugal pumps have a very limited tolerance of gas or vapor. **Figure 9** taken from *Gülich J.F. 2010, Figure 13.17* below shows the measured performance of a 1300 specific speed (26 metric) centrifugal pump with a gas content of 0 to 6.4%. It can be seen that above 2% gas, performance becomes significantly degraded. In general, it can be stated that the application limit for conventional centrifugal pumps in transfer service is 5% gas. Beyond this no reasonable operation is possible.

Note that equipping a centrifugal pump with an inducer can significantly improve the performance gas handling performance. The authors are aware of designs operating successfully with up to 25% vapor. However, this creates several design tradeoffs (increased bearing span, special casing design, reduced POR) and it is only seldom used in practice.

Dry running is not recommended for centrifugal pumps because once fluid priming is lost, re-priming of the pump is required. If the probability of dry run is high, the pump should be equipped with an automatic dry run detection and priming system.

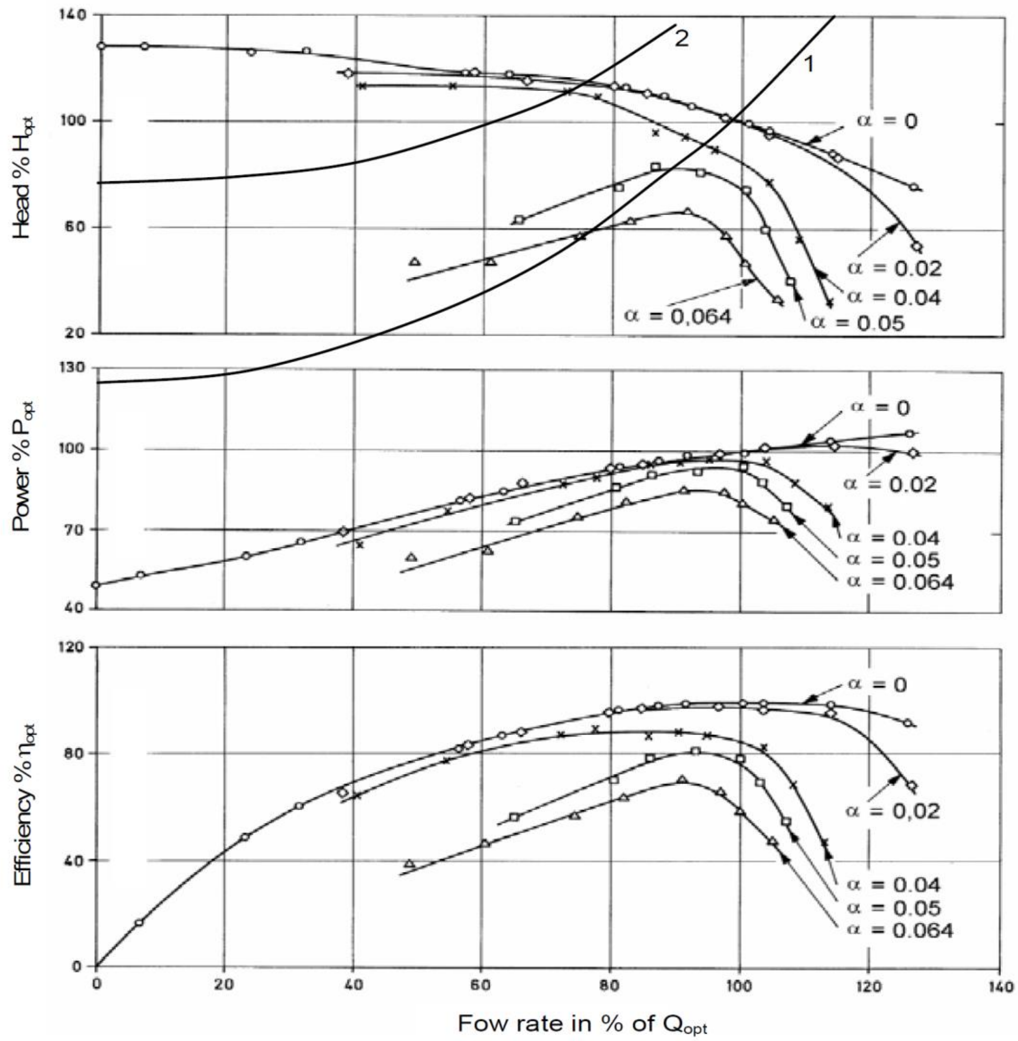

Figure 9. From *Gülich J.F. 2010, Figure 13.17*

When engineered to do so 3-Screw designs can tolerate continuous gas contents of up to 20% although this comes at the cost of lower volumetric efficiency. This is because the internal clearances of the pump need to be adjusted to ensure even compression of the gas as it passes through the pump. It is therefore important that the maximum gas content be specified in advance if a 3-Screw pump is utilized.

Standard Twin screw pumps can tolerate in excess of 80% continuous gas content with only minimal performance change. Specially engineered multiphase Twin-Screw pumps can tolerate in excess of 95% gas. Refer to *Prang, A. J., Cooper, P., 2004* for further discussion on this.

If dry running is anticipated (and provided the piping and pump casing are configured to support it), Twin-Screw pumps are capable of limited dry running - the limit being the resulting heating of the remaining fluid in the casing. 3-Screw pumps are not recommended for dry running as they rely on fluid films to maintain separation of the rotors and casing.

VARIABLE SPEED & VFD OPERATIONAL REQUIREMENTS

Variable speed capability is strongly indicated for any tank transfer service and should be considered mandatory for efficient tank stripping and residue removal. It should also be considered mandatory when a significant range of different viscosities must be handled. A VFD (Variable Frequency Drive) is most commonly used to provide variable speed, however alternatives such as variable fill fluid couplings or gas/diesel engine drives are also available.

