

# GUIDELINES TO MAXIMIZE RELIABILITY AND MINIMIZE RISK IN PLANTS USING HIGH PRESSURE PROCESS DIAPHRAGM PUMPS

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## ABSTRACT

A significant risk to production in chemical plants is the possibility of pump failures. The cost of the resulting downtime and unscheduled service can quickly exceed the initial cost of the pump. Prime areas for attention are proactive steps to increase reliability, simplify the installation, and optimize spare parts availability for fast repairs.

To increase reliability with reciprocating positive displacement pumps, their pulsating flow characteristic, the process, and the piping system must be considered together—not as separate items. Especially with this type of pump, risks can be eliminated right from the design stage with open-minded cooperation between the pump user and manufacturer.

To provide sufficient information for safe operation, and to demonstrate the benefit of a common understanding, different realistic failure scenarios from NPSH, system vibration, and slurry handling are discussed and solutions for improvement are presented, along with early failure diagnosis, online monitoring, and special pump items.

## INTRODUCTION

Hydraulically actuated diaphragm pumps are used in the broad range of 0.1 ml/h to about 100 m<sup>3</sup>/h and up to more than 3000 bar.

Additionally these pump types offer a number of important and often beneficial advantages, such as:

- Hermetically sealed pumping chamber
- Gentle conveyance
- High efficiency
- High reliability
- Absolutely safe dry running
- Internal safety valve (can provide a level of protection for the system when the pump is the only pressure generating device in the system)

These advantages indicate such pumps are to be used in high value production processes in the chemical, pharmaceutical, onshore/offshore oil, and gas industries. All too often, easily preventable mistakes are made in the application of these pumps. This can create expensive production losses and damage the reputation of this technology.

The essential difference in reciprocating pumps, as compared with continuous flow pumps, is the pulsating flow characteristic from the oscillating, volumetric displacement principle. Due to this property, these pumps have certain net positive suction head (NPSH) requirements, and are possible “vibration exciters” in the system. The noncontinuous conveyance behavior, coupled to and accelerating and decelerating the fluid columns in the connected pipeline, reaches far into the plant system.

Particularly problematical is when a plant engineer believes a reciprocating pump can easily be interchanged with another type of pump if the flowrate and pressure specifications are the same.

Nevertheless, plant engineers who develop a sensitivity and know-how about the special nature, benefits, and limitations of diaphragm pumps have a real opportunity to make important contributions in their plants, in improved reliability and reduced maintenance costs.

Therefore, a fundamental understanding of diaphragm pump internals is necessary, as well as how to avoid and/or minimize system vibration, how to optimize NPSH requirements, and how to design piping systems for slurries or high temperatures. The best approach is to start with basics, and discuss problems and solutions from real life examples. All the cases cited in this paper actually occurred within the past five years.

**BASICS OF RECIPROCATING POSITIVE DISPLACEMENT PUMPS**

The pulsation laws from the pump acting on the suction and discharge pipelines depend on the kinematics of the pump drive, the number of pump heads, and the elasticity of the fluid chamber and the fluid. Figure 1 shows the conveying characteristic generated by a harmonic crank drive with a plunger.

Actually, due to the elasticity of real fluids (during compression and decompression phases), the harmonic plunger kinematics and the pulsation kinematics acting in the pipeline are somewhat different (rapid velocity increases). This difference grows larger with reduced efficiency of the pump head (larger volumetric losses throughout the stroke between points 1 and 2), and pressure shocks, as shown in Figure 2, are generated.

*Flow Equation and Volumetric Efficiency*

The theoretical flow of a reciprocating positive displacement pump (RPDP) is:

$$\dot{V}_{th} = V_S n \tag{1}$$

However, due to the compressibility of the fluid in the pump chamber, the elasticity of the pump body, and a small flow loss during the valve closing, the real flow is smaller than  $\dot{V}_{th}$ . The loss

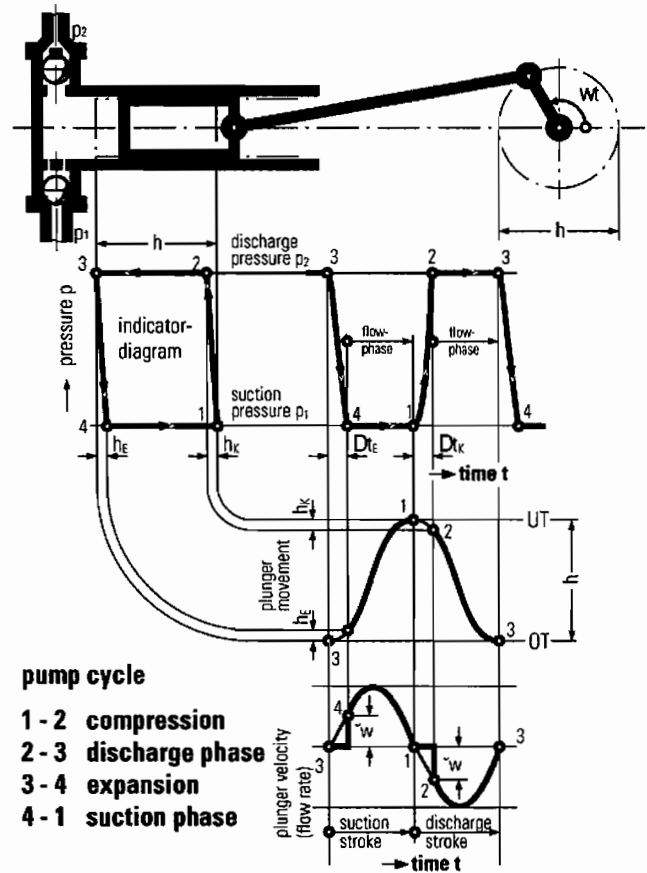


Figure 1. Harmonic Plunger Movement with a Crank Drive—Pressure Curve and Indicator Diagram.

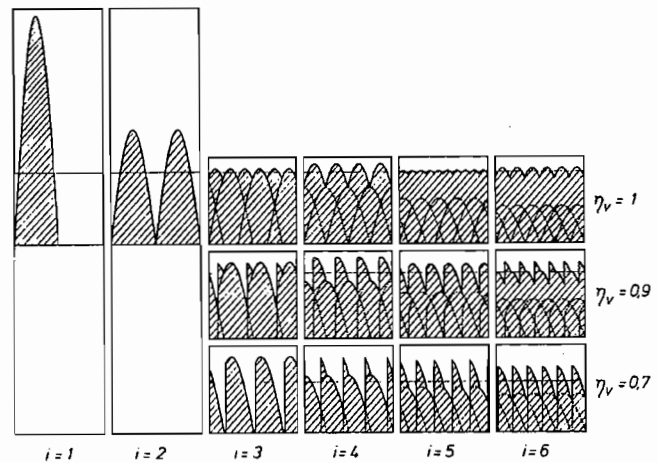


Figure 2. Flow Characteristics of a Different Number of Pump Heads (i) at Different Volumetric Efficiency Levels  $\eta_v$ .

in flow is included in the volumetric efficiency  $\eta_v$ . Therefore, the actual flow from a pump can be calculated by Equation (2).

$$\dot{V} = \dot{V}_{th} \eta_v = V_S n \eta_v \tag{2}$$

*Pressure Head Loss Due to Flow Friction*

The friction in the pipeline of an RPDP is a function of the flow velocity,  $w(t)$ . For noncompressible fluids,  $w(t)$  results from the kinematics law of the displacing plunger:

$$w(t) = \dot{x}(t) \frac{A_P}{A} \tag{3}$$

The friction head loss curve as a function of time can be calculated using the equation:

$$\Delta H_F(t) = \left( \frac{\lambda L_S}{D_S} + \sum \zeta_S \right) \frac{w^2(t)}{2g} \quad (4)$$

*Pressure Pulsation Due to Mass Acceleration*

The time-dependent acceleration and deceleration of the fluid in the pipeline also result from the displacement kinematics equation:

$$a_{\text{Fluid}}(t) = a_P(t) \frac{A_P}{A} \quad (5)$$

The acceleration head can be calculated based on the Newtonian law (Schlücker, et al., 1997):

$$\Delta H_A(t) = \frac{L_S}{g} \dot{w}(t) \quad (6)$$

*Pressure Shocks*

An additional aspect of acceleration head is pressure shock, resulting from the fact that the discharge and suction process always starts with a rapid increase in velocity ( $\Delta w$ ). According to Joukowski (1898), the acceleration head resulting from this is:

$$H_J(t) = \frac{a(t)\Delta w}{g} \quad (7)$$

*NPSH of Reciprocating Positive Displacement Pumps*

Figure 3 shows the values required for calculating the NPSHR (required) of RPDs. These are based on the principle used by pump manufacturers that all equations concerning the NPSH use a level height equal to the height of the pump suction flange as a reference level. To avoid cavitation that may be harmful, the suction head at the reference level must usually be at a specific height, to fulfill suction conditions. This NPSHA (available) value states how much the actual suction head of the installation is actually higher than the vapor pressure of the fluid conveyed. Pump installations must always meet the requirement  $\text{NPSHA} \geq \text{NPSHR}$ .

Continuous conveying pumps with even, smooth flow (i.e., ideal centrifugal pumps) have a pump independent NPSHA value and are easy to estimate characteristic values of the installation. The NPSHR value is a characteristic value of the pump, and is not a function of the installation ( $\Delta H_A = 0$  and friction value  $\Delta H_F = \text{constant}$ ).

In contrast to continuous conveying pumps, pulsating type pumps produce a complicated interaction with the pipeline system due to its kinematics. The liquid column in the pipelines coupled to the pump is accelerated and decelerated. NPSH values for RPDs, therefore, are usually estimated based on the minimum suction pressure required, due to the acceleration and friction pressure losses during the suction stroke (Figure 4).

The maximum friction pressure loss,  $\Delta p_{F, \text{max}}$  ( $\Delta H_{F, \text{max}}$ ), of RPDs occurs in the middle of the suction stroke (maximum of plunger velocity). The maximum acceleration pressure loss,  $\Delta p_{A, \text{max}}$  ( $\Delta H_{A, \text{max}}$ ), on the other hand, occurs at the beginning of the suction stroke. Superimposing both pressure loss types, the maximum total pressure loss, therefore, occurs always in the first half of the suction stroke. In the second half of the suction stroke, pressure is increasing, because the acceleration pressure loss becomes positive (deceleration of the plunger) and the friction loss is decreasing due the decreasing plunger speed. The transmission of a time-dependent pressure loss situation to the NPSH situation leads, of course, to also time-dependent NPSH curves, as shown in Figure 5.

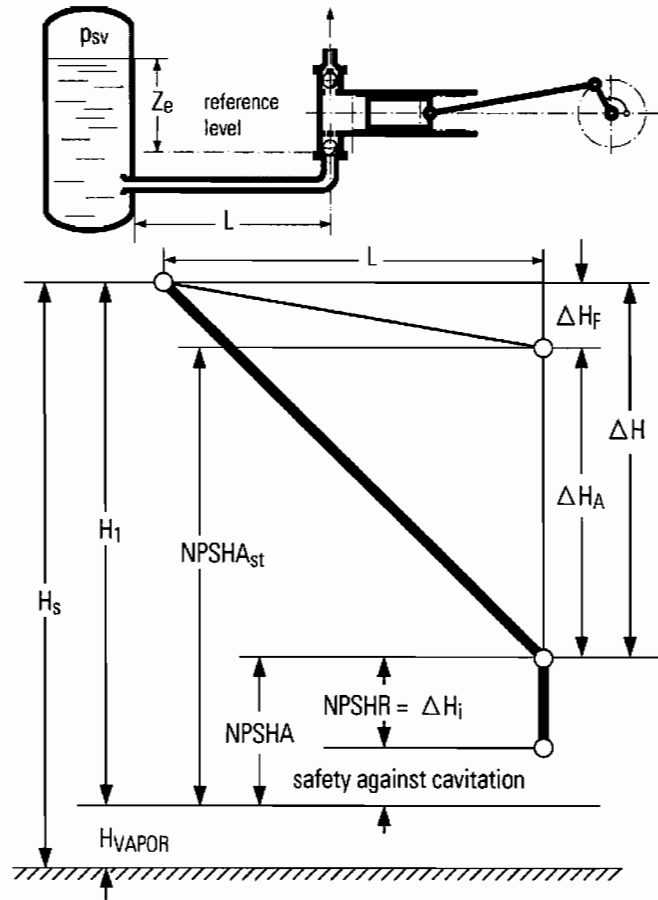


Figure 3. Suction Head Loss in the Suction Line Due to Flow Friction and Acceleration.

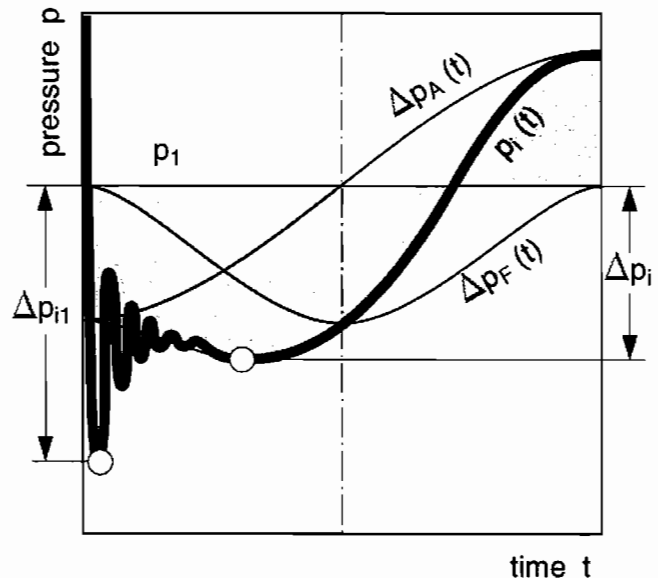


Figure 4. Pressure Loss Curves  $p_i(t)$  (Inlet Pressure Loss) in the Pump Chamber During Suction Stroke ( $\Delta p_F$ : Friction Pressure Loss,  $\Delta p_A$ : Acceleration Pressure Loss).

*Remark:* The amount of change (increase) of the NPSHA between suction beginning and end increases with increasing suction pipe length (with a short pipeline the  $\text{NPSHA}(t)$  is a horizontal curve).

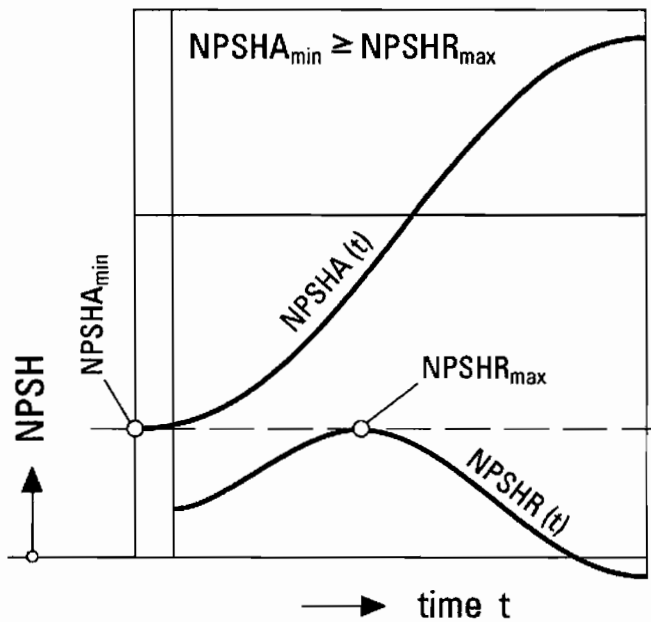


Figure 5. Usual NPSH Situation of a System with RPDPs with  $NPSHA_{min} = NPSHR_{max}$

As shown in Figure 5, it is common for an NPSH realization with RPDPs to use the maximum of  $NPSHR(t)$  and the minimum of  $NPSHA(t)$  following:

$$NPSHA_{min} = NPSHR_{max} \quad (8)$$

without any respect to the time-dependent curve characteristics. This, of course, results in a significant safety margin against cavitation. The longer the suction line, the higher the safety (compare section INSUFFICIENT NPSH).

This method is simple—but useful—because it is possible to determine NPSH values as dependent on the maximum flowrate, as a mean flowrate.

## HYDRAULICALLY ACTUATED DIAPHRAGM PUMP TECHNOLOGY

Diaphragm pumps are a type of reciprocating positive displacement pump (RPDP) with obvious convincing advantages (refer to INTRODUCTION).

In spite of their superior advantages, however, diaphragm pumps are very often only chosen as the last alternative. This may result from the misunderstandings that the technology is complex, and the effect of the pulsating conveying characteristic on a piping system cannot be determined easily.

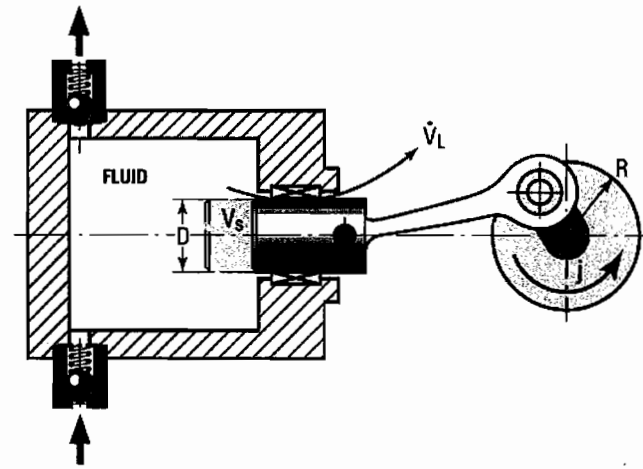
To assist pump users in understanding hydraulically actuated diaphragm pump technology, the evolution from the plunger pump to the diaphragm pump in five clearly defined steps is presented below, each step detailing the advantages and disadvantages of the development process. Important technical items and the evaluation of the reliability of diaphragm pumps are also included.

### Evolution of a Plunger Pump to a Hydraulically Actuated Diaphragm Pump

A central risk of all types of RPDPs is wear at the check valves. Depending on the experience of pump manufacturers, the risk may be small or large. Usually a high number of valve designs and material variations (plastics, hard metal, ceramics, etc.) are available. Depending on the fluid conveyed, valve lifetimes from a few weeks (abrasive fluids) up to several years are reached. Under usual conditions, two to three years is quite normal.

