DIAPHRAGM DEVELOPMENT TRENDS FOR SAFE LEAKFREE RECIPROCATING PROCESS PUMPS

by

Gerhard Vetter

Professor, Apparatus and Chemical Machinery

Eberhard Schlücker

Research Engineer

University of Erlangen-Nuremberg, Germany

Waldemar Horn

Research and Development Engineer

Lewa, Leonberg, Germany

and

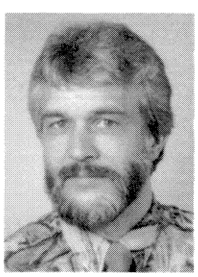
William E. Neis

Sales Manager

American Lewa, Holliston, Massachusetts

Gerhard Vetter obtained his Dipl.-Ing. degree (Mechanical Engineering) at Technische Universität Karlsruhe, Germany. After some years as a Research Engineer in turbomachinery at the same university, he joined Lewa, Leonberg (Germany) as head of the R&D Department. He became Chief Engineer and, in 1970, Technical Managing Director. In 1981, he accepted a chair (professorship) for Apparatus and Chemical Machinery at the University of Erlangen-Nuremberg.

Dr. Vetter has dedicated more than 25 years to research, development and design of pumps and metering equipment.



Eberhard Schlücker is a Research Engineer at the Department of Apparatus and Chemical Machinery at the University of Erlangen-Nuremberg. He leads a team researching in the area of positive displacement and metering pumps.

After receiving a Dipl.-Ing. degree (Mechanical Engineering) in technical college, he worked as a R&D engineer for six years with Lewa. In 1984, he began studies in chemical engineering at the University of Erlangen-Nuremberg. He obtained a Dipl.-Ing. (Univ.) degree there in 1989 and his Dr.-Ing. degree in 1993. As a part of his doctoral thesis, he performed research on design and stress computation of plastomer diaphragms.

Waldemar Horn obtained his Dipl.-Ing. degree at Stuttgart Technical University.

Since 1983, he has been a R&D engineer with Lewa, responsible for diaphragm pump development.

ABSTRACT

The application of diaphragm pumps for metering and conveying of fluids are continuously expanding because they provide zero-leakage, can run dry, have superior efficiency, offer high reliability, maximize safety and minimize maintenance. However, end users should develop an understanding of the functional details involved and the necessity of a systems approach when installing and operating these pumps. The diaphragm design and the various influences on endurance and reliability are evaluated. Among others: material selection, the diaphragm motive system, the pump head/diaphragm, and installation/diaphragm interactions, diaphragm clamping, sandwich diaphragm design, computation of stresses and fatigue for both metal and PTFE diaphragm. Objectives for further optimization of diaphragm designs and comments about economy, performance and reliability close the discussion.

INTRODUCTION

Concerns for protection of the environment are increasing. Legal requirements cove-ring this field are getting more stringent. These conditions pose heavy demands on the manufacturers and users of industrial machinery. In most cases, only hermetically sealed process equipment will be able to meet the resulting restrictions. In this context, leakproof machines (i.e., pumps and compressors) are key components. Diaphragm pumps offer the optimal solution for handling toxic, dangerous, noxious, sensitive, abrasive or corrosive fluids, if favorable hydraulic conditions (high head, low volume) for reciprocating, positive displacement pumps exist.

The diaphragm constitutes the central element in these pumps, simultaneously serving a dual function as a static seal, and a flexible isolation of the working volume by acting as the displacement element. In its role as a static seal, the diaphragm replaces the piston seal of conventional piston type pumps. With the diahragm in direct contact with the fluid, it consequently assures hermetic, leakproof operation. Details of diaphragm pumps are described in the current literature [2, 3, 4, 5, 6].

A review is offered in Figure 1 of the operational ranges of mechanically and hydraulicallly actuated diaphragms, of diaphragm materials, and current power ratings of diaphragm pumps.

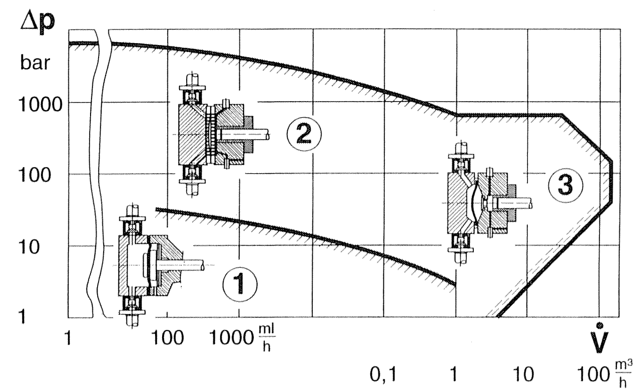


Figure 1. Application Range of Process Diaphragm Pumps: 1) mechanically driven diaphragm; 2) metal diaphragm, hydraulically driven; and, 3) PTFE diaphragm, hydraulically driven.

Characteristic features of diaphragm process pumps for metering or conveying applications are:

- Service life of the diaphragms reliably exceeding 10,000 hr.
- Diaphragms made of polytetrafluorethylene (PTFE) for pressures up to 5000 psig (350 bar) and temperatures up to 300°F (150°C). For higher pressures/temperatures, the diaphragms are made of austenitic steel (or similar materials). This material selection reliably meets the high demands for chemical and thermal durability. The use of elastomeric diaphragms coated with a layer of PTFE is limited.
- The design of the diaphragm clamping area assures a hermetic seal. Diaphragm condition can be monitored while hermetic pump conditions are maintained (sandwich type diaphragm).
- Compatibility with the process plant.
- Compatibility with the automation system of the plant.
- Excellent volumetric and energetic efficiency.
- Safely capable of running dry.
- Capability of selfpriming and automatic selfventing.

The present state-of-the-art and foreseeable trends are reported on for further development of this type of pump, with special attention towards the design of the diaphragm.

THE DIAPHRAGM WITHIN THE ENTIRE SYSTEM

The diaphragm transmits the reciprocating motion of the drive of the pump to the fluid. This inherent mode of operation not only results in a pulsating delivery of fluid, but also in an interaction with the masses of fluid within the pipe system (Figure 2). The diaphragm design has to pay attention to the various loads specific to the way the diaphragm is incorporated into the design of the pumping system.