If variable speed operation is ***not*** used then the pump needs to be sized to have both low NPSHr and achieve the required flow (transfer speed). This normally results in a slow running pump selection (to achieve the NPSHr target) and because the pump is slow running it must be made large to achieve the required flow. Slower running drivers are also physically larger. The net result will be a higher 1st cost for the pump, driver, package, pipework and foundation.

A further concern with fixed speed operation is when the range of fluid viscosities and vapor pressures that must be handled cover a wide range. Selecting a large slow running pump to cover the "**edge**" cases where NPSHa is low due to increased friction in the suction piping (usually due to viscosity) or a high vapor pressure may not be economic.

Tank stripping and residue removal attempted without the benefit of variable speed is much more likely to be incomplete due to the higher pump NPSHr and the need (with some pump types) to avoid extended dry running. This may require secondary techniques and/or systems to handle the residual fluid left in the tank. This will increase both 1st cost and ongoing maintenance costs.

Centrifugal Pumps application of variable speed

For centrifugal pumps, a change in speed affects the pump HQ and power characteristics in accordance with the standard affinity laws:

$$Q_2 = Q_1 \left(\frac{N_2}{N_1}\right)$$

$$H_2 = H_1 \left(\frac{N_2}{N_1}\right)^2$$

$$P_2 = P_1 \left(\frac{N_2}{N_1}\right)^3$$

The pump NPSHr is also affected by a speed change. While some sources choose to use the same exponent of 2 used for the change in head with speed, the authors recommend a more conservative exponent of 1.7. The reader is referred to *Guelich J.F. 2010 section 6.2.3* for a more in-depth discussion of the reasons why exponents in the range of 1.3 to 1.8 may be encountered especially in low NPSHr services. Following this we can write:

$$NPSHr_2 = NPSHr_1 \left(\frac{N_2}{N_1}\right)^{1.7}$$

In a continuation of **Figure 4**, **Figure 10** shows the resulting typical centrifugal pump performance when a speed reduction from 1785 RPM is applied in order to get pump operation inside the POR for a 1000 cSt viscosity fluid. It can be seen that even a significant speed reduction to 1185 RPM still results in the pump operating well outside of the POR negatively impacting reliability. NPSHr is reduced in line with the equation given previously.

In this example no amount of speed reduction will achieve running in the POR. From this we can conclude for this specific service a centrifugal pump is not the best selection for reliability.

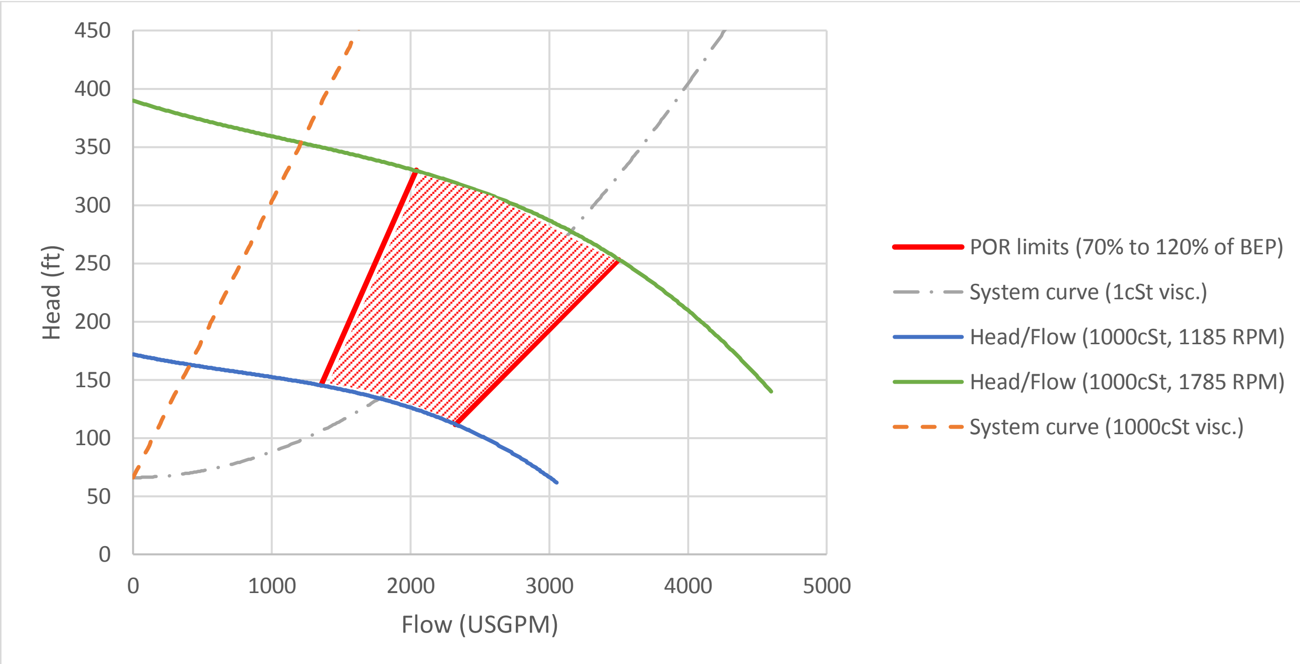

Figure 10. Centrifugal pump POR with system resistance curves of varying viscosity & variable speed

Performing the same evaluation with the Twin-Screw pump **Figure 11** shows that 1000cSt fluid can be handled reliably by slowing the pump down to around 750 RPM.