### Evolution Step 1—Plunger Pump

The simplest type of a reciprocating positive displacement pump is the plunger pump. The advantages, disadvantages, and risks of such a principle are shown in Figure 6.



Advantages	Disadvantages	Risks
<ul style="list-style-type: none"> <li>• pressure stiff characteristic</li> <li>• inexpensive</li> <li>• high efficiency</li> <li>• simple to use</li> <li>• widely used</li> </ul>	<ul style="list-style-type: none"> <li>• Plunger seal in fluid contact</li> <li>• wear at the plunger and plunger seal</li> <li>• leakage flow to environment or flushing fluid required</li> <li>• No internal safety valve</li> </ul>	<ul style="list-style-type: none"> <li>• pollution of the environment or flushing fluid disposal</li> <li>• high spare part cost: plunger and seal</li> <li>• safety valve in the system (high maintenance cost after opening)</li> </ul>

Figure 6. Evolution Step 1—Acting Principle, Advantages, Disadvantages, and Risks of a Plunger Pump ( $V_s$ : Stroke Volume).

### Evolution Step 2

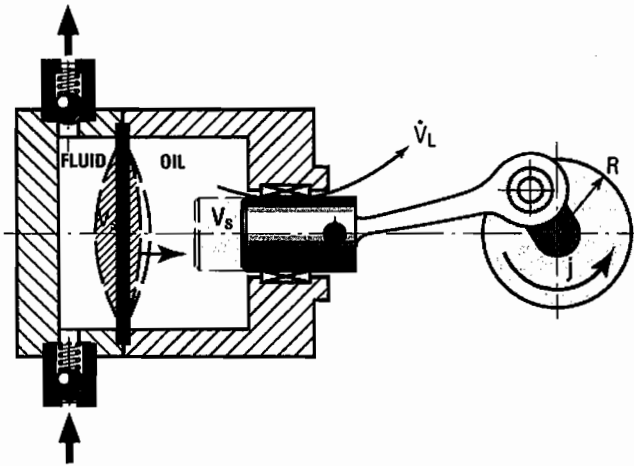
A hydraulically acting diaphragm pump is essentially a plunger pump with a “membrane” that separates the discharge chamber to a process fluid chamber and a hydraulic fluid chamber (Figure 7). Membranes, or “diaphragms,” are usually highly flexible. Under normal operation, the pressure on both sides of the diaphragm is nearly equal (difference of about 1 psi or less). The volume displaced by the plunger is transmitted via hydraulic fluid through the diaphragm to the process fluid.

### Evolution Step 3

- To contain the leakage flow  $\dot{V}_L$ , a reservoir is required (Figure 8), and
- To avoid overstretching of the diaphragm, a back support plate is required.
- Due to the addition of the support plate, with the leakage flow  $\dot{V}_L$ , the diaphragm meets the support plate before the plunger has reached the end of the suction stroke. This leads to a pressure drop and oil-cavitation. If cavitation happens on the oil side, we have the possibility of erratic flow. To avoid this, the  $\dot{V}_L$  loss must be replaced. So, this mandates to the requirement for a replenishing valve that, due to the pressure drop in the hydraulic cylinder, opens and allows replenishing of the oil volume at the end of the suction stroke.
- But when the suction line is closed, the replenishing valve acts like a suction valve. Overfilling of the oil chamber and overstretching of the diaphragm to the front is possible.

### Evolution Step 4

- To avoid overstretching of the diaphragm, a front support plate is required (Figure 9). Nevertheless, due to the front support plate,



Advantages	Disadvantages	Risks
<ul style="list-style-type: none"> <li>plunger seals sees only oil, no abrasion on the seal or pollution from leakage</li> <li>hermetically tight process fluid chamber</li> <li>absolutely safe if run dry</li> <li>high efficiency</li> <li>gentle conveyance</li> </ul>	<ul style="list-style-type: none"> <li>due to the leakage flow <math>\dot{V}_L</math>, the diaphragm moves step-by-step rearwards and is doomed to get over stretched (no replenishing action)</li> </ul>	<ul style="list-style-type: none"> <li>cannot be made into a truly useful pump</li> </ul>

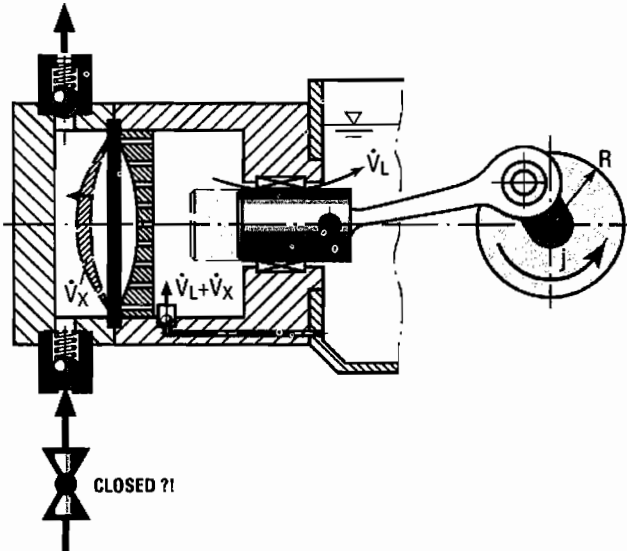
Figure 7. Evolution Step 2—A Deflectable Diaphragm Separates the Pump Chamber into a Hermetically Sealed Fluid Chamber and a Hydraulic Chamber.

when the suction line is closed and the replenishing valve opens (instead of the suction valve), after some strokes the diaphragm is pushed to the front support plate before the plunger has reached its most forward position. On the next stroke, the pressure will rise until the diaphragm is perforated (ruptured) through the small support plate holes, or the reciprocating drive will fail in overload. So, to avoid this, an adjustable pressure relief valve (PRV) on the hydraulic side is required. Then, if the pressure rises over the preset limit, this valve opens and relieves the surplus oil back to the reservoir. This valve is also useful when the discharge line is closed (Figure 10) or as a system safety valve when the pump is the only pressure generating device. Now the hydraulic diaphragm pump contains an important part of the system.

*Remark:* Depending on the diaphragm material and the service temperature, the pressure limit will be reached very quickly. For example, with a 1 mm thick PTFE diaphragm at 300°F with a channel diameter of 2 mm, the pressure limit is about 700 psig to 1120 psig. With metal diaphragms, the pressure can be much higher. With PTFE diaphragms at higher pressure, perforated support surfaces are always a risk.

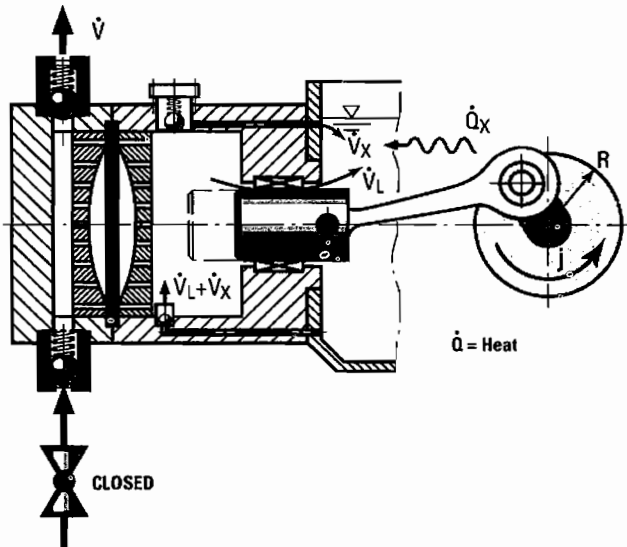
Evolution Step 5

- To overcome the disadvantages of the Step 4 design, a new basic design is required.
- Instead of the front support plate, a “gate valve” is used (Figure 11). This normally spring loaded valve keeps the channel to the replenishing valve closed, until the diaphragm travels back to the rearmost position and mechanically pushes the gate valve open. Only when the diaphragm is in this position can replenishing occur. Therefore, the hydraulic system cannot overfill and the diaphragm cannot get overstretched. Additionally, the special design of the gate valve surface touching the diaphragm, allows the back support plate to be eliminated.



Advantages	Disadvantages	Risks
<p>this design is already used in a real pump, but only when it can be guaranteed that the pressure in the pump never drops under a certain limit.</p>	<p>the replenishing valve is simply a small suction valve. A closed suction line, or any pressure drop below the preset pressure limit of the replenishing valve, and this valve opens.</p>	<p>when pressure drops below atmospheric, the diaphragm gets overstretched due opening of the replenishing valve (overfilling)</p>

Figure 8. Evolution Step 3—Additional Back Support Plate and Replenishing Valve.



Advantages	Disadvantages	Risks
<ul style="list-style-type: none"> <li>really useful and safe pump. Diaphragm is safe when suction line or discharge line is closed.</li> <li>pressure relief valve can act as one of the safety valves protecting the system (in some cases)</li> </ul>	<ul style="list-style-type: none"> <li>diameters of the hole in the support plates limits the pressure the diaphragm can take when pressed against it</li> <li>the higher the pressure, the smaller the diameter required for holes, and thus the more holes neces sary.</li> </ul>	<ul style="list-style-type: none"> <li>when pressure exceeds limits, diaphragm can get damaged when pressed against the support plates.</li> <li>the holes in the support plate are often small and can get clogged by particles</li> </ul>

Figure 9. Evolution Step 4—Additional Front Support Plate and a Pressure Relief Valve (With Closed Suction Line— $V_x$ : Output of PRV;  $Q_x$ : Heat Input Due Relief).

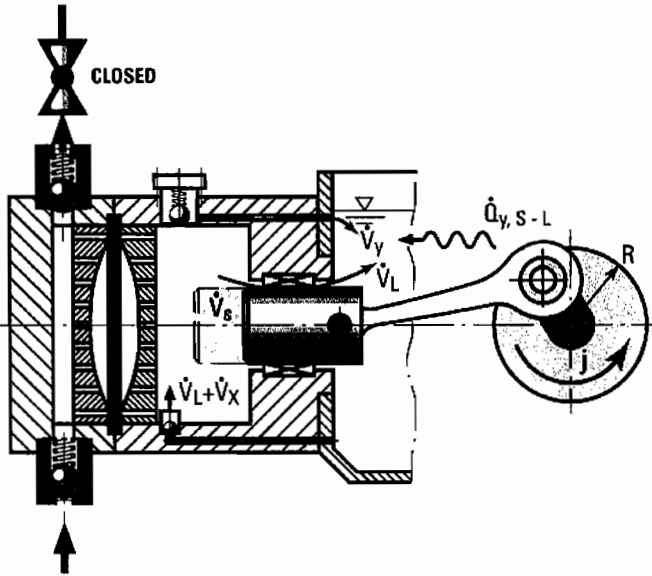
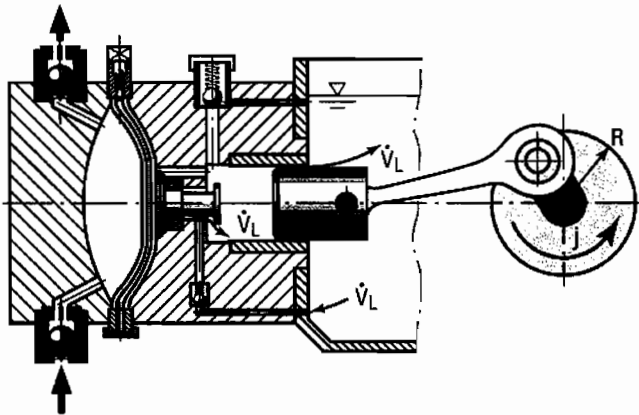


Figure 10. Evolution Step 4 with Closed Discharge Line ( $V_x$ : Maximum Replenishing Volume and Output Volume after the First Relief Stroke;  $Q_y$ : S-L Heat Input Sniffling Minus Leakage;  $V_y$ : First Output Volume through PRV. Advantages, Disadvantages, and Risks (Figure 9)).



Advantages	Disadvantages	Risks
<ul style="list-style-type: none"> <li>hermetically tight</li> <li>high efficiency</li> <li>gentle conveyance</li> <li>absolutely safe if run dry</li> <li>internal safety valve can be useful as one system safety valve</li> <li>free-flow fluid chamber</li> </ul>	<ul style="list-style-type: none"> <li>higher first cost compared to plunger pumps</li> <li>more complex-needs a higher level of understanding of how the pump functions</li> </ul>	<ul style="list-style-type: none"> <li>increased complexity can increase probability for failures</li> <li>greater number of different possible problem areas</li> <li>more spare parts to stock</li> <li>some risk that certain spare parts may not be readily available</li> </ul>

Figure 11. Evolution Step 5—Diaphragm Pump Design with a Diaphragm Position Control System.

Special Features of Evolution Steps 4 and 5

• When a discharge line is closed (pump deadheaded), a circulation of oil is generated (hydraulic chamber→pressure relief valve→reservoir→replenishing valve→hydraulic chamber...). This means the hydraulic energy generated by the pump must be fully dissipated by the pressure relief valve. Dissipation of energy always means a transformation to heat ( $Q$ ). Therefore, the hydraulic oil gets heated. The more oil that is circulated, the faster

the temperature rise. This must be considered especially with bigger pumps.

• To keep the heat rise as low as possible, the replenishing valve is usually sized much smaller than the suction valve to restrict the oil flow, and the inlet pressure loss is also larger (due to the spring). Thus, the replenishing valve restricts the recirculated flow to much less than the whole stroke volume. Therefore, cavitation occurs in the oil during oil circulation.

• Mineral oil used as a hydraulic oil usually contains a lot of dissolved air. When the pressure drops down to about 4 psia, the air comes out of solution (→air bubbles). This process occurs mostly without significant pressure shocks, noise, or damage of parts, and the bubbles act like a damper. The only disadvantage is that the dissolving process of the air bubbles back into the oil takes much more time than the opposite process. After stopping the circulation, the flow from the pump will be greatly reduced due to the high compressibility of these bubbles, and it would take a long period of time until the pump will again convey the required flow. To shorten this process and to also eliminate other sources of air bubbles, a venting valve is required. This valve expels the gas bubbles from the top of the hydraulic chamber by dispensing a small but constant amount of oil (or air plus oil) every stroke back to the reservoir.

• NPSH of hydraulic diaphragm pumps is limited by the pressure at which the air in the oil comes out of solution.

Safety Philosophy—Sandwich Diaphragm Technology

One of the main reasons to use diaphragm pumps (Figure 12) is that they are hermetically tight. To guarantee this even when diaphragm failures occur, the technology usually chosen is that of a double barrier. The concept is that when one barrier fails, the other still maintains safe operation for a certain period of time.

The double barrier technology in diaphragm pumps can be realized by a sandwich or a double diaphragm. Because of the advantages, the sandwich technology is mainly used.

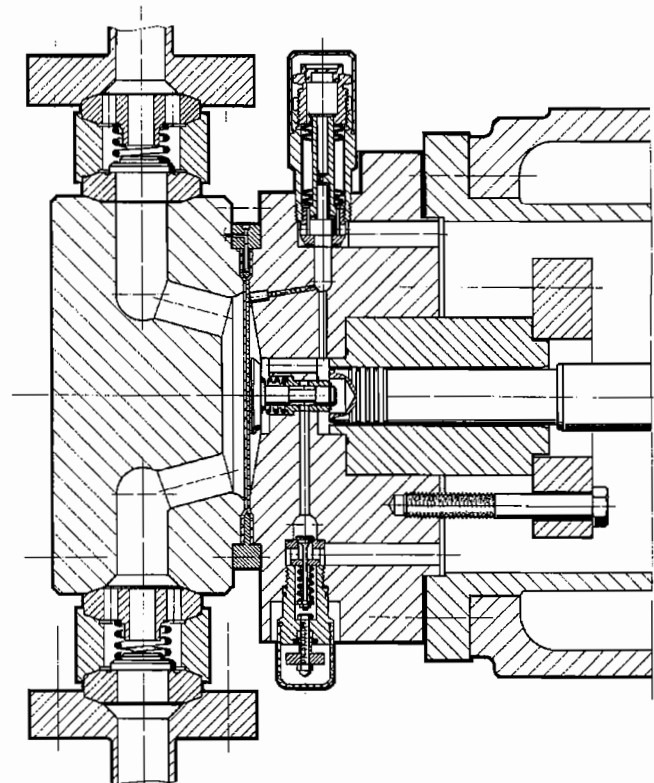


Figure 12. Design of a High Pressure Diaphragm Pump Head with PTFE Diaphragm.

Sandwich diaphragm technology is characterized by two diaphragms in close contact (Figure 13, positions 1 and 2) assembled into an intermediate ring (5). The intermediate ring has a channel system that is connected to small bores in the diaphragm cover, leading to the inlet and outlet. The channel system is completely filled with an oil or fluid compatible with the process fluid. Because of their compressibility, air bubbles must be avoided. After filling, the diaphragms get pressed together by the pump pressure during the beginning of operation, and most of the oil will be expressed from the outlet. Only a tiny volume of oil in a very thin layer remains between the diaphragms for lubrication and to "stick" the diaphragms together by molecular adhesion to act as one.

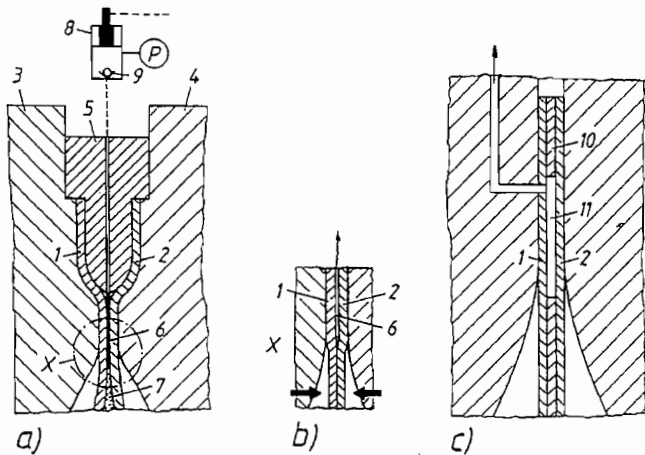


Figure 13. Sandwich Diaphragm Assemblies with Rupture Indicator. (1, 2: Diaphragms; 3, 4: Pump Parts; 5: Intermediate Ring; 6: Inlay/Gaze; 7: Intermediate Oil; 8: Rupture Indicator; 9: Nonreturn Valve; 10,11: Intermediate Diaphragm with Metal Diaphragm Assembly.)