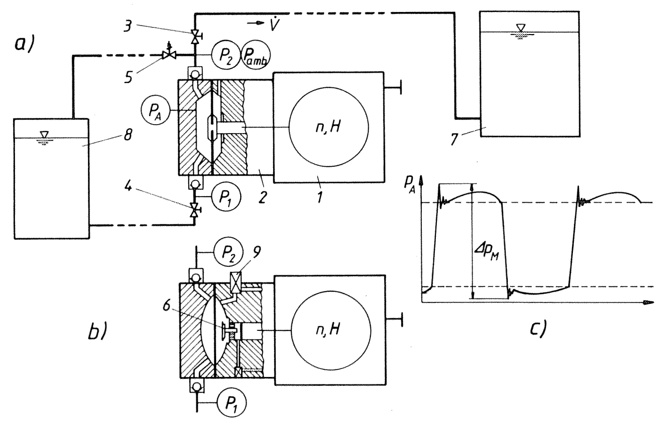


Figure 2. Diaphragm (Pump) within the System: a) mechanically driven diaphragm, b) hydraulically driven diaphragm, and c) tempory working space pressure.

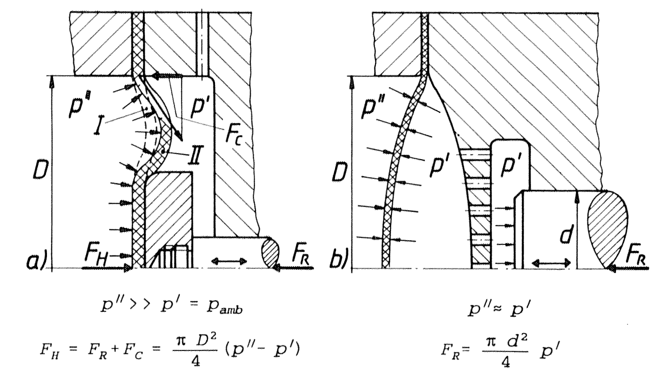


Figure 3. Diaphragm Drive System and Strain: a) mechanically driven diaphragm, b) hydraulically driven diaphragm, and I loaded, II nonloaded diaphragm.

Mechanically Driven Diaphragm (Figure 3 (a))

In addition to the loads resulting from diaphragm deflection while acting as the displacing element, in this configuration, the diaphragm has to support the loads caused by the pressure difference between the fluid side and the drive side of the diaphragm. The load to be supported increases not only proportional to the pressure difference, but also with the square of the diameter of the diaphragm (Figure 2 (a), bottom). For this reason, mechanical actuation of the diaphragm is limited to moderate pressure differences (< 300 psig/20 bar) and power ratings (hydraulic load < 0.75 hp/0.5 kW).

In this application, the diaphragm is not only subject to the effective pressure differences (Figure 2 (a), Δp_M), but also to all shocks and pulsations transmitted by the fluid [7, 8, 9]. In the case where valves 3 and 4 are closed by mistake during operation of the plant, or clogging of the pipe system, or clogging at the valves of the pump, the diaphragm will be stressed additionally, up to the pressure setting of the safety valve 5, by the vapor pressure of the fluid or by pressure shocks due to cavitation.

In applications employing mechanically driven diaphragms, undesirable stressing of the diaphragm must be carefully avoided by the corresponding layout of the entire system. Since the diaphragm is directly linked with the drive mechanism, it will always travel back and forth as designed.

To achieve a dependable return of the diaphragm, a spring of sufficient force must be employed [10].

In practice, mechanically driven diaphragms are mainly used in metering pumps with mechanical or hydraulic lost motion control. In these applications, attention has to be paid to the periodic pressure spikes occurring during reduced stroke length settings, and sufficient damping of these pulses must be provided when necessary.

In general, the operation of diaphragm pumps with mechanically driven diaphragm within a pipe system is not very demanding however, provided pressure shocks due to cavitation or other reasons are safely prevented.

In practically all designs equipped with mechanically driven diaphragms, the diaphragm always stays within a defined distance from the walls of the working space. Dirt particles or solids in the fluids will, therefore, usually not harm the diaphragm.

Hydraulically Actuated Diaphragm (Figure 3 (b))

In this design, virtually no unbalanced external forces are acted on the very flexible diaphragm. The only stresses the diaphragm experiences are due to deflection. The pressure difference is negligibly low and this design therefore adaptable to almost unlimited delivery pressures. Further, the size of the diaphragm has no effect on the stress. These favorable conditions for the diaphragm operation are the reason for the widespread and successful application of this design, since only by these features can a long service life be reliably assured.

In addition to the negligible stress to which the diaphragm is subjected, hydraulic actuation simultaneously permits precise transmission of the displacing action of the piston on the hydraulic fluid, by the diaphragm, on the process fluid. The characteristics of these pumps are virtually pressure-independent, assuring good volumetric and excellent energetic efficiency.

An overflow valve in the hydraulic system (Figure 2 (b), item 9) provides simple implementation of an important safety feature. In many cases, this device makes a separate safety valve in the fluid system unnecessary.

Hydraulic actuation of the diaphragm also involves various design details that influence the interaction with the system in which the pumps are installed.

Hydraulic Loss Replenishment

During the operation of the pump, an infinitesimal, but nevertheless continuous loss occurs in the hydraulic system. The hydraulic piston travels back and forth in its cylinder with a minute clearance through which a small amount of hydraulic fluid escapes. Secondly, automatic air bleed venting in the hydraulic system results in a second small loss. The hydraulic system is kept full by means of the diaphragm position control (DPC) [3]. Volumetric replenishment of the hydraulic fluid returned to the reservoir is performed at the rear dead center position of the diaphragm travel (Figure 4, item 2) by operation of a sliding gate valve (item 1). In Figure 5, the configurations of various approaches to solve this problem are illustrated and evaluated.