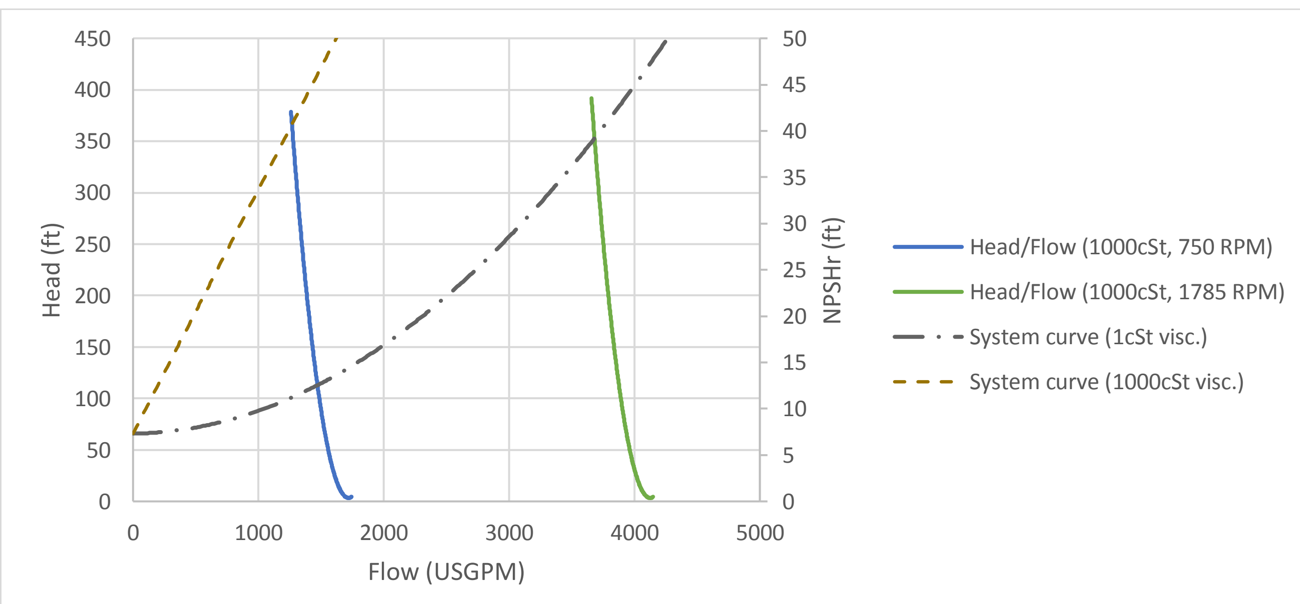

Figure 11. Twin-Screw pump interaction with system resistance curves of varying viscosity & variable speed

TEST PUMP SETUP

Table 1. Test pump details

Model	FSXA655
Fluid Viscosity	200 SSU (43 cSt)
RPM	1750
Discharge Pressure	500 psi (34.5 bar)
Capacity	163 USGPM (37 m3/h)
Suction lift	15 IN HG (5.12m)
Power	67.5 HP (50.3 KW)
Screw Diameter	4.875" (124mm)
Pitch	1.0625" (27mm)
Fluid Closures	3.5

The pump selected for experimental testing was single piece casing design without a liner which simplifies the installation of test instrumentation. The rotors were integral to the shafts and had two single component seals fitted at each shaft end. The pump is a "Low" NPSHr design with a constant-area suction, particularly suited to Loading and Unloading applications. The pump specifications are presented in **Table 1**.

The pump was modified by the addition of three pressure taps, one at the suction, one at the discharge and one approximately midway along the screw flights. **(Figure 12**)