If a diaphragm perforates, process fluid (or hydraulic oil) enters the space between the diaphragms and reduces the hydraulic coupling of the two diaphragms. Oil or fluid passes to the diaphragm monitor. The pump user receives an alarm signal and can prepare for an orderly shutdown and maintenance (because the pump can act safely with one diaphragm for several days).

#### ABOUT RELIABILITY AND COMPLEXITY OF HYDRAULIC DIAPHRAGM PUMPS

For many applications, the environmental benefits of a diaphragm pump (sealless, leak-tight) have turned more and more into a must for users. Together with the other convincing advantages, diaphragm pumps are gaining wider and wider acceptance. In many cases, diaphragm pumps are the only approach for a safe, clean, and reliable process.

When discussing its reliability and its dependence on complexity, it is useful to divide the complexity of diaphragm pumps into the liquid end and the hydraulic area. The liquid end, of course, is absolutely equal to plunger pumps (exceptionally, the often problematic plunger packing and the plunger material). The hydraulic area, as an additional feature, guarantees a self protection of the pump due to the pressure relief valve, and an option for the pump's PRV to be part of a plant protecting system. This is given with a certified pressure relief valve and when the diaphragm pump is the only pressure generating device in the system. A diaphragm pump, therefore, already contains an important safety component of the system. Due to that, the right reliability comparison should be based on pump and protection device (safety valves).

The safety valves used in the fluid pipeline are very often problematic, and should be changed when opened (popped) only

once. Conversely, the integrated pressure relief valve in diaphragm pumps can usually act as often as necessary without any meaningful wear. This is, of course, an important plus in reliability.

Also, all the other hydraulic valves (gate valve, venting valve, replenishing valve), and all other moving parts (plunger and plunger seals) on the oil side always operate under nonabrasive and relatively controlled conditions. As long as these assemblies are properly designed (compare Vetter, et al., 1995b), and the diaphragm keeps the process fluid contained, any problems will be small.

Therefore the central point of risk, besides some differences between various manufacturers concerning their hydraulic valves (Table 1), seems to be in the diaphragm! It is the most important task for the manufacturer to deliver a safe technology in the diaphragm. Reliability improvements in that area have already reached a high level. Nevertheless, there are still differences in the details. An old saying states: "The devil is in the details!" A special problem for the pump user in that concern is to assess the design quality and to recognize the small, but very important differences that determine long, happy service life...or short service life.

Table 1. Quality Differences in the Hydraulic Valve Designs.

Hydraulic valve type	Quality differences
Replenishing valve	Adjustable vs. non-adjustable. Today, the technology allows for a fixed valve setting safely. Advantage: No mistakes, no tampering.
Venting valve	Volumetrically action vs. pressure-dependent action. Volumetric is much safer and more accurate.
Gate (control) valve	Different designs with quite similar properties (see Vetter et. al. 1995)
Pressure Relief Valve	High stroke types (amplifying effect) guarantee a higher safety at low discharge pressure. A more proportional acting type is more suitable at high head-high flow as the pressure shocks are smaller. Hardened sealing parts necessary.

#### Safe and Reliable Diaphragm Technology

The diaphragm is the central element that determines the entire pump head design and its size. PTFE diaphragms are predominantly used, while for higher pressures, metal diaphragms are common, and for special designs there are elastomeric diaphragms (Table 2).

Table 2. Diaphragm Material and Range of Applications. (Key—++: Resistant Against Almost All Fluids; -: Resistant in a Few Cases; V: Deflection; D: Main Dimensions of the Pump.)

DIAPHRAGM MATERIAL	PTFE	METAL	ELASTOMER
Realized pressure range	to 350 bar (5,000 psi)	2000 bar (29,000 psi)	to 1000 bar (14,500 psi)
Temperature range possible	- 20 to 150°C (- 4 to 300°F)	- 20 to 200°C (- 4 to 400 °F)	5 to 80 °C (40 to 180 °F)
Chemical stability	++	++	-
Material-Effect on design	V large, D small	V small, D large	V large, D small
Diaphragm shapes used	plane parallel & corrugated	plane parallel	plane parallel

The important difference between PTFE and metal is the higher displacement ability of the PTFE material, which leads to about eight times higher displacement intensity. With the same flowrate requirements, the dimensions of a pump head using PTFE diaphragms are much smaller, compared to metal, and, therefore, pump heads using PTFE diaphragms are much less costly.

With PTFE diaphragms, usually more than 8000 hours service life are reached. Depending on the design, more than 20,000 hours can be reached. For such high lifetime, not only the diaphragm design and deflection but also the clamping is important.

The clamping geometry contains an important part of the know-how developed in the hydraulic diaphragm technique. The higher the pressure or the bigger the stroke volume, the more important it is to design a clamp acting like an absolutely static sealing.

#### Safe Rear Support

The diaphragm in a hydraulically acting diaphragm pump is usually pressure balanced (with the exception of the clamping

area). In a stable and normal pumping process, the diaphragm is deflected between a position close to, but not at, the rear support surface and a front position, depending on the stroke volume. This range of movement is stable as long as the discharge valve is tight. However, it is not unusual for the discharge valve to develop a small leakage flow during the suction stroke. If this happens, one of the most important details for safe diaphragm operation with PTFE at high pressure is a full and absolutely gap-free rear support (Figure 12). The safest solution is a disk shaped front to the gate valve fitting exactly into the backup plate, forming a gap and edge-free support surface. If the rear support contains bores and the diaphragm is pressed against it with discharge pressure, there is a high risk of damage or failure of the diaphragm.

#### Users Influence on Reliability with Diaphragm Pumps

A safe design is not the only factor in reliability. Competent handling by the user based on knowledge about diaphragm pump internals and interactions with the peripherals is needed. The examples below are given to show some important plant aspects based on actual scenarios that help to point out that: "When a hydraulic diaphragm pump is treated as required, the reliability of such pump types can clearly exceed the reliability of alternative pumps."

### SYSTEM DESIGN CAN CAUSE DIAPHRAGM RUPTURES

*Scenario:* A pump user complained about diaphragm rupture after restart of the pump using a (control) gate valve design.

Diaphragm pumps are hermetically sealed, since the fluid is always contained in a chamber surrounded by static seals (diaphragm clamping area). In spite of this, diaphragm pumps also have plunger seals that can allow a small leakage flow. Depending on the direction of the pressure slope, a small amount of oil can flow from the oil chamber to the reservoir (standard situation during operation), or from the reservoir to the hydraulic chamber. In modern, reliable, and safe diaphragm pumps, the latter possibility can only occur under special situations in combination with a pump and a plant out of service (vacuum on the suction side or siphoning).

The malfunctioning system contained a process at an elevated temperature. The suction line looks as shown in Figure 14. Between valve three and pump head one, there was a significant amount of fluid  $V_{SP}$ . When the process stops, the procedure of Figure 15 can lead to diaphragm damage.

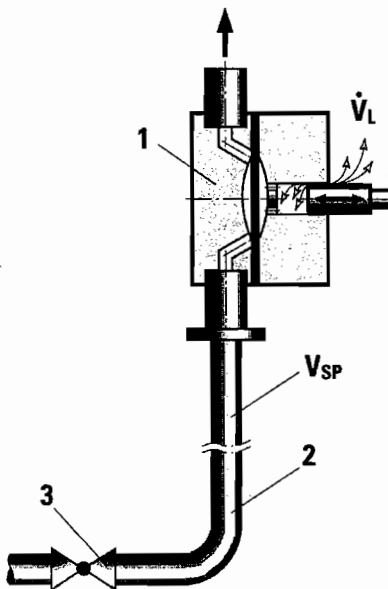


Figure 14. Suction Pipe System ( $V_{sp}$ : Volume in the Suction Pipe).

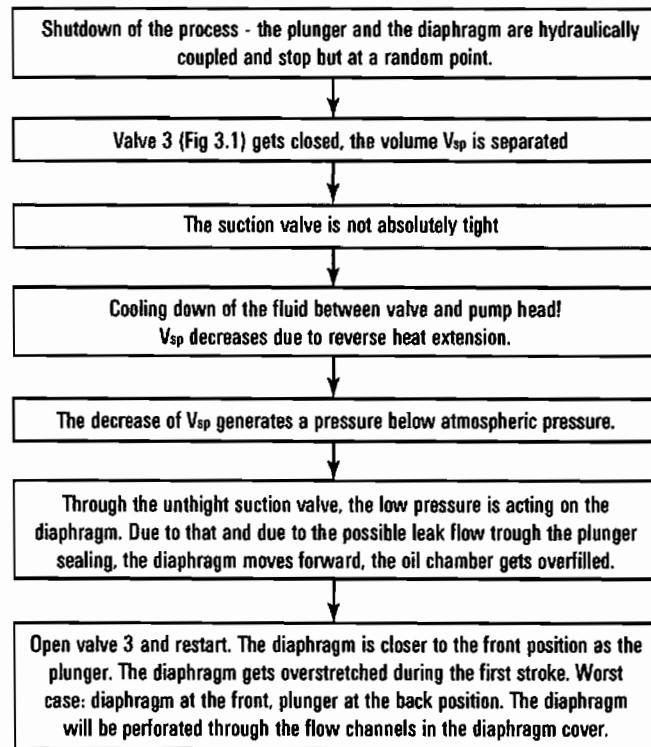


Figure 15. Procedure Leading to Diaphragm Rupture Due to Cooling Down a Closed Suction Line.

Low pressure driven leakage flow from the reservoir to the hydraulic chamber is absolutely unavoidable by the pump manufacturer, as a dynamic seal (e.g., the plunger seal) can never be made 100 percent leak-tight. Normally, there is only a difference in leak flow between the different types of plunger seals used. The only chances therefore are:

- Avoid the contraction of volume  $V_{SP}$  by an always positive pressure at the suction flange, or
- Include a vacuum support in the pump head as shown in Figure 16 (c), or
- Complete the pump head with a diaphragm positioning system (Figure 16 (b)) acting as follows: before startup, via the positioning valve, a channel to the reservoir is opened. A positive pressure moves the diaphragm to its back position (needs a possibility to increase suction pressure over atmospheric pressure before restart).

The vacuum support is simply a perforated disk with a number of flow channels, as already discussed in Evolution Step 4.

At high pressures with a PTFE diaphragm, the channels (bores) must be very small. This is very expensive and the risk of the channel becoming plugged is appreciable. The front support plate is, therefore, not useful for conveying suspensions or slurries.

### INSUFFICIENT NPSHA

*Scenario:* Testing different pumping techniques in a high value chemical research process produced only a lot of unsatisfactory experiences, such as:

- Crystallization at the plunger seal in a plunger pump,
- Crystallization at the bypass control valve used with a small centrifugal pump,
- Crystallization at the clearances in various rotary positive displacement pumps, and
- Insufficient metering accuracy in all of the above cases.



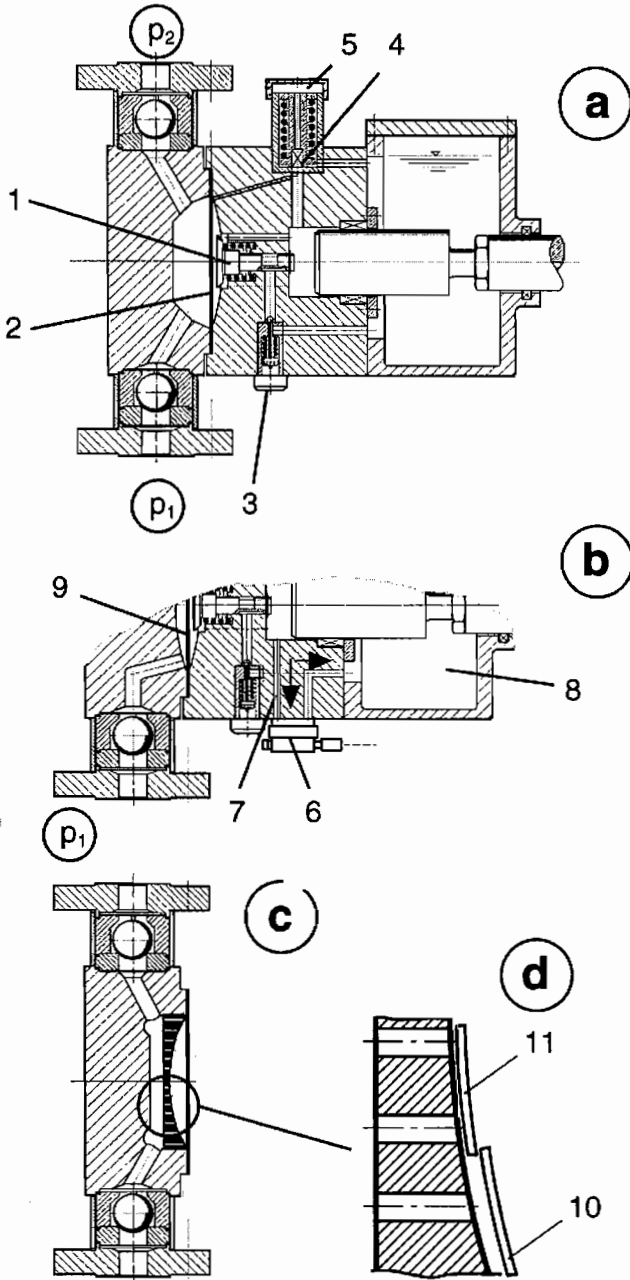


Figure 16. Pump Possibilities to Avoid Diaphragm Damages Due to Volume Increase in the Oil Chamber. (a: Standard Pump Head, b: Diaphragm Positioning System, c: Vacuum Support Plate, d: Diaphragm (11) Pressed Against Front Support.)

A hydraulically actuated diaphragm pump seemed to be the last alternative, but it had to achieve a sum of demanding properties like the following:

- Margin over vapor pressure at the suction flange only 0.24 bar (3.48 psi), fixed by the fluid preparation process. No chance to increase.
- The fluid temperature should be  $152^{\circ}\text{C}, \pm 2^{\circ}\text{C}$ . Below  $150^{\circ}\text{C}$ , crystallization will begin, and above  $154^{\circ}\text{C}$ , the pump user fears fluid will vaporize. Vaporization of fluid in the pump head would produce erratic or reduced flow.
- High metering accuracy and long service life
- Gentle conveyance required as the fluid is shear sensitive and crystallizes when sheared

### Pump Head Design Aspects

Due to the high temperature and the narrow allowable range of only  $4^{\circ}\text{C}$ , the pump head needed a special design: to be completely encased in a heating jacket. To be sure that the temperature in the pump is always in the permissible range—even when the pressure relief valve was acting—some additional features were included, and temperature sensors for controlling were assembled very close to the fluid and oil chambers. In addition, the entire pump head was packed in an insulation cover. To avoid crystallization, a medium pump speed was chosen.

### Remaining Problem

After optimizing the pump design as much as possible, there still was at least one problem open. Due to the need for a higher NPSHA compared with other pump types, the suction condition was just at the borderline, with a high probability of cavitation. An improvement of the plant would be too expensive, and would unacceptably delay the investigation schedule. Because they had already experienced crystallization with different pumps, a booster pump was not a possibility. The only approach was to live with some partial cavitation.

### Partial Cavitation as a Chance to Increase NPSHA

In a normal NPSH application, displayed in Figure 5, it is clearly seen that during the whole suction stroke there is always a safety margin between the  $\text{NPSHA}(t)$  and the  $\text{NPSHR}(t)$  curve, meaning this pump installation meets the basic rule:

$$\text{NPSHA} > \text{NPSHR} \quad (9)$$

Yet Figure 5 also shows there is still a possibility to increase the NPSH by allowing a shift in the  $\text{NPSHA}$  curve downward, until it meets the  $\text{NPSHR}$  curve ( $\text{NPSHA} = \text{NPSHR}$ , Figure 17).

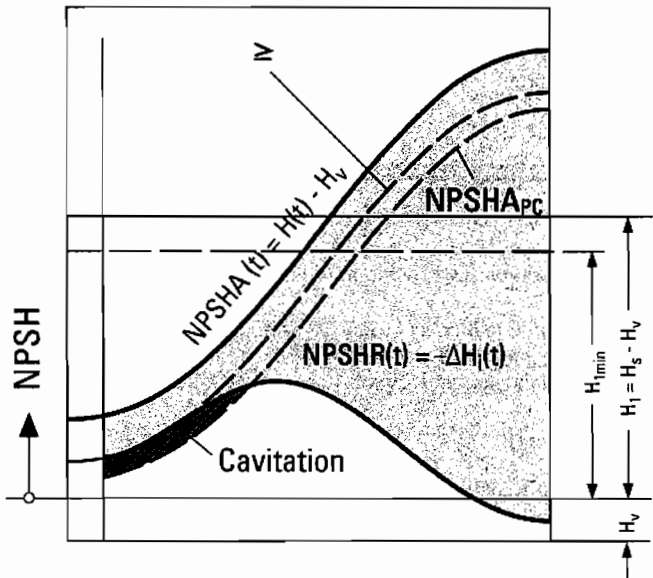


Figure 17. NPSHA and NPSHR Curves of a RPDP without and with Partial Cavitation ( $\text{NPSHA}_{PC}$ : NPSHA with Partial Cavitation).

It is recognized that the safety margin ( $\text{NPSHA}(t)$  to  $\text{NPSHR}(t)$ ) against cavitation increases with the increasing suction stroke. Due to that behavior, cavitation in RPDPs is much different from continuous flow pump types. If a small amount of cavitation occurs ( $\text{NPSHA}_{PC}$ ) by an additional small downshift of the  $\text{NPSHA}$  curve, it occurs only at the beginning of the suction process; until that moment, the  $\text{NPSHA}$  curve exceeds the  $\text{NPSHR}$  curve. It is

possible for partial cavitation to only appear with pulsating pumps. Unlike normal cavitation, it *does not lead to flowrate loss* as long as back formation of cavitation is short.

This leads to the question of whether partial cavitation is acceptable when it does not cause damage in the pump and system.