Snifting, or replenishment of hydraulic fluid, has to be disabled until the diaphragm reaches the rear dead center of the stroke, otherwise undesirable replenishment could overfill the hydraulic system, overextend the diaphragm and force it against the front wall of the pump head during the forward stroke. In the design of Figure 5 (2), for example, the conical locking pin (item 2) enables the snifting valve (item 4) with pin (item 3) only when the diaphragm (item 2) pushes the actuating disk (with axial holes) back towards the rear wall, which ultimately forms a

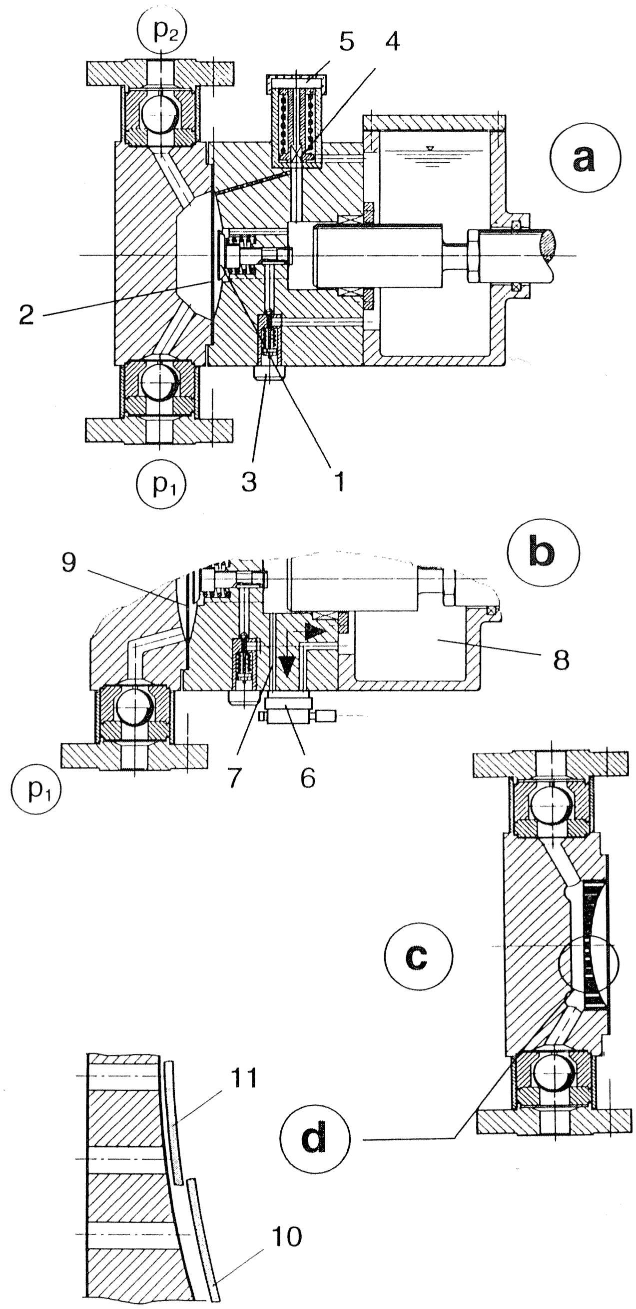
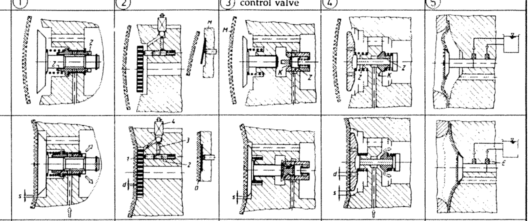


Figure 4. Diaphragm Position Control (DPC): a) principle function DPC; b) details and starting up relieve system; c) additional hole plate; and d) diaphragm position. 1) sliding gate valve; 2) snifting valve; 4) venting valve; 5) relief valve; 6) control valve; 7) connection bore; 8) hydraulic reservoir; 9) rear diaphragm position; 10) normal front diaphragm position; and 11) diaphragm support position.

mechanical stop. The design of Figure 5 (5), with external replenishment control is only used for very large diaphragm pumps in slurry services.

Working Chamber Walls/Diaphragm Interaction

Diaphragm position control (DPC) is essential to the proper functioning of the pump. The diaphragm must be kept a safe distance from the front face of the working chamber during



	Control valve with support disk ①	Mechanical limiting via control pin ②	Separate support disk with central pin and sliding control valve ③	Closing force compensation ④	Inductive diapragm position and control function pickup ⑤
Leakage replacement closed					
Leakage replacement open					
Diaphragm-material	PTFE	PTFE, metal	PTFE	PTFE - EL	EL
Sealing towards the leakage replenishing valve	Cylindrical gap (Z)	Mechanically limited replenishing valve (3)	Cylindrical gap (Z) and contacting seal (K)	Cylindrical gap (Z) and contacting seal (K)	Electromechanical valve
Sensing of the diaphragm position	Movable support disk	Spring loaded sensing disk	Spring loaded sensing disk	Spring loaded sensing disk	Sensing pin jointed to the diaphragm
Support of the diaphragm in the rear limit position	Smooth membrane support area with small gaps (S) for high operational pressure	Support disk with axial holes, operational pressure limited by the diameter d	Smooth membrane support area only with small gaps (S) for high operational pressure	Support area with axial holes (d) and a circular gap (S). Both limiting the operat. pressure	Smooth support area without any gaps or clearances

Figure 5. Morphology of Various DPC Systems.

normal operation (Figure 6 (a)) to avoid perforation of the diaphragm by any particles suspended in the fluid (Figure 6 (d)) or damage to the diaphragm by excessive local stress at discontinuities (e.g., holes) in the surface the diaphragm is forced against (Figure 6 (b)).