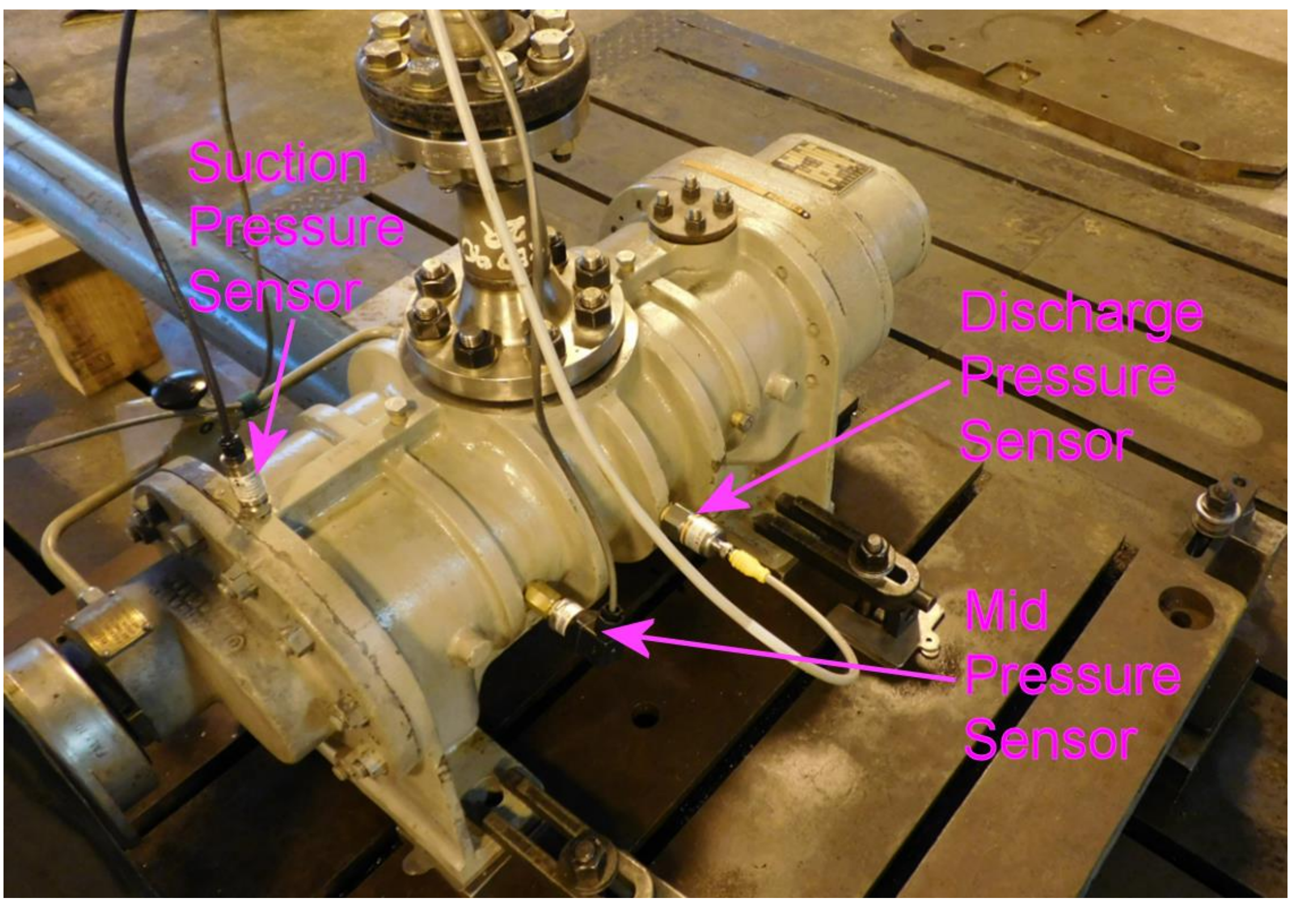

Figure 12. Pump setup showing pressure sensors

A schematic of the test setup is shown in **Figure 13.** An automated valve located on the suction line was used to control the suction pressure and the discharge pressure was controlled manually via the discharge valve.

Flow, temperature, and pressure data are routed to a standard industrial PLC controller which was responsible for relaying the signals to the PC. The PLC controller also executed the Cavitation Control algorithms and controlled the speed of the pump via the VFD.

A PC test interface was connected to the PLC for manipulation of the controller parameters, visualization of the signals and datalogging. The pumped fluid was heavy fuel oil with a viscosity range of 3000 to 6000 SSU (600cSt to1300cSt) at 80°F to100°F (27°C to 38°C).

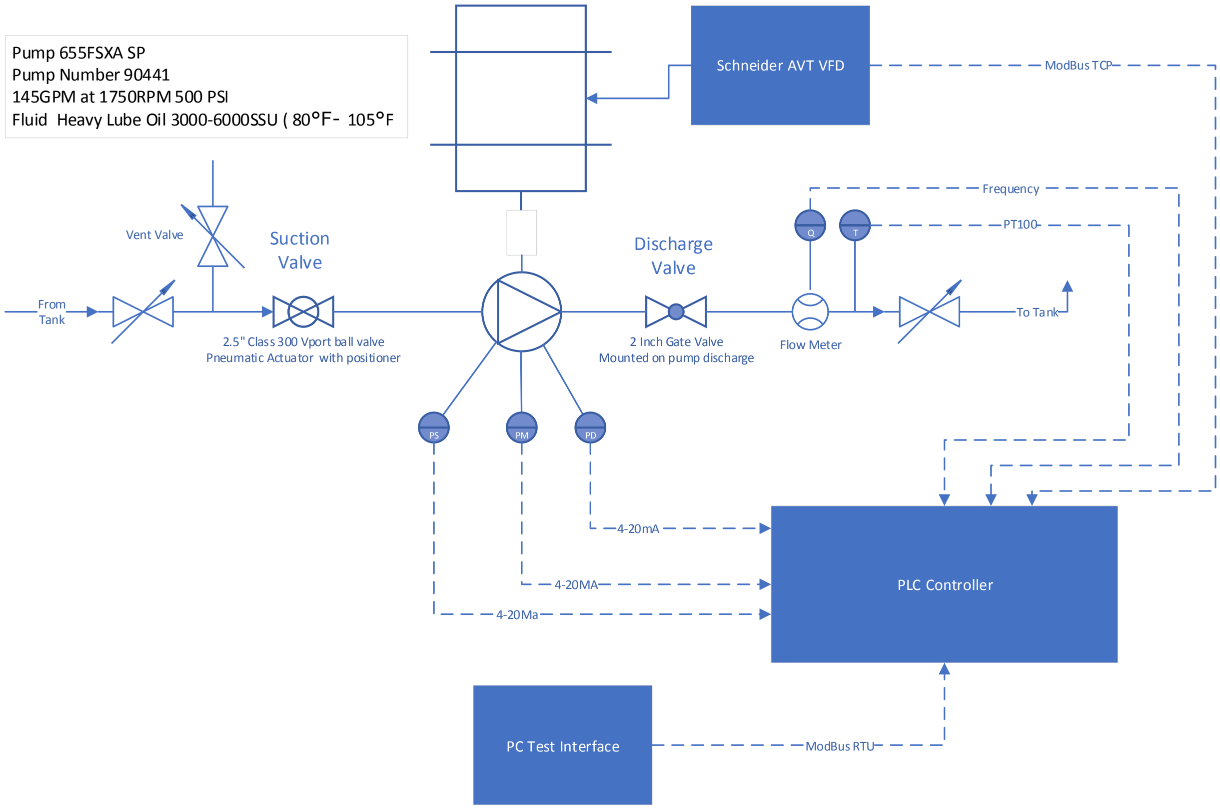

Figure 13. Test schematic

TEST RESULTS

Cavitation Control Performance

Figure 14 is a plot of the Discharge Pressure, Pump Speed, Flow Rate and Suction Valve Position with the pump running.