*Remark:* The German VDMA and the EUROPUMP Technical Commission decided to accept an NPSH law (Equation (10)) for pulsating pumps using an  $NPSHA_C$  for continual flow and a  $\Delta H_P$ -factor considering all influences coming from pulsations. Important in that regard is that the pump manufacturer is completely responsible for this  $\Delta H_P$ -factor, that is, whether he will accept partial cavitation or not. The only exception is when the pump user deals with sensitive fluids.

$$NPSHA = NPSHA_C - \Delta H_P \quad (10)$$

#### Mathematical Theory of Partial Cavitation

The thought behind this type of cavitation is that when cavitation with RPDPs occurs, the gas bubbles due to cavitation will first be generated in the pump head (Schlücker, et al., 1997). As long as gas bubbles are in the pump head, the liquid column in the suction pipe is decoupled from the plunger or diaphragm (Figure 18). The pressure difference between pump head and suction vessel, therefore, is:

$$\Delta p_x = \Delta p_{sv} - \Delta p_v \quad (11)$$

As long as Equation (11) is valid, it happens during cavitation that only the constant pressure  $\Delta p_x$  is acting on the liquid column in the suction pipeline. Due to that, the suction liquid column gets accelerated and can catch up with the plunger again (back formation of the cavitation), depending on the amount of cavitation.

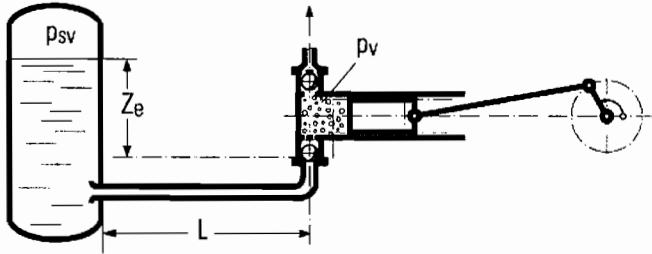


Figure 18. Beginning Cavitation in a System with RPDPs.

Based on Equation (11) and harmonic pump kinematics, the following equation set was developed to calculate such procedures with partial cavitation. If  $\Delta p_x$  is constant, then also:

$$H_1 = H_S - H_V = \Delta H_F + \Delta H_A + \Delta H_i = C = \text{const.} \quad (12)$$

Applying Equations (4), (6), and the internal pressure loss  $\Delta H_i$  of Equation (12) results in the differential Equation (13), which defines the progress of flow during cavitation.

$$Aw^2(t) + Bw(t) = C \quad (13)$$

with:

$$A = \frac{1}{2g} \left( \frac{\lambda L_S}{D_S} + \zeta_S + \zeta_V \right) \quad (14)$$

$$B = \frac{L_S}{g} \quad (15)$$

For the boundary conditions of a single cylinder pump, the solution of Equation (13) is:

$$w(t) = D \frac{1 - e^{-at}}{1 + e^{-at}} \quad (16)$$

and transformed to the flowrate:

$$\dot{V}(t) = A_S w(t) = A_S D \frac{1 - e^{-at}}{1 + e^{-at}} \quad (17)$$

with:

$$D = \sqrt{\frac{C}{A}} \quad (18)$$

$$a = \frac{2}{B} \sqrt{\frac{C}{A}} \quad (19)$$

By integration of Equation (17), the inlet column into the pump's operating chamber under cavitation can be calculated.

$$V(t) = A_S D \left[ t + \frac{2}{a} \ln \left( \frac{1 + e^{-at}}{2} \right) \right] \quad (20)$$

Curves according to Equations (17) and (20) are shown in Figure 19.

The suction velocity curve resulting from cavitation during suction stroke is qualitatively shown in Figure 20. Should the requirement for complete regress of any cavitation until the end of suction stroke exist, the following Equation (21) describes the velocity-time curve.

$$\int_{t_3}^{t_1} \dot{V}(t) dt = \eta_V V_h = \text{const.} \quad (21)$$

This results in the area under the curve II being equal for all operation conditions. It is remarkable that with increasing cavitation, the rapid velocity decrease at the point of back formation of the cavitation (III), gets more significant, and occurs more closely to the end of the suction stroke.

The experimental verification in Figure 21 proves that this theory matches the reality. When the rapid velocity decrease  $\Delta w$  occurs (when the liquid column in the suction line catches the plunger again), the calculated and measured curves look quite similar.

#### Harmful or Harmless Partial Cavitation

A sum of investigations has shown that, in most cases, partial cavitation was harmless. It seems that regress of cavitation in oscillating pumps occurs, mostly, not close enough to the wall so that damage is possible. This can especially be stated when the vapor energy of the conveyed fluid is small (research is still going on). More important, from an actual point of view, is that the rapid velocity drop produces a significant pressure shock, as shown in Figure 22.

It is logical that the size of the pressure shock must be influenced by the physical liquid data, and the pump kinematics must be in a connection to the Joukowsky shock, but uninfluenced by the discharge pressure. Some measurements at a medium pump size with different rapid velocity decrease sizes ( $\Delta w$ ) delivered the pressure shock data given in Figure 23.

If  $\Delta p_p$  is greater than the discharge pressure, a result of partial cavitation can be an overload of the crank case, as it would happen in a situation shown in Figure 22.

#### Solution of the High Temperature Diaphragm Pump Application Problem

After offering the pump user the possibility of partial cavitation as a way to increase NPSH, he first hesitated, but then he checked

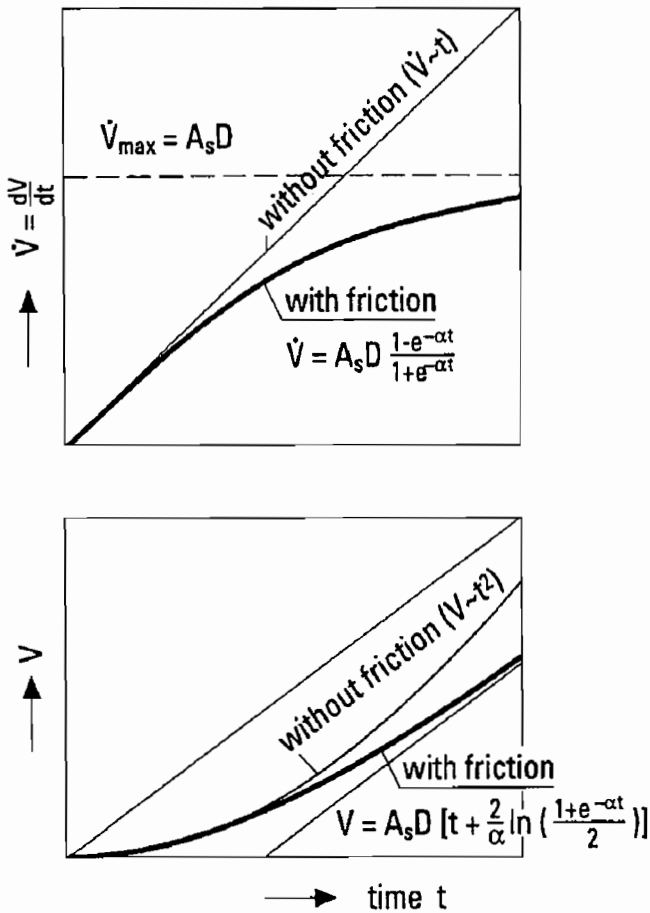


Figure 19. Realistic Flow Line Curve with Cavitation in the Pump Head.

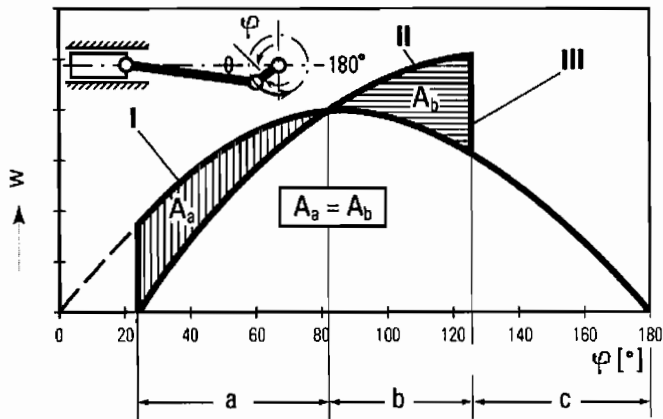


Figure 20. Theoretical Flow Behavior with Partial Cavitation. (I: Without Cavitation; II: With Cavitation; III: Back Formation of the Cavitation Due to a Velocity Drop.)

if cavitation would harm his fluid. The answer was “No!” The second step was the estimation of the  $\Delta p_p$ . The result was that the crankcase and bearings were not in any danger of being overloaded. Additionally, the vapor energy of the fluid was medium, compared with water. The risk of damages due to cavitation was negligible. The decision to try it was based on this sum of positive answers. The result was impressive—several thousands of hours service life without any damage or failure.

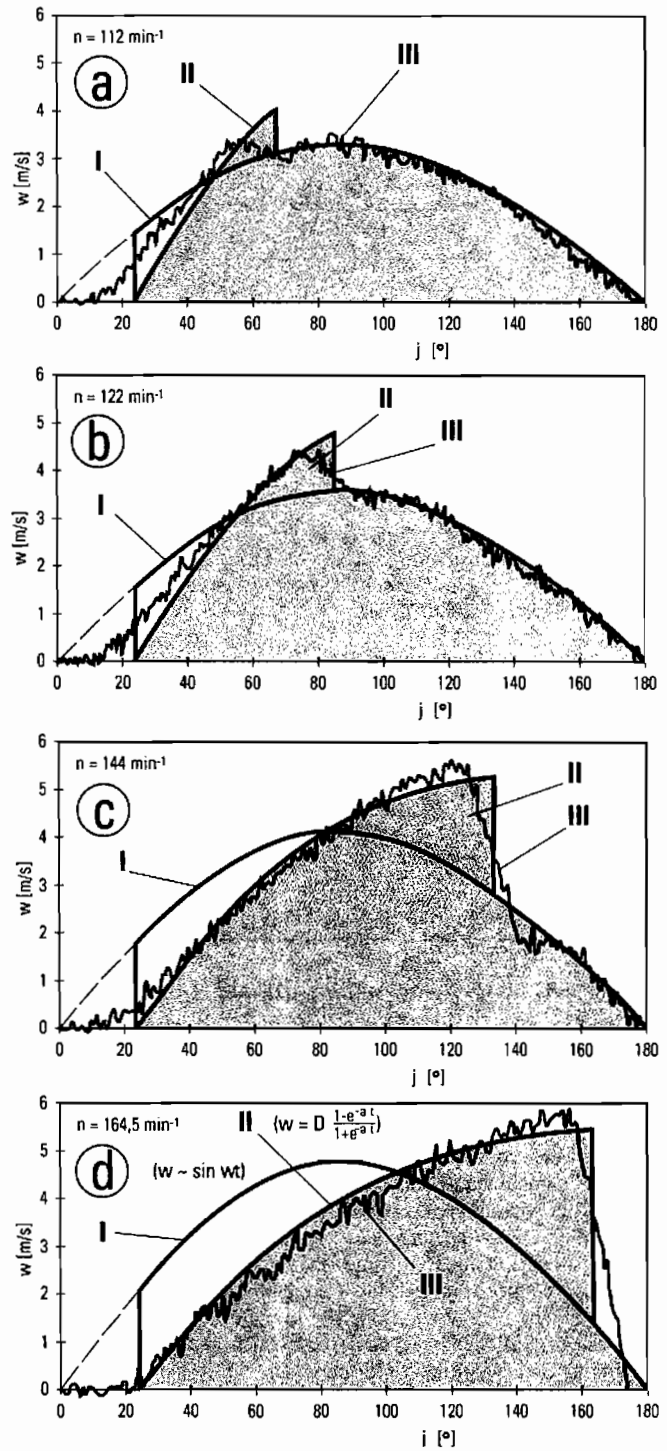


Figure 21. Comparison of the Theoretical and Measured Inlet Flow Velocity at Different Pump Speed. (I: Curves without Cavitation; II: Theoretical Curves with Cavitation; III: Measured Curves with a Similar Level of Cavitation.)

## PIPELINE VIBRATION

*Scenario:* A new plant was built with a triplex reciprocating positive displacement diaphragm pump installed in a system, without any pipeline analysis. When the production process was started, unacceptable pipeline vibrations occurred. The plant user feared possible damages and decided to improve the installation.

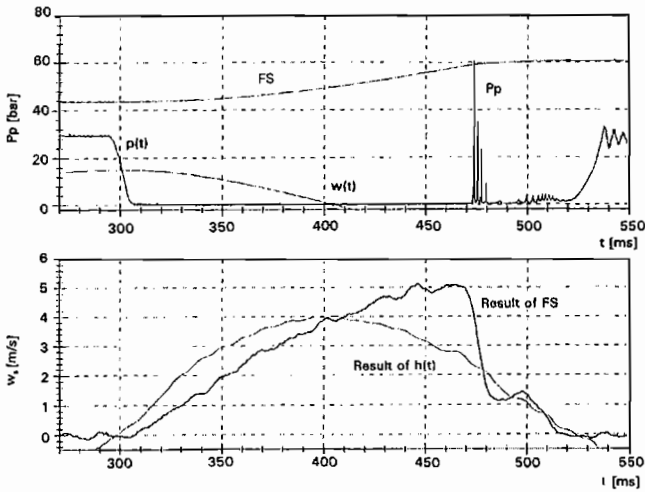


Figure 22. Pressure Peak ( $\Delta p_p$ ) in a Water Pump During Cavitation Back Formation. ( $\Delta p_p \sim 60$  Bar;  $p_D = 30$  Bars at  $h(t)$  Stroke Curve; FS: Flow Velocity in the Suction Line;  $w(t)$ : Plunger Velocity;  $p(t)$ : Pressure Curve.)

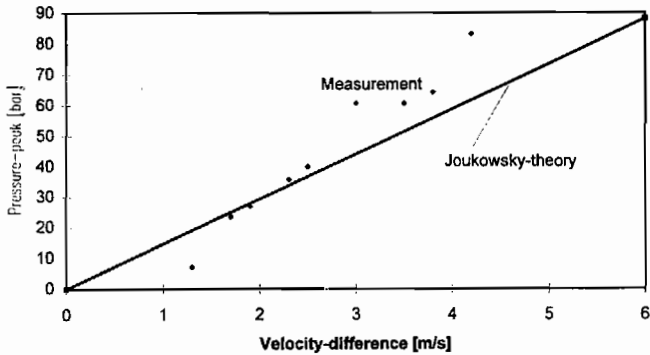


Figure 23. Pressure Peaks Versus Velocity Drop (Verification with the Joukowsky Equation (7) Is Not Exactly Possible).

First Approach for Improvement

As a very common solution, the plant engineer calculated a pulsation damper that should reduce the pressure pulsation below  $\pm 3$  percent, but without giving any consideration to the pipeline system. He installed a pulsation damper of the “right size” (basic rule: damper volume should be about 10 times the stroke volume) at the “right place” in the pipeline. The results were quite surprising: the vibration was remarkably *increased*, especially in the upper pump speed, and the seal of a flange was damaged, which results in shutdown condition. The data of the triplex pump and the system were:

- Stroke volume  $V_S = 0,12$  liter
- Installed damper  $V_D = 1$  liter
- Pump speed  $n = 125$  rpm to  $145$  rpm
- Discharge line length  $L = 26$  m
- Discharge line diameter  $D = 38.1$  mm
- Speed of sound  $a = 1200$  m/s to  $1400$  m/s

Before starting with the real improvement, some basic vibration information is presented.

Basic Information about Pipeline Vibrations and Resonance

With periodic pulsating flow, the liquid column in the pipeline behaves very similarly to the air column in a wind-instrument when excited to vibrate by the lips of the musician. If the excitation frequency matches the natural frequency of the air or liquid column, the system goes into resonance. Resonance is characterized by standing pressure waves with particularly high vibration amplitudes.

This effect (the generation of extremely high pressure pulsations), while useful in wind-instruments, can be otherwise in liquid filled pipelines. If this occurs, the safe operation of a plant can be significantly endangered. Disturbances due to pressure pulsations other than resonance are also possible, especially with long or small diameter pipelines. Therefore, the suction line of pulsating pumps should be short, and the diameter as large as possible.

The maximum pressure amplitude is easily determined by using nondimensional characteristic values (Table 3), if turbulent pipe flow is presumed (Fritsch and Müller, 1986). For a pipeline with low flow friction losses from the installation (fittings, etc.), Figure 24 shows the dependence of the pressure amplitude character,  $\psi$ , on the system character,  $\varphi$ . Below the first resonance frequency, it is  $\psi = \varphi$ . In this case (common when reciprocating positive displacement pumps are used), the fluid in the suction pipeline behaves like an incompressible continuum. The calculation of the pressure amplitude is quite simple, and is also usable for viscous fluids with different rheological properties (Newtonian, intrinsically viscous, dilatant viscous, viscoelastic-plastic, and thixotropic).

Table 3. Characteristic Values for Estimating the Pressure Amplitudes.

Non-dimensional characteristic values for estimating pressure amplitudes	
Throttling characteristic value	$\vartheta = \zeta \frac{w_m}{a}$ (22)
System characteristic value	$\varphi = \frac{\omega}{a} L$ (23)
Natural value	$\varphi = \frac{\Omega}{a} L$ (24)
Pressure amplitude characteristic value	$\psi = \frac{\Delta p}{\rho \cdot a \cdot \omega \cdot s_p}$ (25)
Elasticity characteristic value	$\tau = \frac{c_k}{c}$ (26)

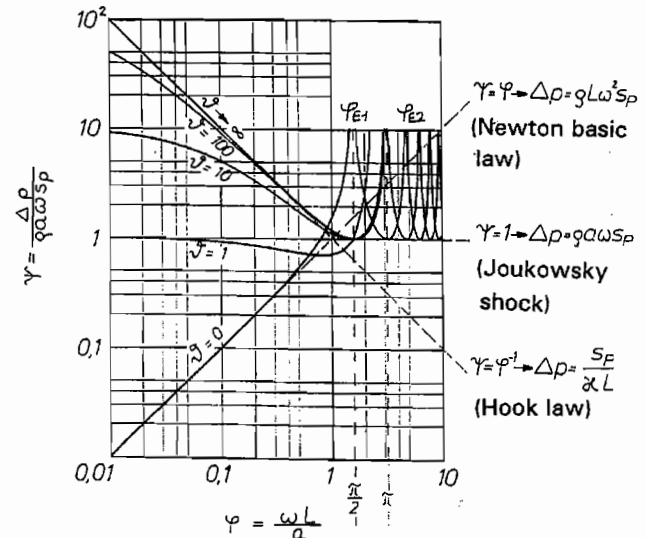
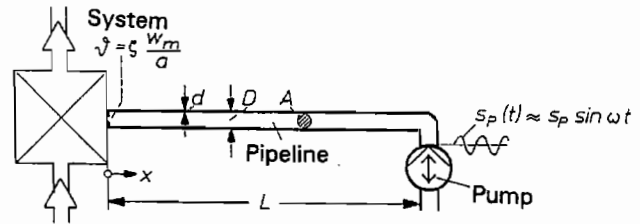


Figure 24. Diagram for Determination of the Maximum Pressure Amplitude in Pipeline without Damping (Dimensionless Display).