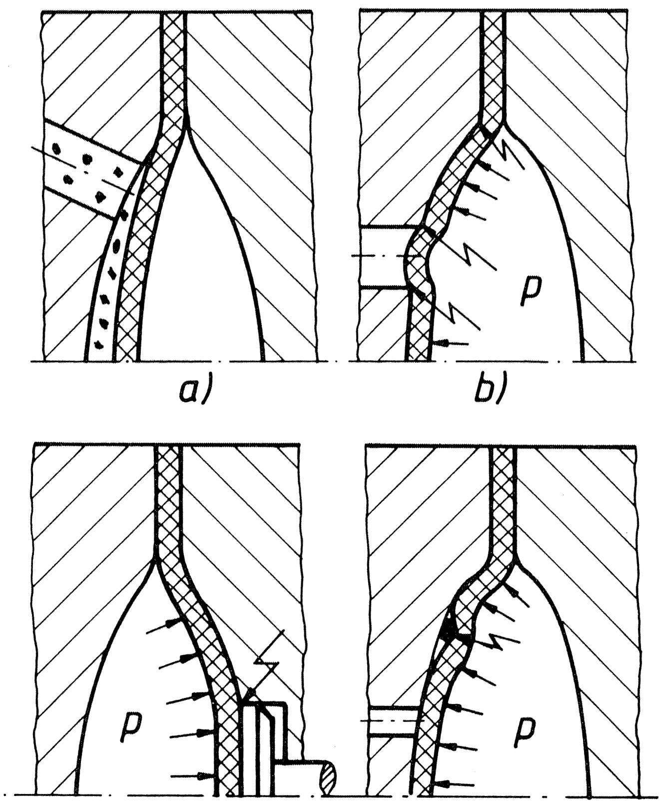


Figure 6. Diaphragm and Working Chamber Walls: a) diaphragm in safe distance from working space walls; b) damages by supporting compression (edges); c) damages by supporting compression (elevated suction pressure); and d) damages by particles.

To safely support the diaphragm in the presence of significant suction side pressure (Figure 6 (c)) requires a rear surface of the working chamber virtually without gaps or opportunities for stress accumulation. The design of these areas depends on the shape and the material of the diaphragm.

Basically, any compressive contact of the diaphragm with the surfaces of the working chamber will cause adhesive forces, implementing harmful asymmetrical local deformations of the diaphragm. To avoid such forces, particularly for metal diaphragms which are very sensitive to these disturbances, the surfaces of the working chamber are sometimes profiled by tangential or radial grooves (Figure 7).

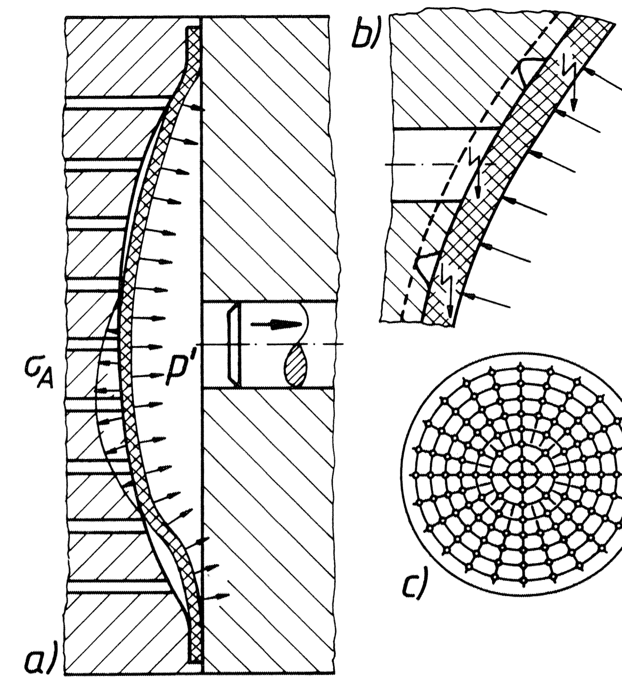


Figure 7. Diaphragm and Adhesive Forces: a) superload by local adhesion; and b) c) surface profile (metal diaphragms).

Underpressure Safety

All hydraulic drive systems for diaphragm pumps presently known are sensitive to long term suction side underpressure, regardless of the system used for replenishment of hydraulic losses. If suction side underpressure persists or occurs frequently, i.e., during standstill intervals, the hydraulic fluid will be aspired slowly into the hydraulic system along the piston clearance (forward leakage) and through the snifting and air bleed vent valve. The diaphragm gradually bulges more and more towards the fluid side and may become overstressed or damaged by being forced against the forward surface of the working chamber by the first stroke after restart.

Attempts to protect the diaphragm can be made by spring forces (which result in additional loading of the diaphragm as a disadvantage), by providing a safety stop for the diaphragm by means of a support disk with axial holes (Figure 4 (c)), by incorporating a specially shaped circular area supporting the diaphragm (Figure 8), by a coupling and sensing piece (Figure 9), respectively, by the total exclusion of underpressure conditions by monitoring the suction pressure level and taking suitable provisions to prevent such conditions.

All concepts providing a mechanical stop for the diaphragm may, however, involve the danger of local wear on the diaphragm if the fluid regularly or by chance contains particles. In

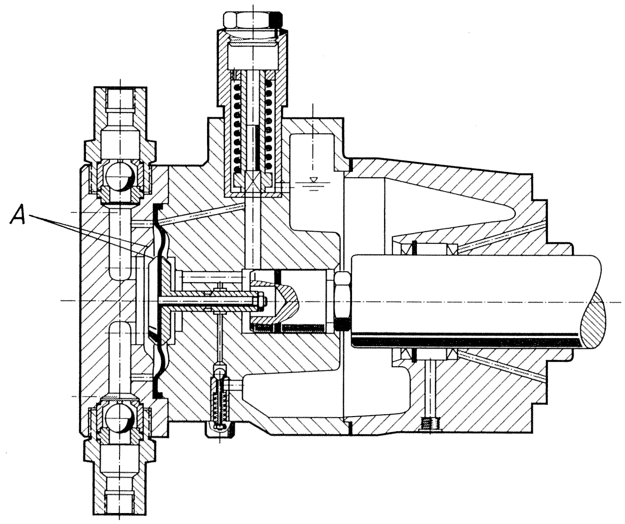


Figure 8. Diaphragm Control by Supporting Area.