At 0 seconds cavitation control is not enabled. The pump speed is 1000 RPM, the Suction Valve is 48 percent open, the flow is 95 USGPM and there is audible cavitation. The average flow loss due to cavitation of approximately 5%. The effects of the cavitation can clearly be seen be seen in the variation of the discharge pressure signal, and to a lesser extent on the flow signal. At approximately 22 second the cavitation control is enabled, and the system responds by ramping down the pump speed to below the level that cavitation can be detected. The speed is reduced to approximately 900 RPM resulting in average flow of 90 USGPM (20.4 m3/h), 90% of the starting flow, which is 5% lower than the cavitating flow rate. At approximately 165 seconds the suction valve is closed by 5 degrees, again resulting in cavitation as can be seen by the Discharge Pressure signal. The system responds by reducing the speed further until 'no cavitation' speed is reached.

During cavitation control, the pump speed modulates between a pre-defined dead band and at a frequency defined by control parameters which control the ramp up and ramp down speed of the VFD. In **Figure 14** the speed variation is +/- 6rpm at a frequency of 0.2hz.

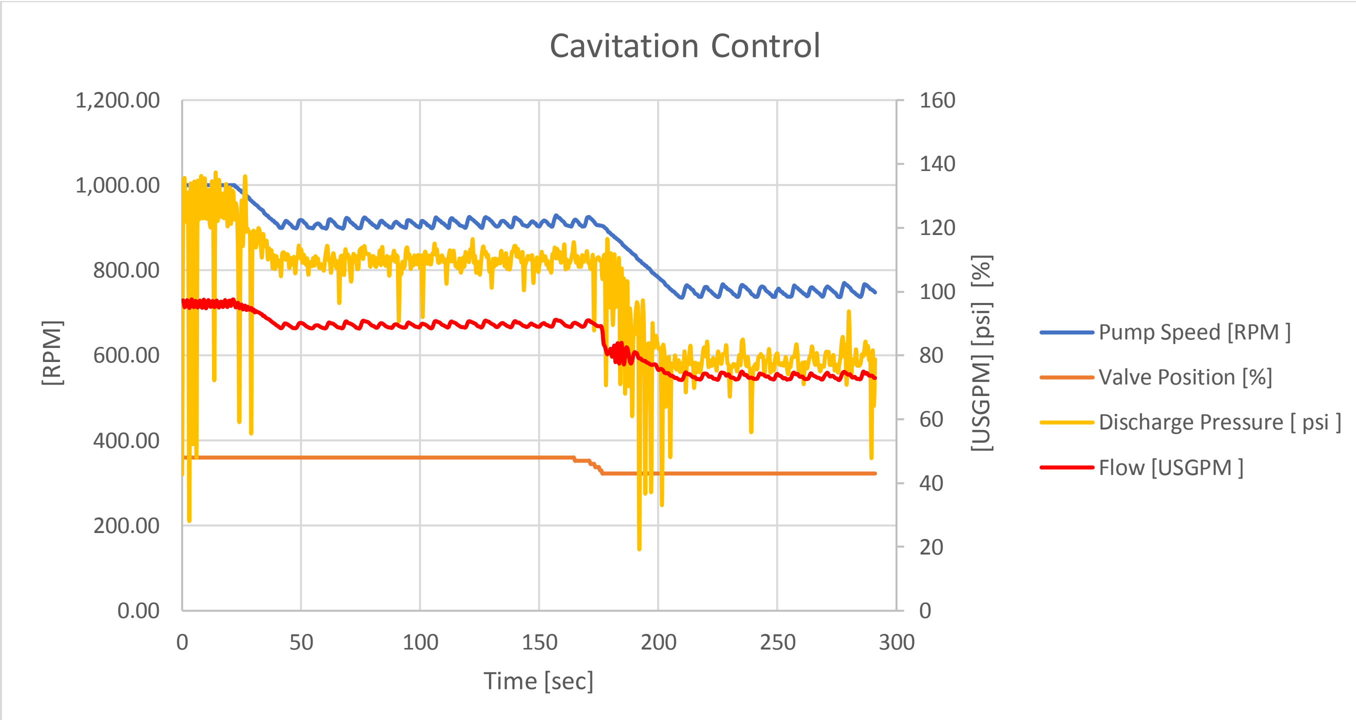

Figure 14. Cavitation Control operation

Table 2. Cavitation Flow Loss vs Cavitation Control Flow

					Cavitation Control Active			
Condition	Valve Position	Speed [RPM]	Average Flow [USGPM]	% Full Flow	Speed [RPM]	Average Flow [USGPM]	% Full Flow	Diff
1	100%	1000	100.40	100%	1000.00	100.40	100%	0.00%
2	48%	1000	99.30	98.90%	986.00	97.70	97.31%	1.59%
3	47%	1000	96.25	96.92%	905.22	89.17	91.27%	5.66%
4	45%	1000	90.71	90.35% Severe Cavitation	822.41	80.78	80.46%	9.89%

Analysis of data obtained from tests similar to those plotted in **Figure 14** are listed in **Table 2**. The four test conditions were generated by modifying the position of the Suction Valve. Condition 1 has the valve full open, no cavitation is present therefore the cavitation control maintains the speed at 1000 rpm. Conditions 2 and 3 have cavitation flow loss of 1% and 3% respectively. When cavitation control is active the flow is further reduced by 1.6% and 5.7%.