Close to the first resonance point ( $\varphi = \varphi_{E1}$ ), the movement of the fluid column in the suction pipeline is decoupled from the movement of the displacer (i.e., the plunger), due to fluid elasticity. The pressure amplitudes reach a maximum at the resonance point and a minimum between two resonance points ( $\psi = 1$ ).

The theory for Figure 24 neglects the friction of the fluid at the pipeline surfaces. This is acceptable, as measurements have shown that the friction of turbulent pipe flow (the usual situation in actual plants) has only a damping effect around the resonance point ( $\varphi = \varphi_E$ ). Because resonance should always be avoided, the divergence between theory and reality has no practical importance (turbulent flow presumed).

*Application of the Vibration Calculation Method*

With respect to the compressibility of the fluids used and the kinematics, a flow curve can be calculated (Figure 25). This flow curve is acting on the pipeline system and can cause pressure pulsation that can generate pipeline vibration. It is important to recognize that every elastic part or system has its natural frequencies.

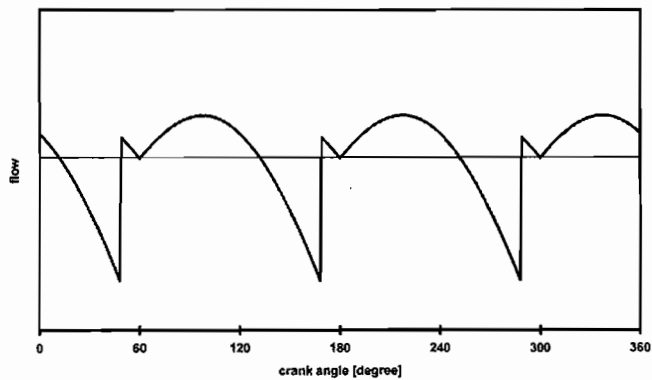


Figure 25. Theoretical Flow Curve of a Triplex Pump at High Pressure.

A simple and useful method to investigate the effect of such a flow behavior on a piping system is to do a Fourier analysis. The Fourier analysis of the curve of Figure 25 delivers a sum of Fourier coefficients. The result of such an analysis is shown in Figure 26. The size of vertical bars is the amplitude of the single Fourier coefficient, which is similar to the vibrating energy content.

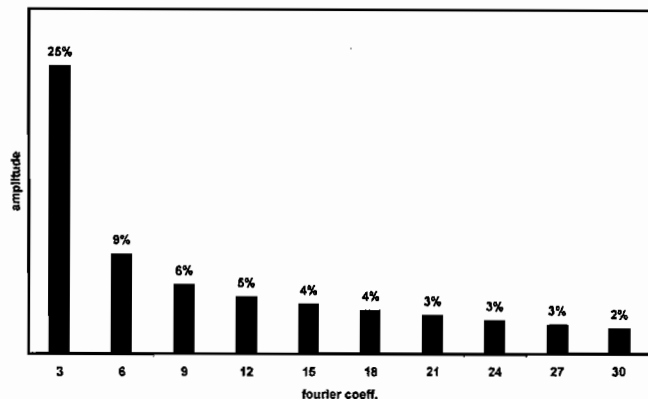


Figure 26. Fourier Coefficients of Figure 25.

It can clearly be seen that the main part of the energy content of the vibration is covered by the first few harmonics (coefficients).

Using the first 10 Fourier harmonics, the rebuild of the curve in Figure 27 looks very similar to Figure 25. Based on experience, to get a satisfactory approximation of the vibration situation in a pump system, the first three to four Fourier harmonics are enough.

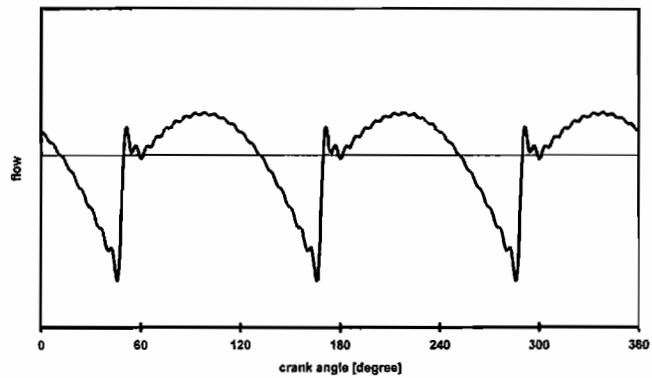


Figure 27. Rebuilt Flow Curve Based on the First Three Fourier Coefficients.

The result of a calculation of the frequency spectrum for a special pipeline example using only three Fourier coefficients is shown in Figure 28. The points in this figure are the result of measurements. The coherence of measurement and theory prove the validity of the calculation. The theory given above is useful for analytical pipeline calculation.

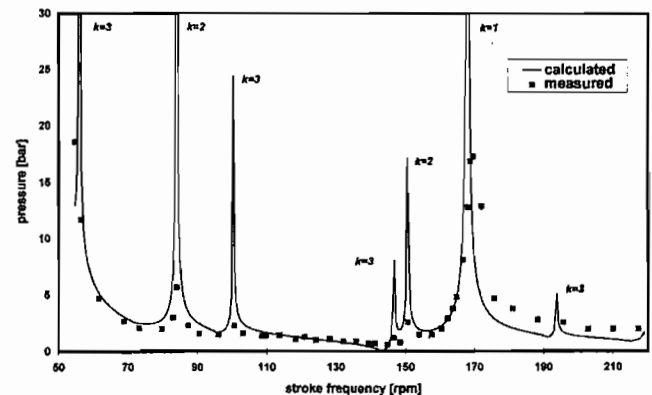


Figure 28. Frequency Spectrum of a Pipeline System Using Only Three Fourier Coefficients (Continuous Line: Theory; Points: Measurement Results).

For more information, the literature (Vetter and Seidl, 1993; Vetter and Schweinfurter, 1989) is recommended.

*Calculation and Improvement of the Vibrating System*

The first calculation of the pipeline based on the triplex pump data given above, without the damper, leads to a pressure amplitude spectrum, as shown in Figure 29. The operating range is close to the first resonance point (first natural frequency). The calculation confirms experience.

With the damper, the system becomes more elastic (softer), which means the maximums of the natural frequencies are shifted to lower frequencies, as shown in Figure 30. Due to the damper, the first resonance point was exactly in the operating range, and the system operated in a dangerous resonance point. Experiences and calculation match again.

Before getting into the actual hardware, another calculation showed the damper could be eliminated and, instead, an orifice with a pressure loss of about 1.5 bar (22 psi) could be installed in the discharge line. The effect is that the system gets stiffer. The natural frequencies are shifted to higher values (Figure 31). A test run in the installation showed a quiet and well operating pump. Pressure pulsations were measured essentially below  $\pm 3$  percent. Also, it is possible to install a much bigger damper, to reach a shift

where the operation range is in a valley between two resonance points.

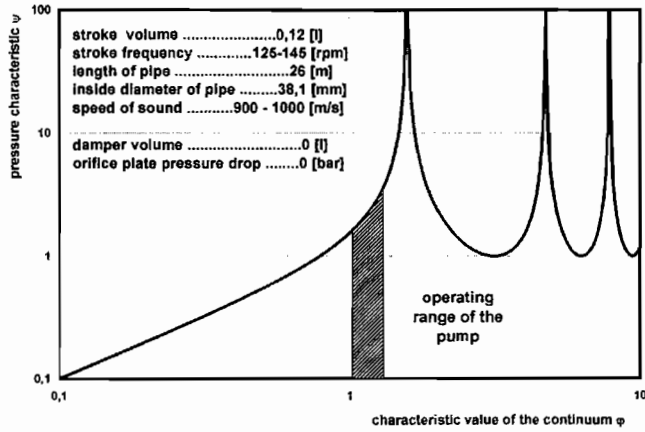


Figure 29. Pressure Amplitude Versus Pump Speed (Spectrum) of the Discharge System without a Damper.

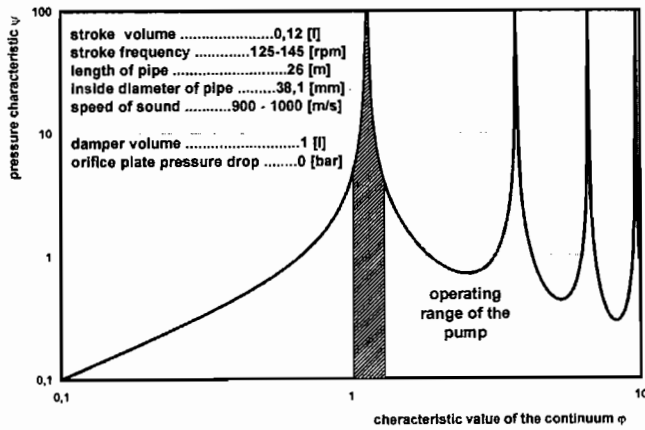


Figure 30. Pressure Amplitude Versus Pump Speed (Spectrum) of the Discharge System with a Damper.

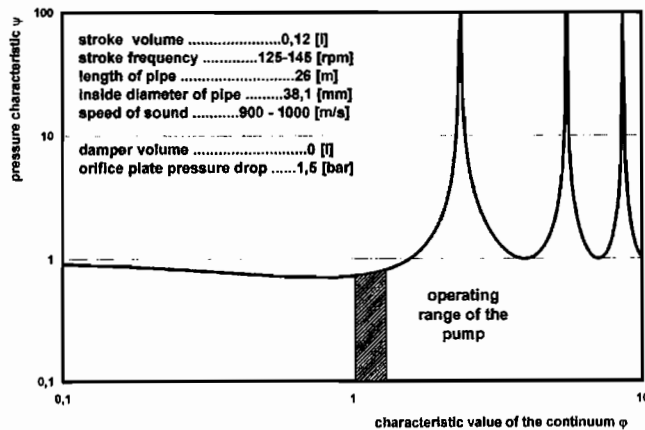


Figure 31. Pressure Amplitude Versus Pump Speed (Spectrum) of the Discharge System with an Orifice with about 1.5 Bar Pressure Loss.

This example is quite simple—but also realistic—and clearly shows that a pipeline analysis concerning vibration potentials should always be done before starting a system.

Remark: Pulsation dampers are only effective when used above the first resonance point.

Operation Range Between Two Resonance Points

The solution is not always as simple as shown above. In another plant, the operating range was much bigger, and was also between two natural frequencies. Figure 32 shows such a situation with a resonance point in the operation range. The solution shown above (installation of an orifice) does not make the situation better (Figure 33), because the throttling effect of an orifice is not sufficient. In this case, contrary to the situation above, a damper installation is the better choice, but this damper should be much larger than normal (about 100 times the stroke volume).

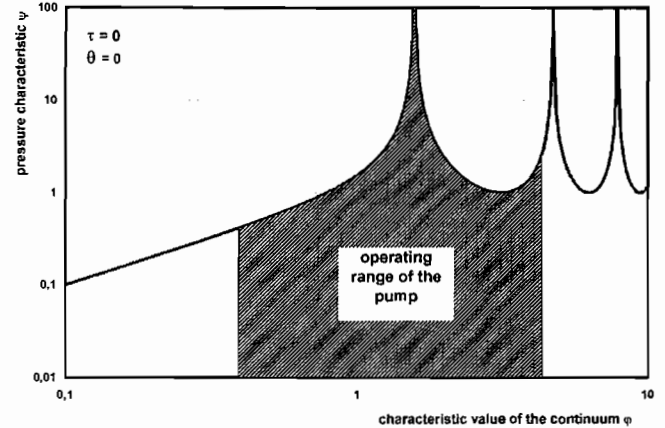


Figure 32. Operation Range Contains the First Resonance Point.

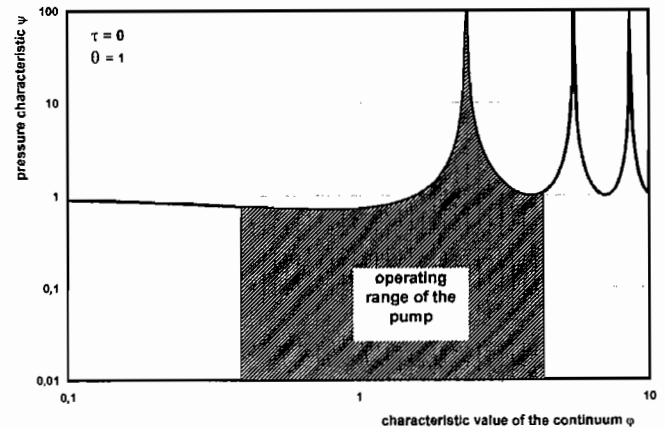


Figure 33. Situation of Figure 31 Including an Orifice.

The effect of this measure is shown in Figure 34. The complete resonance curve experiences a downward shift to smaller values in general. Nevertheless, there is still a resonance point in the operation range. However, this resonance point is very narrow, which usually means that due to damping effects (also compare Figure 28), the actual maximum amplitude is only a small part of the maximum theoretical value. Due to the downward shift of the entire curve and this damping effect, the maximum real amplitude does not exceed the factor one. This was not calculated with the analytical method shown. To be sure that the experience from previous projects also matches this plant configuration, a numerical calculation (Vetter and Seidl, 1993; Vetter and Schweinfurter, 1987) method was used. This method needs more effort, but compared with the possible costs arising from a wrong calculation, it really is worth it.

Additional Possibilities to Avoid Vibrations

Orifices and dampers are effective tools for the prevention of vibration. In that regard, all possibilities and combinations of both are used to find the best solution.

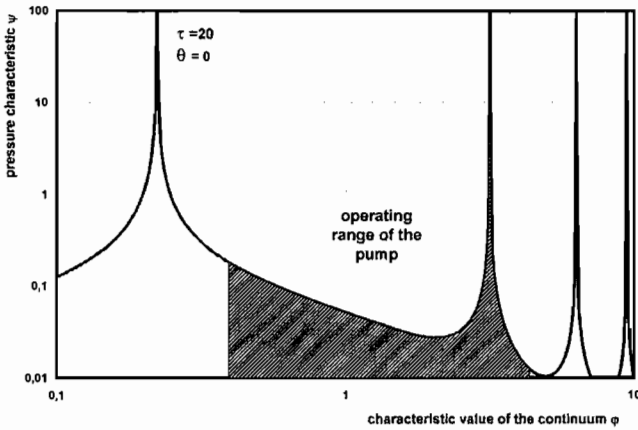


Figure 34. Situation of Figure 31 with an Oversized Damper.

**Summary**

Vibration problems should not be underestimated, because they have the potential to shut down a plant process. These examples show that, even for critical situations, acceptable and reliable solutions can readily be found. It should be clear that a lot of experience is needed to judge an installation, and whether the piping system is critical or not. The pump manufacturer should, and must, have this experience to assist the user to “get it right the first time.”

*Remark:* It is important to know that not only RPDs can generate pulsations and vibrations. This also happens with every rotating positive displacement pump and in some centrifugal pump applications. The tools to avoid vibrations with such pumps are quite similar.

**SLURRIES AND INCORRECT PIPE DESIGN CAN CAUSE HIGH VALVE WEAR**

*Scenario:* Contrary to very good results at one production site (pipe design 1, Figure 35) conveying highly abrasive catalyst slurries for about 15 years, several newer production sites with the same suspension fluid (Figure 35 (2), (3), and (4)), pump size, and check valves, had wear in the check valves (especially the suction valves) that was absolutely unacceptable.

**Valve Design**

The valve type used (Figure 36) was a well-proven design with very sturdy and chemically resistant hard metal inserts, absolutely identical in all pumps. The fluid was also nearly identical in size and concentration of the particles and the carrier fluid.

This valve type uses a stem-guided closing part, guided in polymer sleeves. The main reason for the good results with that design is that the closing part remains centered. When the closing part meets the seat, it always matches quite perfectly. There are only a few “erasing” movements necessary, until the final position of the closing part is reached. A lot of good experience with big valve sizes and suspensions confirms that this theory must be the reality.

**Analysis of the Wear Reason**

At one production plant, the complaint was concentrated on the wear of the valve stem and valve insert, while in another plant, extremely short service life of the hard metal parts (valve seat and closing part) was being experienced. Sometimes the maximum service life was only two to five weeks, while in the plant with the piping design shown in pipe design 1, six to eight months were very usual.