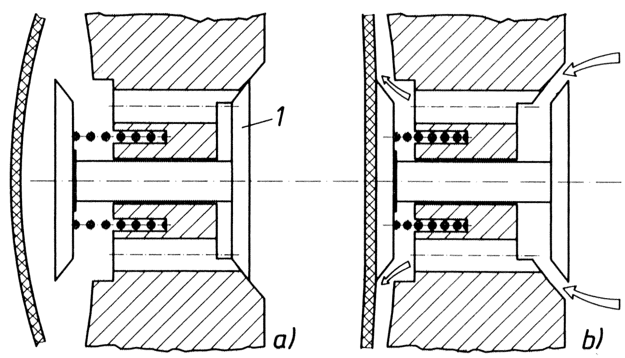


Figure 9. Diaphragm Control by Coupling Gate

plants where underpressure on the suction side can be dependably prevented and where the diaphragm is deflected forward and backward, startup valves (Figure 4 (b), item 6) are used. They open a temporary bypass passage in the hydraulic system (Figure 4 (b), item 7) during startup of the pumps, forcing the diaphragm into the correct position.

In pumps with metal diaphragms (low deflection, special applications), the support disk concept for diaphragm underpressure control (DUC) and safety is generally used (Figure 10). The working space is designed slightly larger than the displacement volume of the diaphragm. Therefore, the diaphragm will normally deflect into the vicinity of the front surface of the support disk, but not touch it periodically (as this is the case in diaphragm compressors for gases).

The snifting valve (Figure 10, item 5) responds in case of underpressure in the hydraulic system, i.e., in the rear dead center position of the diaphragm (underpressure caused by backward leakage of hydraulic fluid into the reservoir). To avoid erroneous replenishment of hydraulic fluid caused by spikes of low pressure on the suction side simulating fluid leakages, the piston can be simultaneously used as control valve (Figure 10 (b), item 6).

Installation/Diaphragm Interaction

Contrasting with the conditions existing in mechanically actuated diaphragms, the balance of the hydraulic forces in the hydraulically actuated diaphragm designs allows them to be relatively unaffected by pressure fluctuations or pulses, as long as the hydraulic system of the pump remains undisturbed.

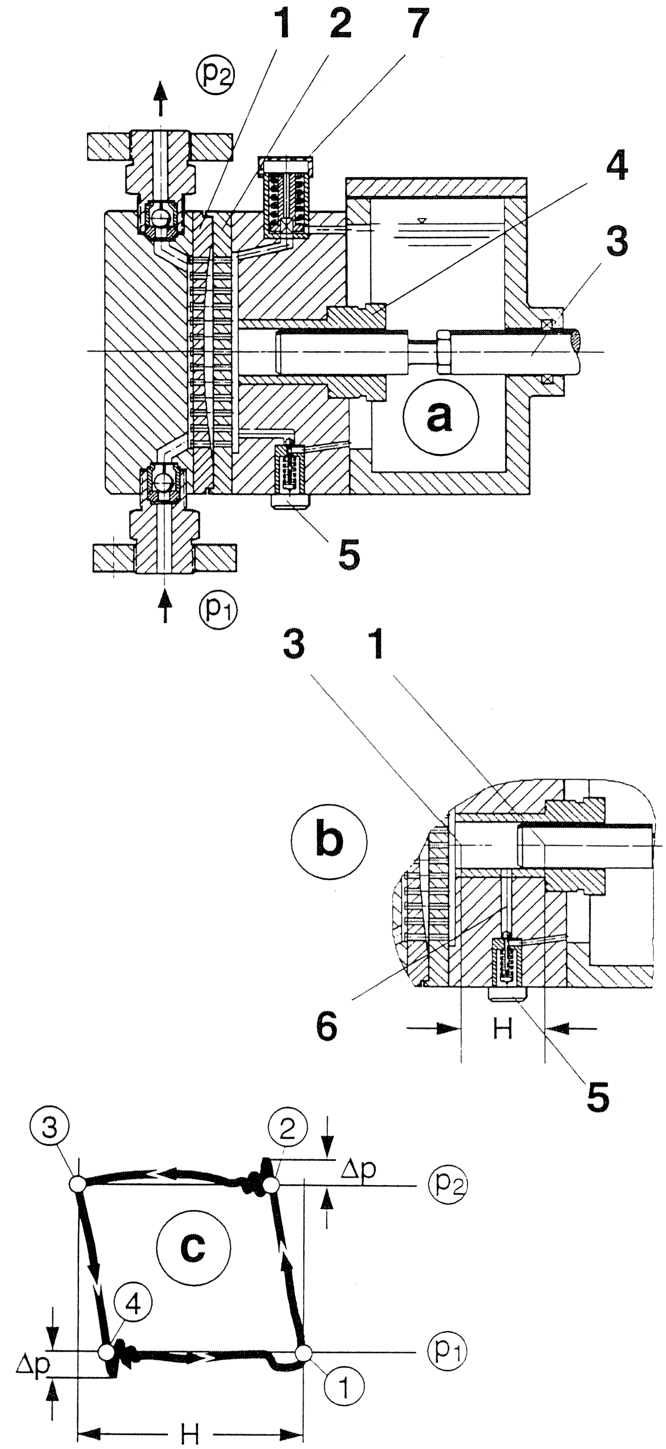


Figure 10. Diaphragm Underpressure Control (DUC): a) principle function DUC; b) DUC with additional sliding gate control by the plunger; c) indicator diagram; and 1,2 hole plates; 3,4 plunger, plunger seal; 5 snifting valve; 6 relief valve; 7 connecting bore.