Condition 4 had severe cavitation when running at full speed, with the flow loss average of almost 10%. Cavitation was eliminated when the cavitation control was activated. The cavitation control reduced the speed resulting in a flow 10% lower than the flow produced by the severe cavitation. It can be seen from the table that the flow reduction, when cavitation control is active, is approximately double that of the flow lost to cavitation but is without the noise or vibration indicative of reduced pump MTBF.

Sensitivity to flow Loss

"***Factor***" is the primary variable that is used in the cavitation control algorithm to regulate the speed of the pump to prevent cavitation. Test were conducted is to examine the correlation between the ***Factor*** calculated by the algorithm and the normal cavitation flow loss experienced by adverse inlet conditions.

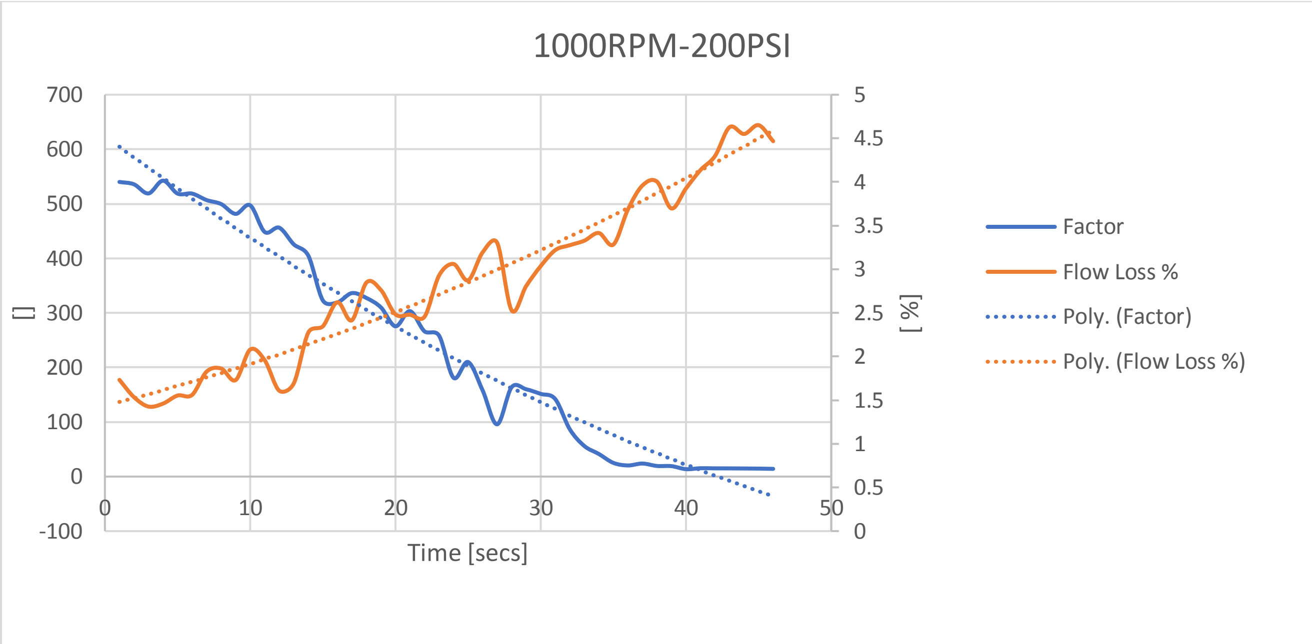

Figure 15. Flow Loss vs. Factor

Nine test conditions were run. For each speed of 1500rpm, 1000rpm and 500rpm, Discharge Pressure was set to 300psi (20.7bar), 200psi (13.8bar) and 100psi (6.9bar) at the start of the test. Cavitation Control was switched off and test then proceeded as a typical NPSH test.

The suction valve was closed slowly and the resulting flow loss due to cavitation was recorded along with the ***Factor***. The ***Factor*** and flow loss were plotted for each condition. **Figure 15** presents the data recorded for the 1000rpm @ 200psi (13.8bar) and is typical of the other 8 conditions. From these plots the corresponding flow loss at a ***Factor*** value of 300 and 100 has been extracted and is listed in **Table 3** and **Table 4**.

Table 3. Flow Loss at different operating conditions with *Factor* Setpoint of 300

Flow Loss at Factor = 300					
		Discharge Pressure psi (bar)			
Speed [RPM]		300 (20.7)	200 (13.8)	100 (6.9)	Avg.
	500	1.0397	0.8782	0.62195	0.846617
	1000	3.5912	2.405963	1.785	2.594054
	1500	1.535306	2.692744	2.658875	2.295642
	Avg.	2.055402	1.992302	1.688608	

Table 4. Flow Loss at different operating conditions with *Factor* Setpoint of 100

Flow Loss at Factor = 100					
		Discharge Pressure [psi]			
Speed [RPM]		300 (20.7)	200 (13.8)	100 (6.9)	Avg.
	500	2.5507	2.491525	2.4032	2.481808
	1000	4.918848	3.4805	2.6488	3.682716
	1500	2.10714	1.706704	2.19	2.001281
	Avg.	3.192229	2.559576	2.414	

The ***Factor*** parameter is sensitive to suction conditions that result in a flow loss of between 0% and 5%. The ***Factor*** decreases at the onset of flow loss and typically reaches its minimum when the flow loss is > 5%. The average flow loss values across all conditions is 1.9% and 2.7 % for ***Factor*** setpoints 300 and 100 respectively. In each ***Factor*** value examined, the equivalent flow loss decreases as pressure decreases.