It is difficult to explain why such wear results occur. A random factor could be excluded, because there must be something common

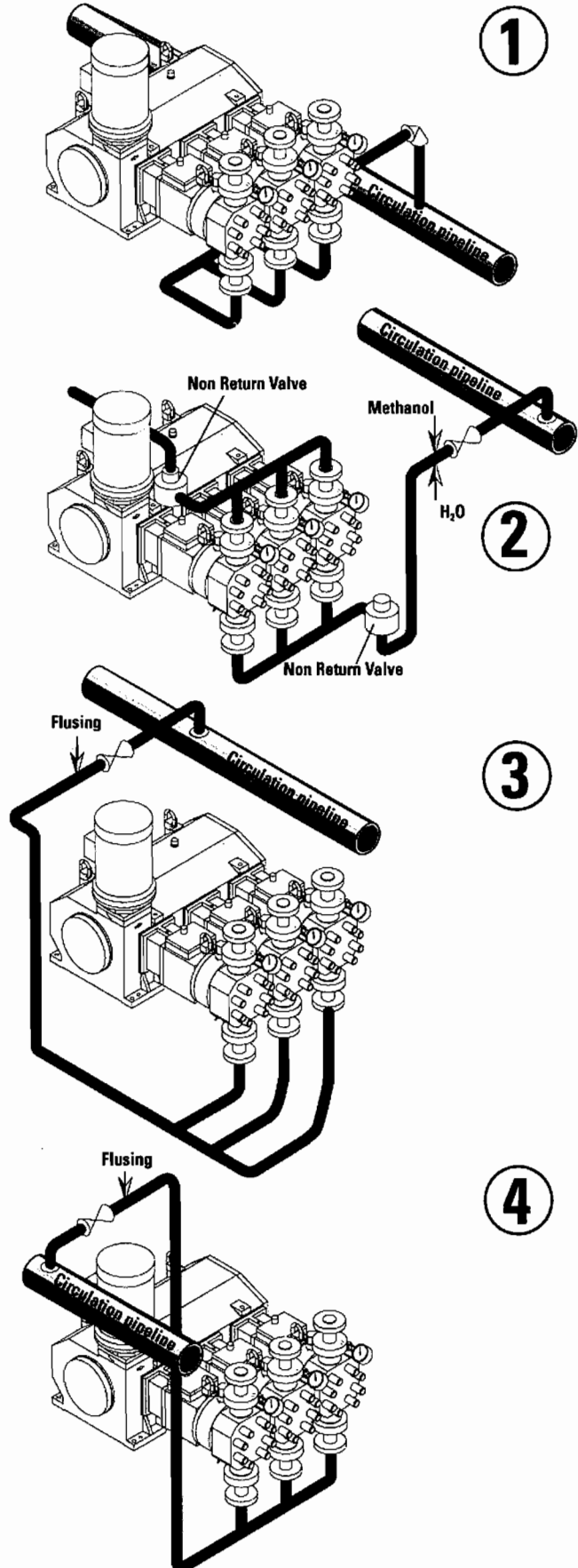


Figure 35. Different Suction Pipeline Designs for Slurry Plants.

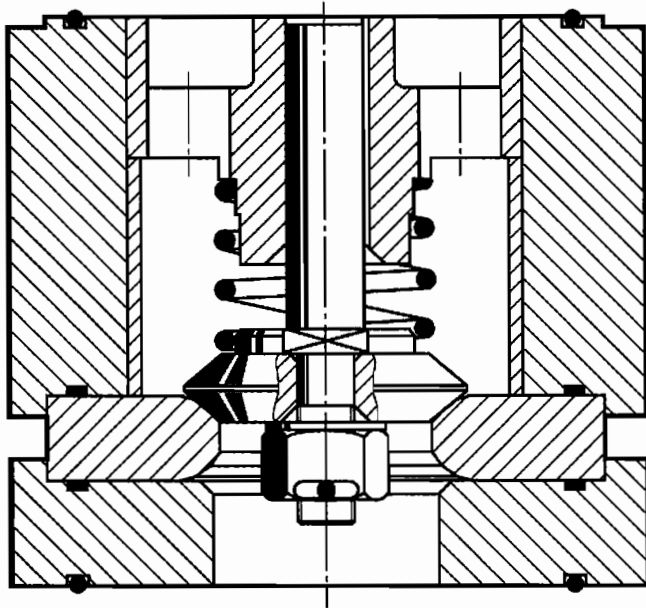


Figure 36. Well-Proven Slurry Cone Valve.

in all four plants. The first thought was that the abrasivity of the particles was not comparable. Yet there was no confirmation for that.

The only remarkable difference in the plant that was operating well was the pipeline system on the suction side. Figure 35 shows the installations in the problematic systems are designed with a circulation pipeline *above* the suction valves. The well-proven design 1, operated and optimized successfully over many years, uses a circulation line *below* the suction valves instead. Pipe design 1 matches the requirements for abrasive slurry application.

#### Failure Mechanism

In all three malfunctioning systems, the circulation pipeline is located far above the suction valve. This means the sedimentation of particles goes in a direction to the pump and suction valves. This is especially bad after an operating interruption, as all the particles from the slurry contents in the upper pipeline will accumulate at the lowest point. The longer the pipeline, the more accumulates and may plug the suction line. After restart, the pump does not see a slurry, but nearly pure particle bulk solids. Depending on the width of the particle size distribution, this bulk is fractionated, which means the biggest particles are at the lowest point.

*Remark:* A similar effect can happen when the fluid velocity in the suction line is not big enough to keep the particles in suspension.

It is easy to see how this leads to higher stress load on the valves, as during the first seconds after restart there is no turbulent flow through the valve. It is more like a highly abrasive plug flow (like a file). Compared with the normal situation with a small flow velocity at the walls and a high flow velocity in the center of the flow chambers, the particles move with a higher speed at the walls in plug flow, and therefore more abrasion occurs. The valve parts are not "lubricated" by the suspension liquid when the particle concentration exceeds a certain value, and the wear rate increases tremendously.

Additionally, it is highly probable the situation (particle accumulation) in the valves remains for long periods of time, because the development of a normal flow condition first needs a "purging" of the valve chambers by flow. The abrasion phase lasts far into the normal process, and perhaps even stays during the whole operating time.

#### Remarks about Abrasivity of Hard Particles

Concerning abrasivity, it must be clearly understood that with special types and hardness of particles, the wear rate decreases by

choosing a harder valve material. However, it is not so simple to see that, even with very hard materials, the wear never gets to zero (Vetter, et al., 1995a). It is well accepted in abrasion science that concentration is also a critical factor in increasing wear situations.

#### Searching for Inexpensive Solutions

Due to the high investment cost to change the system to be like system 1 (the proven design), it was very difficult to convince the engineers responsible. Instead of this, one hope was that a change and improvement of the valve design would give the required results of longer service life.

Not surprisingly, a design using a crowned sealing geometry at the closing part and no guiding stem increased the service life, but not sufficiently, and only on one pump. The reason for the improvement was that such a design reacts more flexibly when meeting the particle bulk. Nevertheless, it also experienced closing part vibrations and some other failure modes.

Another attempt to decrease maintenance cost: the consequence of the severe wear was an expensive spare part consumption. This was, of course, not acceptable. To decrease the spare part cost, one decision was to rework the used parts or manufacture the new parts in their own or local shops. This reduced (or masked) spare part expenditures, but retained high maintenance cost level, and treated only the symptom but avoided the cure.

#### Another Valve Wear Scenario and Solution

A plant engineer complained that he got terrible valve wear in a relatively clean process (no suspension). A reason was not found until it was asked where the fluid comes from. The answer was, "a catalytic process." After more questions, it was obvious that the valve failure always occurred at the beginning of the process. Investigations found that the fluid was relatively pure, but with a low concentration of very small and abrasive catalyst particles. During shutdown, the particles accumulated in the suction pipe connected at the lowest point of the suction vessel. After restart, the same situation as above occurred. Due to the lower abrasivity of the particles, it was enough to use hard metal parts for the valves.

#### Résumé for Abrasive Slurry Systems with RPDPs

The long-lasting positive experiences with system design 1 (eight pumps) and other positive experiences with various types of abrasive particles, and the descriptions of the reasons for the failures, are convincing that the systems and not the pumps or valve designs of system 2, 3, and 4 were the weak point. Therefore, it is recommended that whenever a new system for abrasive slurries would be built, it should be similar to design 1. Of course this is not a blanket guarantee for reliability, but in combination with a broad valve know-how, a very good fundamental approach.

#### MONITORING AND EARLY FAILURE DIAGNOSIS

Failure diagnosis (FD) and early failure diagnosis (EFD) based on a reliable and competent monitoring of the conveying process RPDP pumps becomes more and more important for reaching an economical production process. A reliable FD and EFD offer:

- Recognition of a failure development.
- Early order of spare parts.
- Short price interruptions.
- Smaller production loss.
- Reduction of the extent of damages.

The acting principle of RPDPs generates a stable sequence of function steps (compare with Figure 1). Due to that behavior, this principle offers good possibilities for monitoring and significant failure diagnosis.

The most common early failure diagnosis system, of course, is the sandwich-diaphragm-system with rupture indicator as



described above. More than 90 percent of all diaphragm pumps are delivered with such a sandwich diaphragm. This technology is simple to use and highly accepted. Depending on the quality of the channel system from the diaphragm package and the rupture indicator, a diaphragm rupture is signaled in less than one minute.

Yet this single aspect is not enough for the future. Other methods should be found and implicated in pump systems, giving beneficial support to make the right decision at the right time. Below, three methods are described to present an idea about already existing possibilities.

*Failure Diagnosis Based on Dynamic Pressure Measurement*

Due to the leak through the plunger seal and the venting valve, this leak has to be recovered each stroke. By way of example, Figure 37 shows a normal pressure curve in the hydraulic chamber of a diaphragm pump. Toward the end of suction stroke and continuing until the beginning of the compression stroke, the pressure drop is below atmospheric. This is referred to as the “replenishing window,” is normal and healthy, and represents the recovery of oil in through the replenishing valve. Normally, by design, this window should have a duration of less than seven percent of suction time, which means that the total oil leakage is in the order of about one to two percent of the swept volume per stroke.

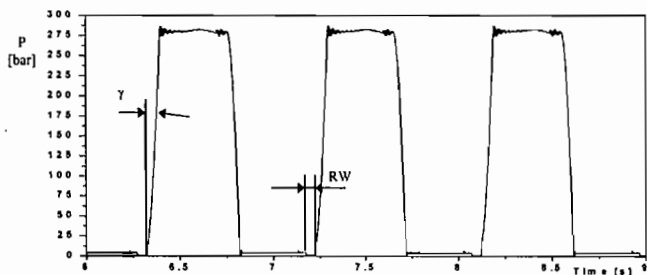


Figure 37. Pressure Curve with a Step Compression Phase and a Short Healthy Replenishing Window.

Additionally, if the compression line gets less step ( $\gamma$  angle), the volumetric efficiency decreases. The reason can be air or gas in the pressure chambers, or leakage. A significant sign of big leakage is an increase of the replenishing window, as shown in Figure 38. Also the popping of the pressure relief valve is easy to recognize (Figure 39).

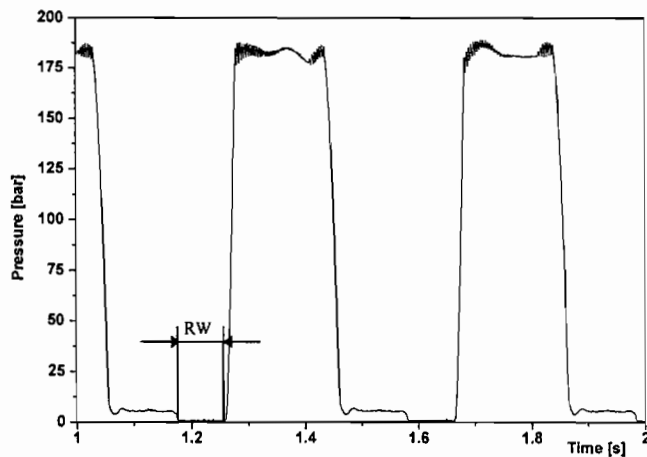


Figure 38. Pressure Curve with Excessive Leakage Flow at Plunger Seal. Replenishing Window Too Wide.

*Structure-Borne Noise Signals for Early Failure Diagnosis*

Well known from failure diagnosis of turbines, the structure-borne noise (SBN) signals content potential for failure diagnosis.

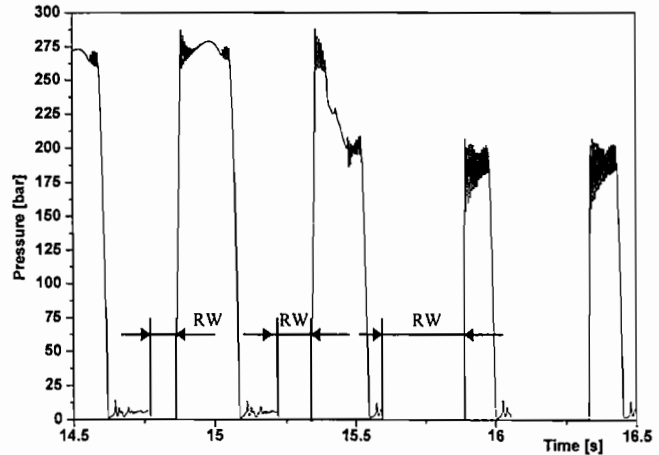


Figure 39. Typical Pressure Curve When the Pressure Relief Valve Begins to Act. Replenishing Window Length Increases.

Figure 40 shows a pressure curve and an SBN signal of a healthy pump. Usually easy to recognize is the discharge valve closing.

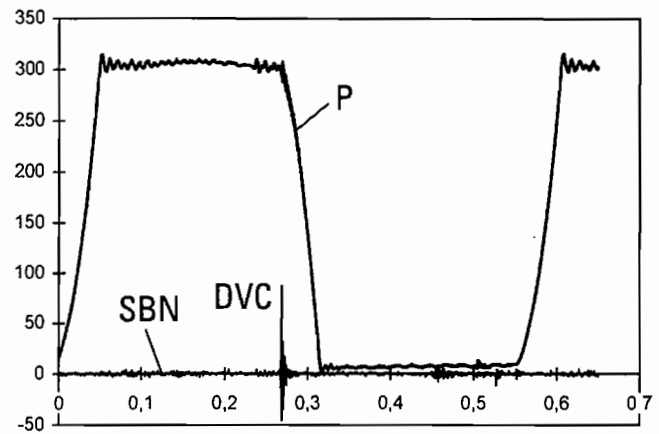


Figure 40. Pressure Curve and Structure-Borne Noise Signal. (DVC: Discharge Valve Closing, P: Dynamic Pressure Curve.)

If the pump has a leakage at the suction valve, the SBN signal looks quite different (Figure 41). During the discharge phase (suction valve is closed), the leakage flow through the valve generates a tremendous SBN burst.

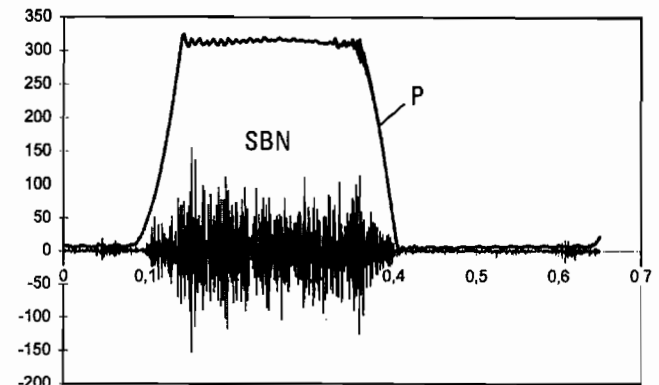


Figure 41. Pressure Curve and SBN Signal When a Suction Valve Leakage Occurs.

### Venting Valve Monitoring

When using a volumetrically acting venting valve, a constant output volume per stroke of that valve to the reservoir is a significant sign for a proper compression phase and air or gas free pump chambers. Usually this means a healthy pump process. This method cannot be monitored electronically and, therefore, is only useful when working at the pump.

### USE OF MEASUREMENT OF DYNAMIC PRESSURE TO EVALUATE FLOW PERFORMANCE AND FLUID PROPERTIES, AND TO DIAGNOSE PROBLEMS IN DIAPHRAGM PUMP SYSTEMS

#### Overview of the Value of Pressure Measurements

Measurement of dynamic pressure in diaphragm pump systems can be a powerful tool to:

- Help evaluate process fluid properties,
- Help establish and maintain optimum performance, and
- Give early warning and help in diagnosing problems so they can be corrected through routine maintenance rather than unplanned production outage.

In recent years, there have been many strides in the areas of continuous monitoring of dynamic measurements, as well as in better understanding of the detailed workings of diaphragm pumps and their effects on process fluid flow performance. Many of these strides and learnings are explored here in the context of practical applications.

Some remarks to preface this discussion relate to the issue of hardware and software developed in recent years, which, to a large degree, have made possible the analytical methods and results discussed here. References to and comparisons of particular vendor products are purposely avoided, for two reasons. First, since such products change and improve rapidly, specific references would be quickly dated. More important, though, the learnings lie in the understanding of how practical needs in engineering analysis, diagnostics, and design drive both the applications and future improvements of these new tools.

#### Engineering Tools for Dynamic Measurements and Monitoring

In chemical process plants and other types of manufacturing facilities, distributed control systems (DCS) are vital to the monitoring and control of the process. Measured variables are generally then stored electronically, usually once per minute, for historical record, trending, and other purposes. In diaphragm pump systems, the variables of interest include mean flowrates, densities, temperatures, pressures, controller settings, and other data. These process data can be critical when needs arise to perform detailed dynamic analyses during brief events, or when particular conditions of interest occur.

Obtaining this type of detailed information has, in the past, been tedious, time-consuming, imprecise, and, in many cases, only available at the plant site. The growth in use of the Internet and Intranets in recent years, however, has spawned software in this area enabling easy access of this detailed information anywhere in the world. Figure 42 shows this for a one-day period, with the cursor placed on a single minute of interest for a pump system.

The frequencies of interest for fluid pressure pulsation and mechanical component dynamics in a diaphragm pumping system can range as high as 200 Hz or more. This is well above the response capabilities of process control system devices and requires the use of special transducers, signal conditioning, and signal analysis. The equipment and technology associated with this have existed for many years. What is relatively new, however, and discussed here are the software system capabilities, linked also through Internet and Intranet media, to provide continuous online monitoring of these dynamics anywhere in the world.

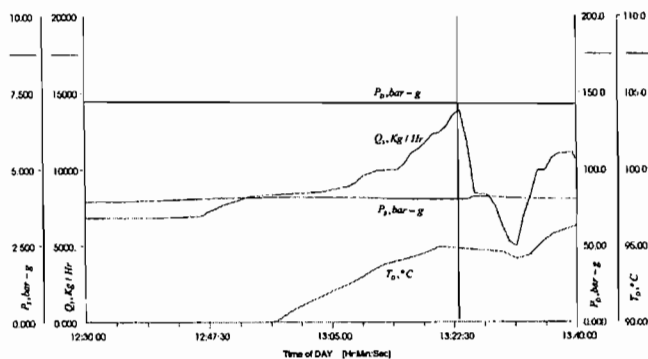


Figure 42. Sample Display of Historical Process Control Data from a European Chemical Plant DCS System—Information Accessed from Engineering Office in the United States.