In case of large pressure fluctuations (Figure 11) caused by pressure shocks or resonances in pressure pulsations however, the pressure amplitudes may exceed the setting of the hydraulic

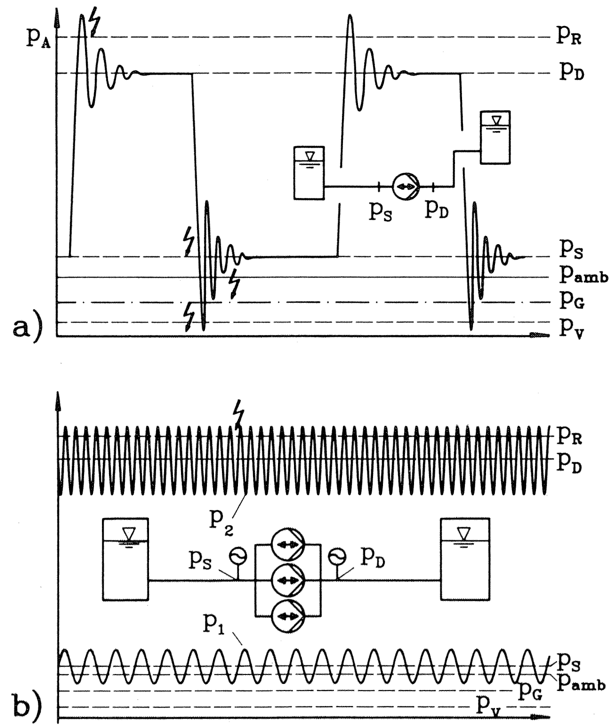


Figure 11. Installation/Diaphragm-Interaction; a) simplex pump (phase-cut by high discharge pressure); and b) triplex pump (resonance situation of pressure pulsations).

relief valve p_R or drop below the vapor pressure p_v of the fluids involved. Disturbances will result, such as

- the relief valve opens frequently ($p_A > p_R$),
- the diaphragm experiences locally excessive stress ($p_A \leq p_V$, cavitation with shocks resulting from implosion of the vapor bubbles), which reduce the service life of the diaphragm.

Furthermore, the process fluid pressure in the pumphead should not fall substantially below the snifting valve pressure setting. This facilitates proper snifting valve operation and keeps degassing of the hydraulic fluid to a minimum. If pumps with hydraulically actuated diaphragms are used, it is highly advisable to perform a pulsation study (API 674) covering the system installation to detect such disturbances early and to suppress them by suitable dimensioning of the system or by the additional installation of dampers and/or filters [7, 8, 9].

THE DIAPHRAGM—THE HEART OF THE PUMP

Within the loads as determined by the entire system (see chapter 3) the diaphragm is expected to meet the required service life (> 10,000 h) in a compact design and a wide range of compatibility with different fluids and fluid temperatures. Considerable potential for improvements and further developments still exists (Figure 12).

Diaphragm Materials and Clamping

The properties of the materials predominantly used for diaphragms are compared in Table 1, clearly indicating the large differences in the modulus of elasticity, which is very important for the longterm deflection.

Metal diaphragms consist of cold rolled sheets (thickness 0.2 to 0.5 mm) of austenitic chromium-nickel steels, respectively, and other metals or alloys with appropriate corrosion resistance

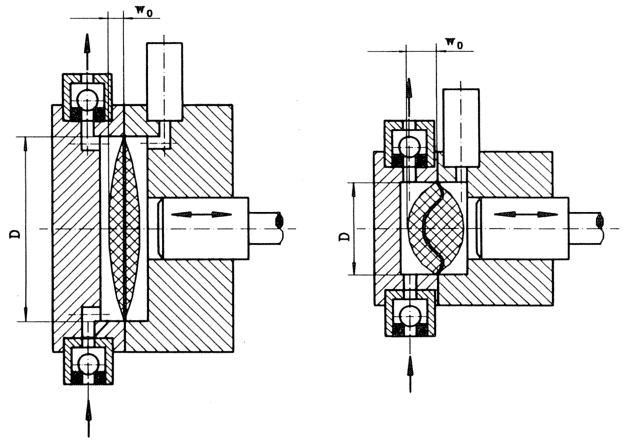


Figure 12. Reduction of Pump Size by Optimal Diaphragm Design.

Table 1. Diaphragm Materials.

	metals M	plastomers PTFE	elastomers EL
modulus of elasticity	2,1 x 10^5	700	5 - 20
strain properties	-	+	++
chemical resistance	++	++	o
thermal resistance	++	+	o
strenght	++	o	o
shape stability	++	-	++
elasticity	--	o	++
notch sensitivity	--	++	-
permeation tightness	++	o	+

(hastelloy, titanium, etc.). Cold rolling assures high strength and good toughness of the material. Until recently, due to ease of manufacture, disk shaped parallel-plane diaphragms are used.

In addition to the cases requiring better chemical resistance than that of PTFE, there are applications requiring elevated thermal resistance (up to 500°F/260°C, limited only by thermal stability of the hydraulic fluid) and high mechanical strength, a favorable property as far as clamping is concerned (Figure 13). These are the prime reasons for using metal diaphragms. Metal diaphragms are always clamped directly at up to 70 percent of the yield strength. The absolute absence of pores represents another important reason for using metal diaphragms (zero permeability).

To avoid fatigue in continuous service, metal diaphragms must only be stressed within the elasticity range, for the service life with respect to material fatigue decreases dramatically if 0.2 percent yield strength is exceeded. At the same time, as higher strength materials are used, they become very sensitive to notches (brittle).

PTFE diaphragms are usually cut out of sheets foils (with typical thickness of 0.020 in/0.5 mm to 0.080 in/2 mm) peeled from sintered blocks of the material. The mechanical properties of the sintered material depend on the polymer chain length, the size of the polymer particles, the manufacturing process, the sintering procedure and the crystallinity of the material. The diaphragms are used in parallel plane or thermally pressed undulation formats.

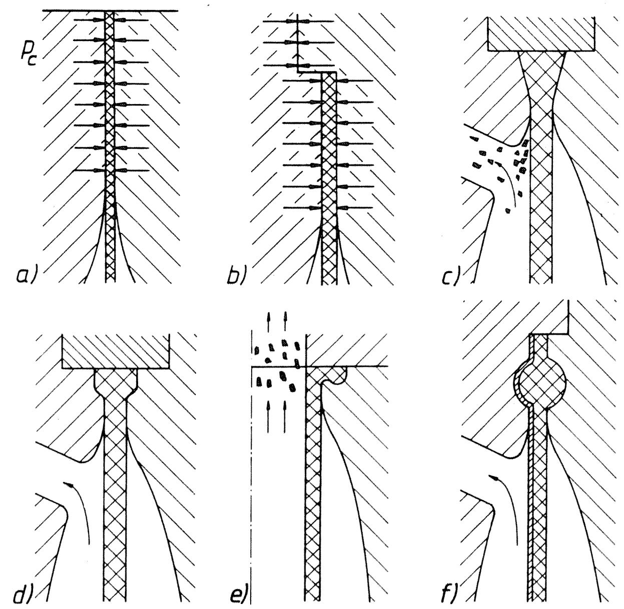


Figure 13. Various Clamping Features: a) direct (metal); b) shunt force (restricted compression, plastomer); c) d) various clamping geometries (elastomer); e) tube diaphragm (elastomer); and f) O-ring profile (PTFE lined elastomer).