Response to Tank Level Reduction

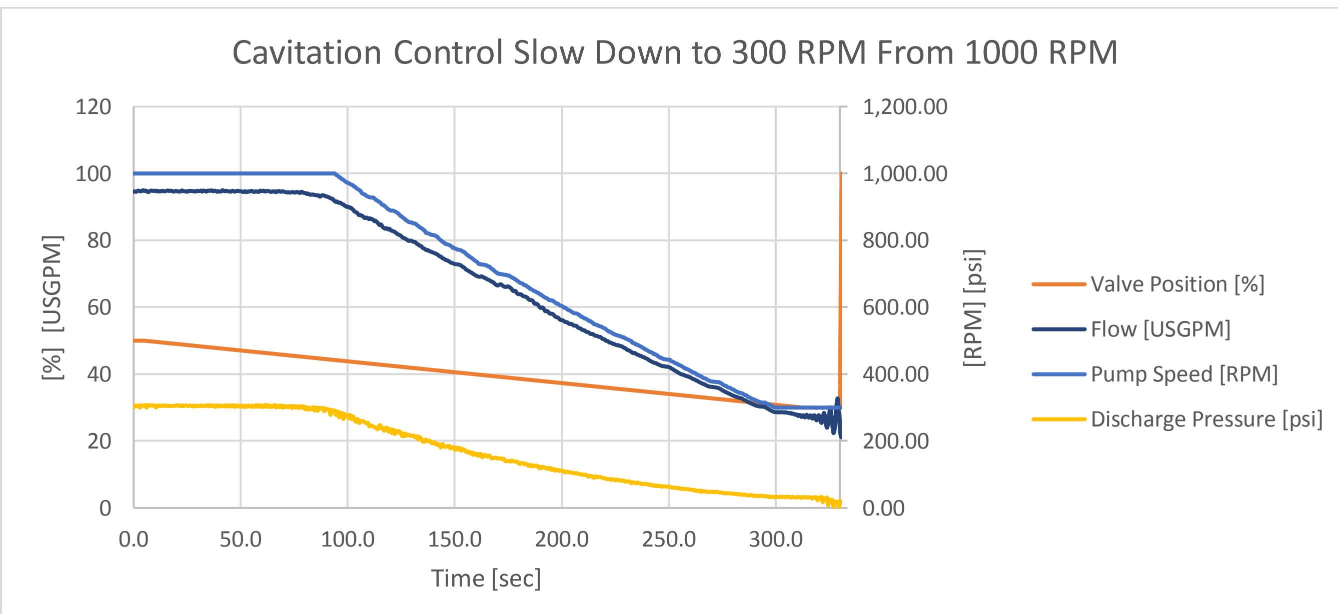

Figure 16. Cavitation Control response to as simulated tank level reduction

The reduction in tank level was simulated by a slow closing of the suction valve. At the onset of cavitation (at approximately 95 seconds), the Cavitation Control engages, and the speed begins to reduce. As the valve continues to close the cavitation control continues to reduce the speed, preventing cavitation, until the speed reaches the preset minimum speed of 300rpm. Thereafter, the valve continued to close without any further reduction in speed resulting in cavitation as can be seen by the disturbance in the flow rate at the end of the plot and the drop off of the Discharge Pressure.

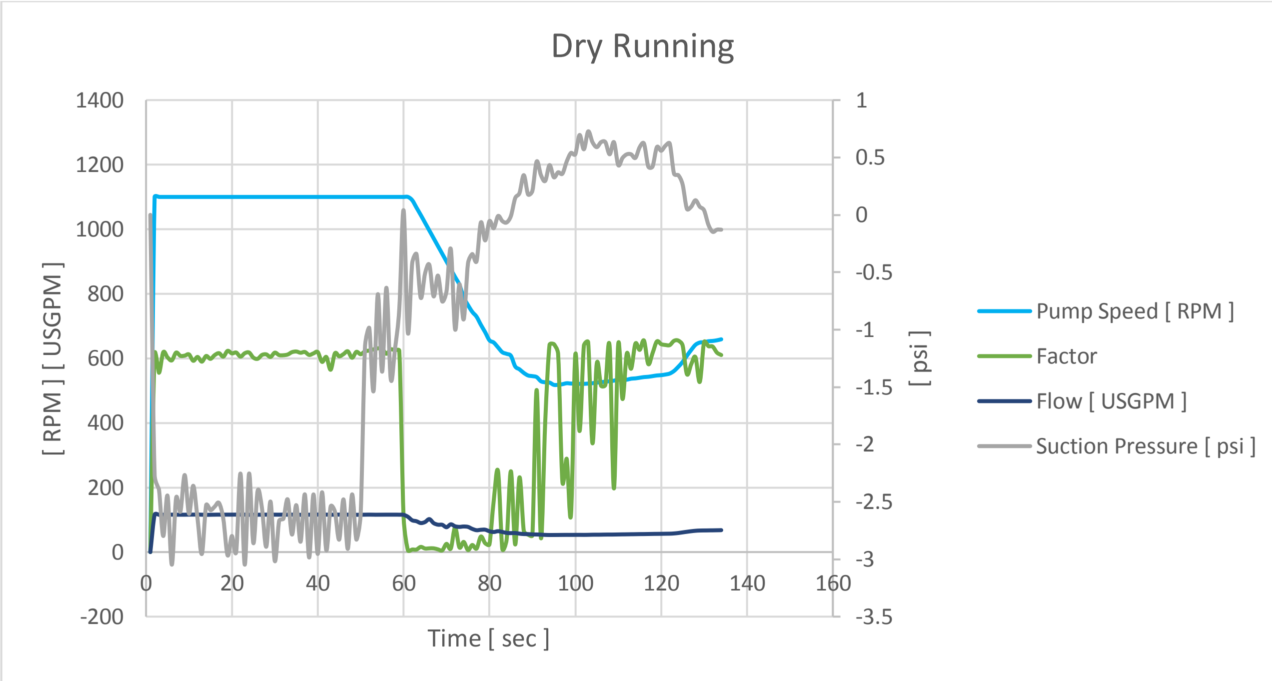

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Figure 17. Dry Running signals

Dry running Detection

A vent valve was opened to the atmosphere permitting air to be drawn into the suction pipe and ingested by the pump. **Figure 17** shows the effect of this action on the recorded signals. The ***Factor*** can be seen to be constant at 600 (no cavitation) until 60 seconds when it rapidly drops to zero as the air is ingested and the cavitation control algorithm commands the pump speed to fall as if under cavitation conditions.