Shown in Figure 43 are time data of dynamic pressure captured during the minute corresponding to the cursor location in Figure 42. This dynamic pressure was measured in the hydraulic side of a triplex diaphragm pump, stored digitally, and available live, both locally at the plant and remotely anywhere in the world.

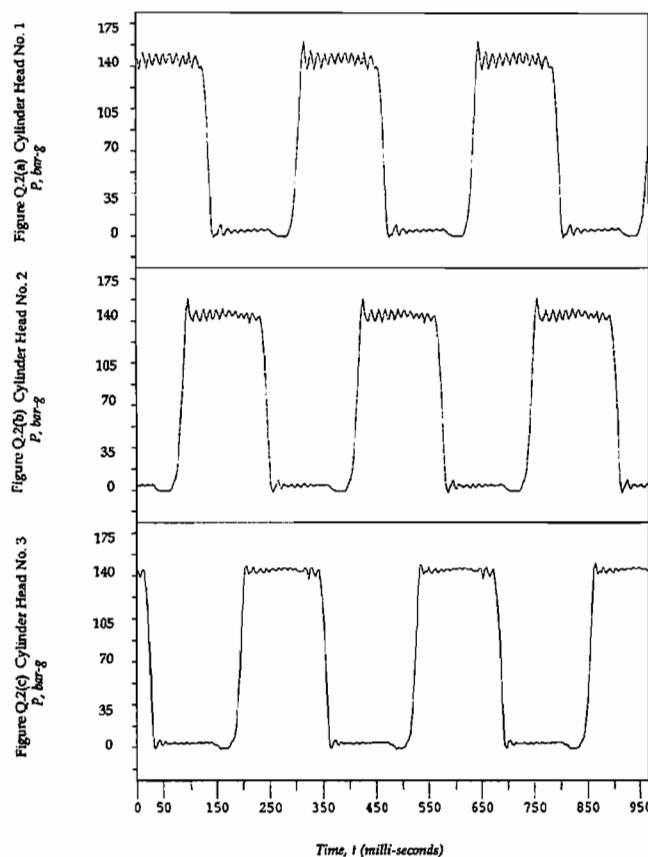


Figure 43. Hydraulic Oil-Side Pressure of Pump Heads of a Triplex Diaphragm Pump Equipped with a Dynamic Monitoring System—Data Capture Rate, 300 Samples per Second.

#### Diaphragm Pump Performance Calculation

Factors that affect efficiency (Equation (1)) in a diaphragm pump include the following, generally in order of significance:

- Compressibility of the process and hydraulic fluids,  $\eta_K$ ,
- Leakage of hydraulic fluid back to the reservoir,  $\eta_{HL}$ ,
- Reverse flow of process fluid through the check valves,  $\eta_{CV}$ , and

- Flexibility of the walls of the cylinder head,  $\eta_w$ .

Often, the most significant effect on pump efficiency relates to fluid compressibility. The magnitude of this loss depends on the particular process fluid being pumped, the amount of entrained and dissolved gas, the pressure ratio, the clearance volumes, and other factors. For a broad range of chemical process applications, this loss can range from a few percent to 20 percent or more, or,

$$(1 - \eta_K) \approx (3\%, 20\%+) \quad (22)$$

In the absence of deficiencies in the basic design and integrity of components, check valve leakage and cylinder wall flexibility are assumed to impose significantly lower order effects on performance, or:

$$(1 - \eta_{CV}) \ll 1\% \quad (23)$$

$$(1 - \eta_w) \ll 1\% \quad (24)$$

Thus, for purposes in this discussion, Equation (2) will be approximated to:

$$\eta_V \approx \eta_K \eta_{HL} = \frac{m_i \nu_{PS}}{\nu_{SW}} \quad (25)$$

Actual measured performance that deviates significantly from Equation (25) will be regarded as indicative of deficiencies in the basic design or integrity of the pump, or components, meriting more detailed scrutiny. An example of this is discussed later.

Figure 44 schematically shows the elements of a diaphragm pump head with several volumes and physical elements indicated. Much of the foregoing discussion has been devoted to the normal functions and workings of these components during normal and abnormal operation. As we begin to explore the subject of flow performance, one of the key differences between diaphragm and simple plunger-type pumps relates to the quantity often referred to as the *clearance volume*. In ordinary plunger-type and other simple positive displacement pumps, the term *clearance volume* or *dead volume* is a fixed, known quantity well understood to be the fluid volume remaining in the cylinder at the end of the discharge stroke. While generally desirable to minimize this value, it is exactly the same on every stroke, comprising valve passageways and any cavity volumes that cannot be swept by the plunger. It is an important parameter because it, along with fluid compressibility and pump head, determine an important portion of the volumetric efficiency, and hence performance, of the pump.

An important difference with the diaphragm pump is that the volume,  $V_{PD}$ , of process liquid at the end of the discharge stroke is *not* fixed, and in fact will vary dependent upon many factors. All these factors, and the roles they play in determining the overall flow performance of the diaphragm pump, are explored in this section.

An important property affecting diaphragm pump performance is bulk modulus or its inverse, compressibility. By definition, these are expressed as follows for the hydraulic and process fluids, respectively.

$$K_H = \frac{-1}{\nu_H} \frac{\partial \nu_H}{\partial P} \quad (26)$$

$$K_P = \frac{-1}{\nu_P} \frac{\partial \nu_P}{\partial P} \quad (27)$$

At this point, a model for hydraulic oil leakage is introduced. There are several leak paths for the hydraulic oil that include the seals, around the plunger, and backflow through the snifter (very small)

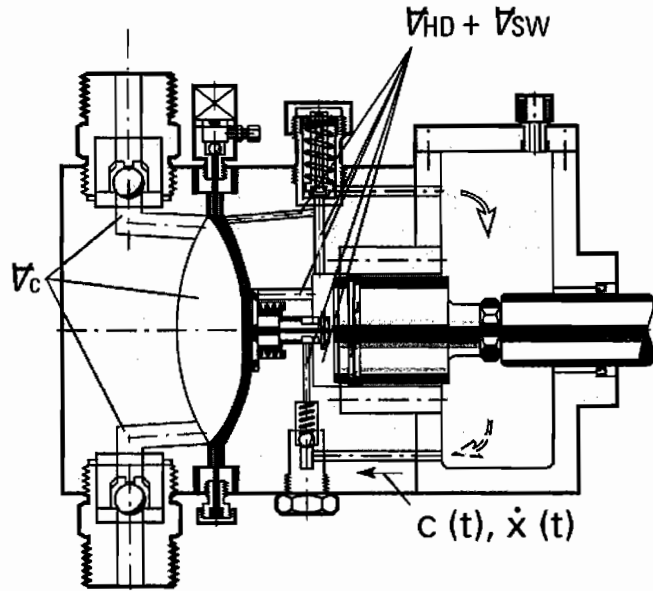


Figure 44. Cutaway of Diaphragm Pump Internals Positioned at the End of the Suction Stroke. ((Time,  $t = 0$ )—Quantities,  $V_C$ ,  $V_{HD} + V_{SW}$ ,  $x(t)$ ,  $\dot{x}(t)$ , and  $\omega t$  Indicated.)

and venting valve. All these paths allow hydraulic fluid flow from the pumping chamber at chamber pressure to the reservoir at atmospheric pressure. As mentioned above, however, the design strategy of diaphragm pumps purposely places most of the leakage through the venting valve. All of the leak paths taken together are modelled here according to the relation:

$$q_{HL}(t) = CP^2 \quad (28)$$

where the flow coefficient,  $C$ , is a parameter that is designed into the pump application. For any particular steady-state pump operation, total leakage and replenishment of hydraulic oil must sum to zero over each crank revolution. Equationally:

$$\int_0^{\tau} \left[ q_{HL} + \langle t - t_S \rangle^0 \frac{dx(t)}{dt} \frac{A}{\nu_H} \right] dt = 0 \quad (29)$$

where the singularity function for a step,  $\langle t - t_S \rangle^0 \equiv 0, t < t_S; \equiv 1, t \geq t_S$ . The flow coefficient,  $C$ , can thus be determined experimentally, using data from one crank revolution according to:

$$C = \frac{- \int_0^{\tau} \langle t - t_S \rangle^0 \frac{dx(t)}{dt} \frac{A}{\nu_H} dt}{\int_0^{\tau} P^2 dt} = 0 \quad (30)$$

Now, consider an incremental change in plunger position,  $\delta x$ , during the compression process corresponding to a time increment,  $\delta t$ . This is accompanied by a change in cylinder pressure,  $\delta P$ , and leakage of some hydraulic oil to the reservoir. The total change in volume of the pressurized fluids, both process and hydraulic, may be expressed:

$$-A\delta x = \delta P(V_P K_P + V_H K_H) + CP^2 \nu_H \delta t \quad (31)$$

Since the mass of the process fluid does not change during compression, then:

$$V_P = \frac{V_C}{V_{PS}} v_P \quad (32)$$

It is now useful to recognize that the sum total volume of pressurized fluid, process and hydraulic, may be expressed:

$$V_P + V_H = V_C + V_{SW} - Ax(t) + V_{HD} \quad (33)$$

which allows expressing in the form:

$$V_H = V_C \left(1 - \frac{v_P}{V_{PS}}\right) + V_{SW} + V_{HD} - Ax(t) \quad (34)$$

Substituting Equations (32) and (34) into Equation (31), and rearranging, the result may be expressed in either of the following forms dependent upon the unknown being sought.

$$\delta P = \frac{-A\delta x(t) - CP^2 v_H \delta t}{K_H(Ax(t) + V_C + V_{HD}) + \frac{v_P V_C}{V_{PS}} (K_P - K_H)} \quad (35)$$

$$K_P = K_H - \frac{v_{PS}}{v_P V_C} \left[ K_H(Ax(t) + V_C + V_{HD}) + \frac{A\delta x + CP^2 v_H \delta t}{\delta P} \right] \quad (36)$$

The final useful equations introduced, before exploring some examples of real processes, are the well-known slider crank equations for piston position and velocity.

$$x(t) = r(1 - \cos(\omega t)) - 1 + \sqrt{1^2 - r^2 \sin^2(\omega t)} \quad (37)$$

$$\dot{x}(t) = r\omega \sin(\omega t) \left[ 1 - \frac{r\cos(\omega t)}{\sqrt{1^2 - r^2 \sin^2(\omega t)}} \right] \quad (38)$$

In the following examples, the equations (including Equation (1)) and concepts introduced above are used to study the performance and behavior of two different triplex diaphragm pumps. In the first example, normal operation of a new healthy pump is analyzed to quantify the performance, hydraulic oil leakage, and process fluid compressibility. In the second example, these same concepts are used to diagnose the source of off-performance in a pump after operation for several months. In both cases, spreadsheets are a valuable tool to perform very detailed calculations based off high-speed dynamic measurement data for several hundred increments throughout a single crank revolution. Due to space limitations here, though, these calculations are briefly summarized.

#### Example 1—A New Diaphragm Pump Installation

The triplex pump, operating as described in Table 4, is instrumented with dynamic monitoring equipment that involves precision pressure transducers to sense the hydraulic-side oil in each of the three heads. Figure 43 shows the measured dynamic pressure for this operating case.

A detailed analysis of the digital data of Figure 43 shows  $t_S = 0.31$  sec for each head. Hence,

$$\begin{aligned} \theta_S &= 22^\circ \\ x(t_S) &= 3.75 \text{ mm} \\ V_{HL} &= Ax(t_S) = 12,500 \text{ mm}^3 \\ \eta_{HL} &= (V_{SW} - V_{HL})V_{SW} = 0.97 \\ C &= 7.07 \times 10^{-6} \text{ Kg/s/bar}^2 \end{aligned}$$

Hence, the loss of efficiency due only to hydraulic oil leakage, at three percent, is on the high end of the desired range of one

Table 4. Basic Pump and Operating Data for Example 1.

Pump Data	Operating Data	Calculated Quantities
$h = 120 \text{ mm}$	$p_S = 4.14 \text{ bar} - g$	$V_{SW} = 398,200 \text{ mm}^3$
$D = 65 \text{ mm}$	$p_D = 141.4 \text{ bar} - g$	$V_{HD} = 541,550 \text{ mm}^3$
$l = 450 \text{ mm}$	$\rho_{HS} = 0.81$	$V_{PD0} = 1,525,100 \text{ mm}^3$
$\epsilon_H = 2.36$	$\rho_{PS} = 1.15$	$V_C = 1,923,300 \text{ mm}^3$
$\epsilon_P = 3.83$	$N = 181.8 \text{ rpm}$	$v_{PS} = 1.15 \times 10^6 \text{ mm}^3 / \text{Kg}$
		$\tau = 0.33 \text{ sec.}$

percent to three percent. So, an additional one percent to two percent performance may be obtained from this machine by reducing the flow through the continuous flow port of the combination valve.

Analysis of this dynamic data using the relations and assumptions introduced here additionally yields the following:

$$m_i = 0.413 \text{ Kg. and} \quad (39)$$

$$\eta_V = 0.90 \quad (40)$$

Thus, from Equation (25),

$$\eta_K = 0.93 \quad (41)$$

At this point, enough information is known to be able to calculate process fluid compressibility. To do this, attention is directed to the dynamic pressure measurement data points during the compression stroke. The sampling rate for this particular test was 0.003333 seconds, yielding about 100 points each crank revolution. Of these, seven points lay on the brief compression stroke. In general, the steeper the compression line (lower  $\gamma$ , Figure 37), the higher the volumetric efficiency. The primary factors affecting this are the bulk compressibility of the fluid, the amount of entrained and dissolved gas, and hydraulic oil leakage. The following application of equations developed here takes all these factors into consideration.

Table 5 summarizes a few of these points with associated calculated quantities leading to the inference of process fluid compressibility.

Table 5. Summary of Detailed Calculations During Compression Example 1 (Expressed in Thousands).

Time sec.	$\omega t$ rad.	$x(t)$ mm.	P, bar-g	$K_H$ , l/bar	leak., Kg	$V_H$ , mm <sup>3</sup>	$V_P$ , mm <sup>3</sup>	$v_P$ , mm <sup>3</sup>	$K_P$ , l/bar	B, psi
0.0100	0.1904	0.9410	17.73	77e-6		939.7	1,920	868.2		
0.0133	0.2539	1.6706	33.90	75e-6	27e-6	938.5	1,919	867.6	40e-6	360
0.0167	0.3173	2.6059	54.44	73e-6	70e-6	937.0	1,917	866.9	41e-6	355
0.0200	0.3808	3.7447	78.85	70e-6	147e-6	935.2	1,915	866.0	43e-6	340
0.0233	0.4442	5.0843	109.5	68e-6	283e-6	932.9	1,913	865.0	37e-6	395

A note regarding this back-calculation of process fluid compressibility is in order. Enough test data conditions have been examined as part of this work to develop some confidence in the approximate value of compressibility. It is also generally expected that with most liquids, as pressure increases, compressibility tends to decrease. In this case, for example, to say the compressibility of the process over this pressure range is about  $0.00004 \text{ [bar}^{-1}] \pm 10$  percent is probably as accurate as the method allows.

#### Example 2—A Pump after Several Months of Operation

The triplex pump described and operating, as shown in Table 6, is instrumented with dynamic monitoring equipment that involves precision pressure transducers to sense the hydraulic-side oil in each of the three heads. Figures 45 and 46 show the measured dynamic pressure for this operating case.

Table 6. Pump and Operating Data for Example 2 Installation.

Pump Data	Operating Data	Calculated Quantities
S = 90 mm	$p_s = 3.33\text{bar} - g$	$V_{SW} = 191,134\text{mm}^3$
D = 52 mm	$p_D = 143\text{bar} - g$	$V_{HD} = 259,942\text{mm}^3$
l = 350mm	$\rho_{HS} = 0.81$	$V_{PD0} = 732,043\text{mm}^3$
$\epsilon_H = 2.36$	$\rho_{PS} = 1.163$	$V_C = 923,177\text{mm}^3$
$\epsilon_P = 3.83$	$N = 142.2\text{rpm}$	$v_{PS} = 1,163,000\text{mm}^3 / \text{Kg}$
		$\tau = 0.416\text{sec.}$

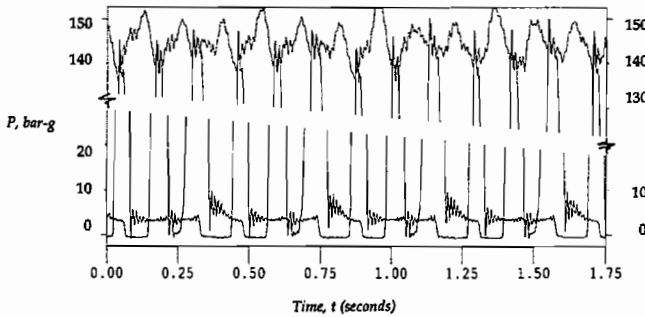


Figure 45. Monitored Pressures of Three Pump Heads Superimposed for a Triplex Diaphragm Pump Experiencing Excessive Hydraulic Oil Leakage to the Reservoirs, Especially at Head 1 and 3.

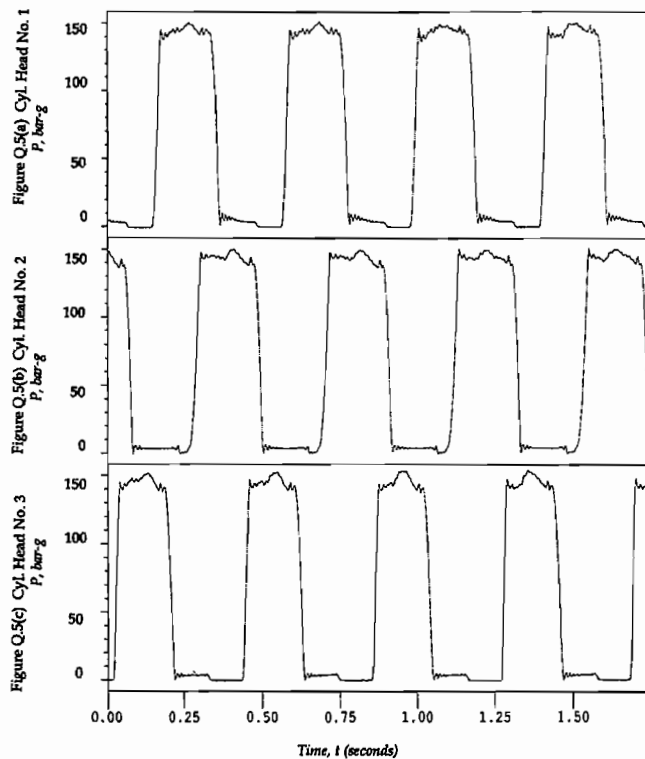


Figure 46. Monitored Pressures of Three Pump Heads Individually for a Triplex Diaphragm Pump Experiencing Excessive Hydraulic Oil Leakage to the Reservoirs.