Their typically superior chemical resistance, together with good resistance to heat (up to a limit of approximately 300°F/150°C) are the prime reasons for popularity of PTFE diaphragms.

The properties concerning mechanical strength are determined by the viscoelasticity of the material. At any level of stress PTFE deforms, to a large extent in plastic fashion (nonlinear, Figure 14) and shows a tendency towards creeping, which is time dependent and gradually lowers stress conditions (Figure 14, detail K). For this reason, PTFE is comparatively insensitive against notches. Even under dynamic loads, the stress peaks of tension are reduced by means of local plastification. Accordingly, this viscoelasticity forms a favorable property in the face of the loads acting on the diaphragm. Other possible materials for diaphragms from the field of plastomers, such as ultra high molecular polyethylene (PE-UHM) are only advantageous in isolated cases.

Due to its sintered structure, PTFE always exhibits some amount of permeability, which can be reduced to technically acceptable low levels by proper selection of the size of the particles, the manufacturing process and sufficient thickness of the diaphragm.

Clamping of PTFE diaphragms is performed mostly with shunted forces (using the principle of limited compression, Figure 13 (b)) acting on sufficiently large clamping areas. By means of grooves in the clamping areas lip type sealing effects and local stops against creeping due to viscoelasticity can be created.

Following the assembly of PTFE diaphragms into pumps, the diaphragms obtain their final operational shape by plastification during the first cycles of actual operation. The stresses performing this initial deformation always exceed the range of elasticity. Thanks to the viscoelasticity, PTFE can carry relatively high elastic/plastic dynamical strains with a high level of endurance.

Elastomer diaphragms are mostly hot molded, diskshaped, undulated parts, frequently in a multilayer design (with thickness > 0.10 in/3mm) and also often with reinforcing fabric. Typically, the elastomers, acryl-nitril rubber (NBR), ethylen-propylen-dien rubber (EPDM), butyl-rubber (IIR), fluor-rubber (FRH), have significantly lower resistance to chemicals and heat as compared with PTFE, but are impermeable and fairly dimensionally stable.

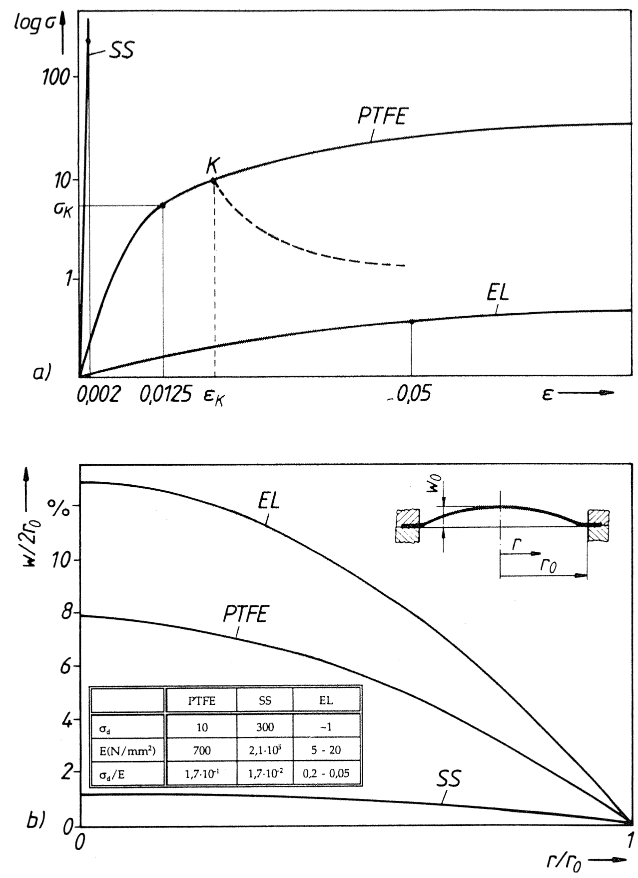
The high tolerance for elastic deformation and, consequently, a large displacement intensity represent the principal reasons for the use of elastomer diaphragms. For a number of reasons, this potential can unfortunately be used for limited process applications. Elastomers are sensitive to notches (stresses at local edges, notches and sharp angles), show limited resistance to chemicals and heat, and, in typical applications, require a multitude of different material types have be kept in stock to meet application specific requirements.

Elastomer diaphragms are always clamped with shunted forces and feature lip, wedge, or O-ring shaped cross sections in the clamping area (Figure 13, item c and d). Tube type diaphragms are feasible and offer operational advantages in the delivery of slurries (Figure 13, item e). Coating elastomer diaphragms with PTFE has some potential for further investigations and future applications. However, in the face of the low thickness of the layer of PTFE (< 0.020 in/0.5 mm), the porosity turns into a problem and limits the applications.

Material Selection

The principal criteria in the selection of diaphragm materials are:

- the resistance to chemical attack and to heat,



	PTFE	SS	EL
σ_d	10	300	~1
E(N/mm²)	700	$2{,}1\cdot 10^5$	5 - 20
σ_d/E	$1{,}7\cdot 10^{-1}$	$1{,}7\cdot 10^{-2}$	0,2 - 0,05

Figure 14. Stress/Strain Correlation (A) and Deflection Potentials of Plane-Parallel Diaphragms (SS Stainless Steel, PTFE Polytetrafluorethylene, El Elastomer).