Some distinct characteristics can be seen which can be used to differentiate dry running from cavitation. The suction pressure can be seen to rise before the ***Factor*** drops to zero and before the pump speed is reduced. This rise can be attributed to the drop in the suction piping resistance as a result of significant amounts of air in the line. The speed falls but ***Factor*** recovery is unstable, which is not the case when responding to cavitation.

DISCUSSION

The subject of cavitation detection is a widely researched subject, particularly in centrifugal pumps. Methods that employ vibration (sound) require a cavitation severity high enough to distinguish between the vibration due to cavitation and the background vibration. This requires a sophisticated frequency analysis in order to be effective, increasing the cost of instrumentation. Refer to *Sloteman 2007*. Significant (and usually audible), cavitation must therefore be present in order for easy detection to be made. The cavitation frequencies of interest often must be tailored for a specific application/fluid/speed etc. This explains the scarcity of operational commercial cavitation detection and control systems in the world today. The use of cavitation detection to actively control the operating envelope of a pump continues to be mostly a research lab project.

Any cavitation control system must also be able to manage cavitation without unduly reducing the pump flow rate beyond that which could be obtained before the cavitation in the pump becomes unacceptably high. The result presented in this paper indicates that the method employed using pressure only detection is sensitive to very low cavitation levels (**Figure 14**), and is typically able to detect if a pump is experiencing mild cavitation. i.e. cavitation which results in a flow loss between 1% and 5%, it also requires no audible indication of cavitation. This operating condition is not considered an issue with Twin-Screw pumps.
Using the 1-5% criteria for a threshold, the control algorithm will always be reducing the speed ahead of any significant cavitation. The system will typically be operational in or around conditions 2 and 3 listed in **Table 2**, where the flow is 1% to 3% below the flow that would otherwise result from a fully cavitating condition. This system (by implication), is operating above the NPSHr of the pump.

The response control algorithm can be tuned to optimize this operation by adjusting thresholds and ramp rates. In typical applications the suction condition generally varies slowly, e.g. from a change in fluid level in tanks, but occasionally can vary rapidly due to switching of valves and suction lines. Since the Twin-Screw pump can tolerate transient cavitation conditions (**Figure 16**), slower response rates are preferable since the result is a more stable "Steady State" operation.

This cavitation control method has also been successfully applied to 3-Screw pumps at the author's company. Used in conjunction with the Dry running detection methods this offers a practical but simple method for 3-Screw pump protection.

CONCLUSIONS

The testing results demonstrate that the pressure based cavitation control technique can be used to protect Twin-Screw and 3-Screw pumps from adverse suction conditions. Since the method monitors the effect of cavitation inside the pump, the variations of suction piping losses due to viscosity, variation of vapor pressure between different fluids, static head variations due falling tank level or tidal variation when unloading ships can all be accommodated without extensive instrumentation or inputting additional parameters into the algorithm.

Selecting a smaller pump equipped with cavitation control is a viable alternative to selecting a large slow running pump to cover the "**edge**" cases, particularly when the edge cases represent a small percentage of the total pumping time. Pumps can therefore be selected to meet the majority of the operating conditions while relying on the cavitation control to manage the less frequent, adverse conditions.

The addition of the instrumentation for cavitation control can also provide additional protections due to the inadvertent closing of suction or discharge valves by providing over / under pressure trips/ alarms or pressure regulation.

Dry Running detection, while not required for pump protection on Twin-Screw pumps can provide a useful indication of the end of an unloading operation.

NOMENCLATURE

λ_C	= Loss coefficient for pump casing	(-)
λ_W	= Loss coefficient for pump impeller blade	(-)
β_1	= Fluid flow angle at impeller inlet	(degrees)
BB1	=Axially split double suction single stage centrifugal pump	
BEP	= Centrifugal pump Best Efficiency Point	
c_{1m}	= Meridonal component of absolute velocity at the impeller eye	(ft/s or m/s)

g	= Gravitational constant	(ft/s2 or m/s2)
H	= Pump differential head	(ft or m)
N	= Pump speed	(RPM)
nq	= Pump Specific Speed in metric basis units	(m3/s, RPM, m)
Ns	= Pump Specific Speed in US basis units	(USGPM, RPM, ft)
$NPSH_a$	= Net Positive Suction Head Available	(ft or m)
$NPSH_r$	= Net Positive Suction Head Required	(ft or m)
$NPSH_{rSF}$	= Net Positive Suction Head Required @ Shockless Flow	(ft or m)
P	= Pump absorbed power	(Hp or kW)
POR	= Preferred Operating Region for a centrifugal pump	
Q	= Pump flow	(USGPM or m3/h)
Q_{opt}	= Centrifugal pump Best Efficiency Point Flow	(USGPM or m3/h)
Q_{Sa}	= The shockless flow of an impeller	(USGPM or m3/h)
Q_{SF}	= The shockless flow of an impeller	(USGPM or m3/h)
w_1	=	

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