A cursory look at this figure reveals noticeable differences among cylinders in the shape and duration of the snifter valve opening events toward the end of the suction stroke. This is seen more clearly in Figure 47, which zooms on the suction pressure behavior. A detailed analysis of the digital data of this shows

significant leakage occurring in cylinder number three, and to a lesser degree, in cylinder number one. The details of this are summarized in Table 7.

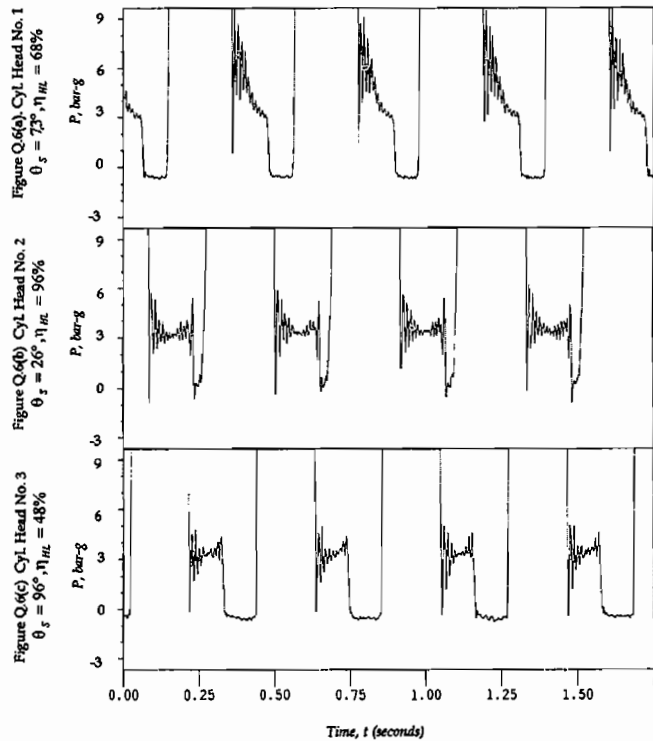


Figure 47. Monitored Pressures of Three Pump Heads, Zoomed on Suction, for a Triplex Diaphragm Pump Experiencing Excessive Hydraulic Oil Leakage to the Reservoirs.

Table 7. Summary of Oil Leakage Parameters per Cylinder—Example 2.

	Cyl. No. 1	Cyl. No. 2	Cyl. No. 3
$t_s$ [s]	0.332	0.386	0.305
$\theta_s$	73°	26°	96°
$x(t_s)$ [mm]	29	4	47
$V_{HL} = Ax(t_s), \text{mm}^3$	61,500	8,500	100,000
$\eta_{HL} = (V_{SW} - V_{HL}) / V_{SW}$	68%	96%	48%

The overall effect on volumetric efficiency in this condition is  $\eta_K = 70$  percent, versus 97 percent to 99 percent for the healthy pump in this application. Because of the knowledge gained through these measurements regarding the source of the diminished performance, correction was accommodated during a planned maintenance shutdown. This also enabled the service personnel to prepare accordingly at the time of overhaul.

Software Systems for Remote Access to Live Dynamic Data

Measurement systems with continuous monitoring capability and ability to access live high-frequency dynamic data from anywhere in the world have become easier to configure in recent years and even relatively inexpensive. This comes thanks to the rapid growth in Internet speed and capability, in general, as well as a competitive industry in hardware and software systems in this area, in particular. Discussed here in general terms is an important component of the system—the software. A host version of this software will reside on a dedicated computer at the plant site, and one or more client versions will reside on individuals' computers that can gain access to this live data from anywhere.

The class of software discussed here, of which there is an array of high quality, competing products, essentially performed three functions.

- Interfaces through analog-to-digital (A/D) cards with antialias filtering to access live data from any dynamic voltage source. The actual transducer element may involve a piezoelectric crystal, a strain gauge bridge, a linear variable differential transformer (LVDT), or other. This signal is sampled at up to tens of thousands of times per second.
- Via one or more “worksheets,” programmable for either automatic or manual operation, performs various mathematical manipulations and statistical calculations; and writes and reads any of the raw or processed data to/from digital storage.
- Enables remote access via client software to all the same raw and manipulated data, live or historical, from anywhere in the world.

Described here for illustration purposes are two such worksheets that happen to have been used to monitor the six pressure transducers in the hydraulic oil side of the heads of the two triplex diaphragm pumps described in Examples 1 and 2 of this paper. The two worksheets are shown in Figures 48 and 49 and are for reading and processing live data, and for accessing and processing stored data, respectively.

- *Measure and monitor live data*—The boxes in Figure 48 labeled NetIn00 and NetIn01 are “ANALOG INPUT” modules. Each module has three channels to read pressure from each of the heads.
- *Reread stored data*—The boxes in Figure 49 labeled Read DEM and Read SPR are “READ DATA” modules. Each module has three channels to read the pressure from each of the heads previously stored by the LIVECAP.DSB worksheet.
- *Recalibration*—Both worksheets have modules following either the NETIN or READ modules, which are labeled DEM-Recal and SPR-Recal. These are SCALING modules and enable gain and offset adjustments to correct the calibration of the transducer signals, if necessary.
- *Display*—Both the worksheets have identical “Y/t Display” modules used to display the pressure versus time plots, DEM-Time and SPR-Time, and the Fourier analyzed pressure pulsation versus frequency plots, DEM-FFT and SPR-FFT.

• *From the three-head hydraulic pressures, create a single signal that approximates the process pressure in the manifold*—Both the worksheets also have identical sets of modules for this. Those for the suction are slightly different from those for the discharge, but perform the same function. For each suction and discharge, one pressure versus time signal is created from the three for each pump, which essentially represents the pressure during the time that the pump check valve is open to the process for that head. The idea is that, at nearly all times, at least one of the three heads is open to the process and the pressure dynamics are nearly what would be measured if one transducer were installed in the manifold.

• *Discharge pulsation*—The two blocks labeled DEM-MAX and SPR-MAX are “Arithmetic with Two or More Operands” modules. These create one output that is the maximum value at each instant in time of the three pressure signals. Thus, this is essentially the dynamic signal assumed to approximate the pulsation in the discharge manifold. The module that follows is an “Arithmetic with One Constant” and simply subtracts off the mean discharge pressure to produce the same dynamic signal, but centered near zero pressure. This operation is not necessary, but is convenient when viewing both the suction and discharge time signals together on the same plot. This signal is input with that for suction into the blocks for fast Fourier transform (FFT) frequency analyze.

• *Suction pulsation*—The three blocks to prepare this are similar to the two for discharge, except it is first inverted with an “Arithmetic with One Operand” module. This may then be viewed

with the “Y/t Display” module, then is formed into one signal by taking the maximum.

- *FFT frequency analysis*—Both the worksheets also have identical sets of five modules used to produce the pressure pulsation versus frequency plots. The first and fifth are simply “Y/t Display” modules. The three between are:
  - A “Data Window” module with Hanning and magnitude correction,
  - An “FFT” module of a REAL signal and amplitude spectrum, and
  - A “Block Average” module, 10 block average, running with restart.
- *Data Storage*—The worksheet LIVECAP.DSB has two switches to save the TIME data for additional analysis and examination later using the ANALYSIS.DSB worksheet. It also has a switch to save FFT data. The switches are created from “Black Box” modules (not shown). When live data are being analyzed, manually activating one of these switches stores these data for access later with ANALYSIS.DSB or another worksheet.

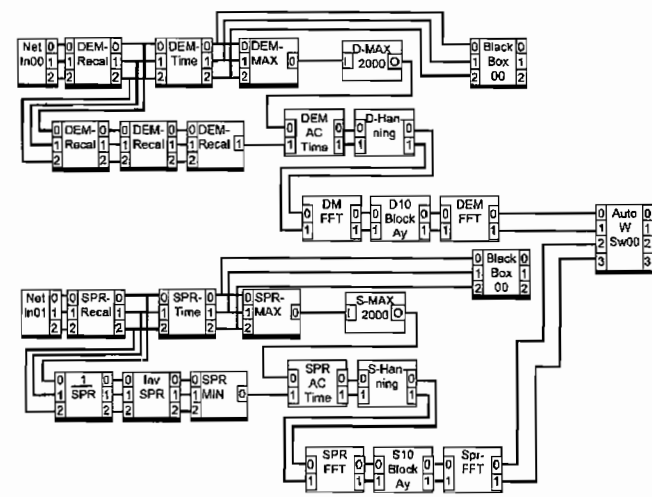


Figure 48. Worksheet, LIVECAP.DSB, for Live Data Capture and Storage.

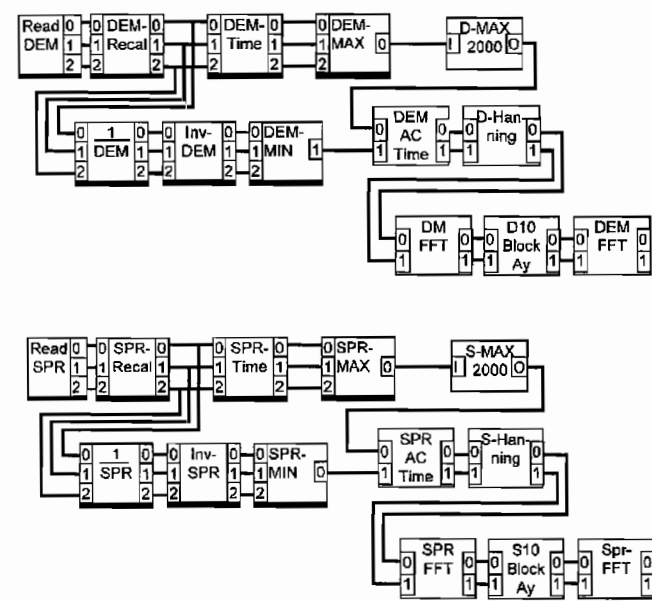


Figure 49. Worksheet, ANALYSIS.DSB, for Analyzing Stored Data.

CONCLUSION

High pressure process diaphragm pumps offer a number of impressive advantages. Yet to enjoy these advantages through a successful operation with low maintenance cost and a minimum of production losses, the pump user needs a basic understanding for oscillating pump technology. This understanding especially is important concerning the interactions of reciprocating positive displacement pumps with the piping system, the behavior of the pump and system requirements when dealing with suspensions, and the NPSH topic.

It is recommended that, for finding the individual optimum in all these topics, it is a great advantage for the pump user to have intensive support from the pump manufacturer and an open-minded dialog about the system and pump design. This is especially important with critical fluid properties and boundary conditions, because the pump manufacturer usually has a broad and often money saving experience.

A new and growing part for increasing reliability and minimizing production interruption intervals are the monitoring of the pump by measuring signals during the process. In that concern, the measurement of the dynamic pressure in the pump heads is a powerful and highly significant tool for the recognition of failure developments. Understanding this type of curve delivers results about leakage flow volume increase, changes in compressibility, pressure relief valve popping, and discharge valve leakage.

Another still very common, but perhaps also powerful, tool is monitoring by using structure-borne noise signals. With this tool, additional failures and failure developments, like suction valve leakage, cavitation, and so on, seem recognizable.

With consideration of the specialties and requirements, and with the satisfying support of the pump manufacturer, the process diaphragm pump is able to be a highly safe and reliable, and therefore highly economical solution.

NOMENCLATURE

A	[m <sup>2</sup> ]	Cross section at suction pipe
A <sub>P</sub>	[m <sup>2</sup> ]	Plunger area ( $\pi D^2/4$ , mm <sup>2</sup> )
a:	[m/s]	Speed of sound
a <sub>p</sub>	[m/s <sup>2</sup> ]	Plunger acceleration
a <sub>F</sub>	[m/s <sup>2</sup> ]	Fluid acceleration
B	[lb <sub>f</sub> /in <sup>2</sup> , psi]	Fluid bulk modulus, inverse of compressibility
C	[Kg/s-bar <sup>2</sup> ]	Flow coefficient for hydraulic oil leakage to reservoir
D <sub>S</sub>	[m]	Inside diameter suction pipe
D	[m]	Plunger diameter
d <sub>i</sub>	[m]	Pipeline diameter
g	[m/s <sup>2</sup> ]	Gravity acceleration
h	[m]	Stroke length
H	[m]	Pressure head
H <sub>D</sub>	[m]	Vapor pressure head
H <sub>S</sub>	[m]	Absolute suction head
H <sub>I</sub>	[m]	Static pressure
$\Delta H_i$	[m]	Pressure head loss in the pump
$\Delta H_A$	[m]	Pressure head loss to due mass acceleration
$\Delta H_F$	[m]	Pressure head loss due to friction
$\Delta H_J$	[m]	Joukowsky pressure shock head
i:	[-]	Number of pump heads (pump cylinders)
K <sub>H</sub>	[1/bar]	Hydraulic fluid compressibility
K <sub>P</sub>	[1/bar]	Process fluid compressibility
L <sub>S</sub>	[m]	Length of suction pipe
l	[mm]	Connecting rod length
m <sub>i</sub>	[Kg]	Mass of process fluid pumped per head per stroke
n,N	[1/min]	Pump speed
NPSH	[m]	Net positive suction head

NPSHA	[m]	Available NPSH
NPSHR	[m]	Required NPSH
P	[N/m <sup>2</sup> ]	Pressure
P <sub>D</sub>	[bar-g]	Mean discharge pressure of the process
P <sub>S</sub>	[bar-g]	Mean suction pressure of the process
P <sub>SV</sub>	[bar]	Pressure in the suction vessel at the fluid surface
$\Delta p:$	[bar]	Maximum pressure amplitude
P	[bar-g]	Pump chamber pressure (bar-g)
r	[mm]	Crank radius, half of stroke S
s <sub>p</sub> = m V <sub>H</sub> /A:	[m]	Excitor amplitude of the pump
t	[s]	Time
t <sub>1</sub>	[s]	Opening time of the pump discharge check valve (seconds)
t <sub>S</sub>	[s]	Opening time of snifter valve (seconds)
V	[m <sup>3</sup> ]	Fluid volume
$\dot{v}$	[m <sup>3</sup> /h]	Flowrate
V <sub>S</sub>	[m <sup>3</sup> ]	Stroke volume
V <sub>C</sub>	[mm <sup>3</sup> ]	Process fluid volume in the pump head at end of suction stroke (V <sub>SW</sub> + V <sub>PD0</sub> )
V <sub>H</sub>	[mm <sup>3</sup> ]	Hydraulic fluid volume in the chamber
V <sub>HD</sub>	[mm <sup>3</sup> ]	Dead volume of the hydraulic-side chamber
V <sub>HL</sub>	[mm <sup>3</sup> ]	Leakage volume of hydraulic fluid per stroke through all leakage sources
V <sub>HR</sub>	[mm <sup>3</sup> ]	Replenishment volume of hydraulic fluid per stroke
V <sub>P</sub>	[mm <sup>3</sup> ]	Process fluid volume in the chamber
V <sub>PD</sub>	[mm <sup>3</sup> ]	Dead volume, actual, of the process fluid
V <sub>PD0</sub>	[mm <sup>3</sup> ]	Dead volume, ideal, of the process fluid
V <sub>SW</sub>	[mm <sup>3</sup> ]	Volume swept by the plunger
V <sub>H</sub>	[mm <sup>3</sup> /Kg]	Hydraulic fluid specific volume in the chamber
v <sub>P</sub>	[mm <sup>3</sup> /Kg]	Process fluid specific volume in the chamber
v <sub>PD</sub>	[mm <sup>3</sup> /Kg]	Process fluid specific volume at mean discharge conditions
v <sub>PS</sub>	[mm <sup>3</sup> /Kg]	Process fluid specific volume at mean suction conditions
w	[m/s]	Fluid velocity
x(t)	[mm <sup>3</sup> /Kg]	Plunger position at time, t, (mm)
$\dot{x}(t)$	[mm <sup>3</sup> /Kg]	Plunger velocity at time, t, (mm/sec)
z <sub>E</sub>	[m]	Geodetic height

Greek Characters

$\epsilon_H$	[-]	Dead volume ratio of the hydraulic side ( $(V_{SW} + V_{HD})/V_{SW}$ )
$\epsilon_P$	[-]	Dead volume ratio of the process side ( $V_{PD0}/V_{SW}$ )
$\eta_V$	[-]	Pump volumetric efficiency
$\eta_{CV}$	[-]	Effect of check valve backflow on volumetric efficiency
$\eta_{HL}$	[-]	Effect of hydraulic oil leakage on volumetric efficiency
$\eta_K$	[-]	Effect of process and oil compressibility on volumetric efficiency
$\eta_W$	[-]	Effect of cylinder wall flexibility on volumetric efficiency
$\eta:$	[mPas]	Dynamic viscosity of the fluid
$\eta_v$	[-]	Volumetric efficiency
$\lambda$	[-]	Pipe friction coefficient
$\rho$	[Kg/m <sup>3</sup> ]	Density
$\rho_{HS}$	[-]	Specific gravity of hydraulic fluid at suction conditions
$\rho_{PS}$	[-]	Specific gravity of process fluid at suction conditions
$\theta_S$	[°]	Snifter valve open duration (degrees (of crank revolution))
$\tau$	[s]	Period of one crank revolution (60/N, sec)

$\omega$	[1/s]	Circular frequency of crank speed ( $2\pi/\tau$ )
$\Omega$	[1/s]	Natural frequency of the system
$\zeta$	[-]	Pressure loss coefficient
$\zeta_S$	[-]	Pressure loss coefficient, suction pipe
$\zeta_V$	[-]	Pressure loss coefficient, pump inlet

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