- the maximum allowable deflection of the diaphragm (high displacement intensity).

In sealless diaphragm pumps for industrial processes, PTFE diaphragms are presently the most widely, since they offer a maximum of displacement intensity, and chemical and thermal resistance. Figure 14 (b) shows an example of parallel plane diaphragms. The diagram is based on parameters experience has shown being tolerable for long term strain during dynamic loads: SS 0.2 percent, PTFE 1.25 percent, EL 5.0 percent.

Additional criteria for entering the selection procedure are:

- capability for cleaning.
- capability for sterilization.
- tightness.

Presently, the potential for developments in the field for diaphragm materials is largely exhausted. Attempts to improve the performance must, therefore, concentrate on the optimization of the shape of the diaphragms.

Potential for future developments still exists in the field of quality assurance. Particularly for PTFE, careful analyses of the production and preparation parameters offer room for improvement.

Diaphragm Design

Within the range of requirements, safety considerations are paramount for sealless process pumps. In diaphragm design, sandwich configurations—double wall barriers, provide the ultimate in safety with: redundancy, true secondary containment, and diaphragm condition monitoring. The optimal design of diaphragms, therefore, is dominated by the quest for:

- long service life.
- high displacement intensity.
- safe operation.

Diaphragm Shape and Clamping—Sandwich Design

During direct or shunt type clamping, the applied compression results in radial extrusion of the diaphragm material (Figure 15 (a)) and accordingly, results in the real installed shape of the diaphragm. Parallel plane diaphragms assume, for example, a shallow domed shape and actually only reach this by deformation to their appropriate degree of displacement intensity.

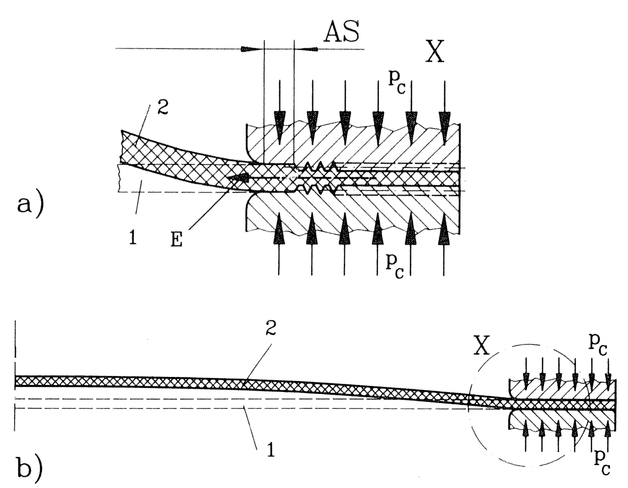


Figure 15. Diaphragm Deformation by Clamping Extrusion: 1) initial; 2) after clamping; and (E) Extrusion.

Sandwich type diaphragms require careful design of the details of clamping (Figure 16) since, in addition to hermetic sealing, attention has to be paid to the signalling channel towards the pressure sensor to avoid any obstructions, and to assure the dependable operation of this important safety device.

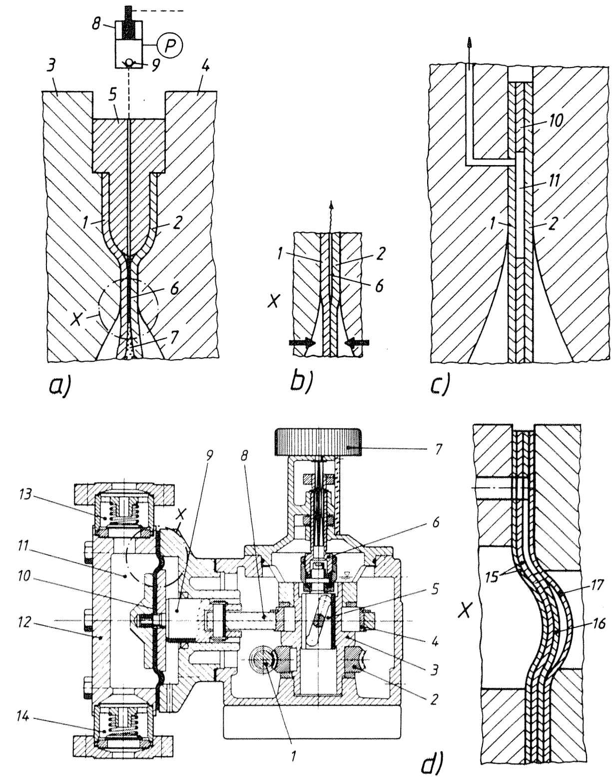


Figure 16. Sandwich Diaphragms: a) hydraulically driven PTFE diaphragm; b) after compression coupling; c) hydraulically driven metal diaphragm; and d) mechanically driven PTFE diaphragm.

For hydraulically actuated PTFE diaphragms, a configuration (Figure 16 (a)) is suitable. Following assembly, the two sheets of the diaphragm are coupled by the pressure acting on them and eventually by an auxiliary inner fluid (Figure 16 (b)). The passage (item 6) to the sensor (nonreturn valve + pressure sensor, items 8) and 9)) must remain unobstructed. The clamping has to be designed to assure dependable function and response of this important safety and maintenance related feature.

For hydraulically actuated metal diaphragms, a fine slot (item 11) in the central diaphragm (item 10) offers the only solution, in most cases. This slot should, however, interfere as little as possible with hermetic clamping of the diaphragm assembly or affect the operation and service life of the diaphragm.

For mechanically actuated PTFE diaphragms, a multiple layer sandwich-configuration as shown in Figure 16 (d) proves to be optimal. The two active layers (item 15) with a slotted layer in between and a normally unloaded layer (item 17) are combined into a package. The normally unloaded (redundant) diaphragm serves as a temporary safety barrier in case both of the two active diaphragms suffer any loss of integrity.

Diaphragm Stress

Different concepts for the shape of the diaphragms (Figure 17) result in specific loads and stresses to which the diaphragm is subjected. It should be noted that a plane diaphragm as shown in

Figure 17 (a), widely used due to its simple production, assumes the shape of a shallow calotte after installation and the first operational strokes.

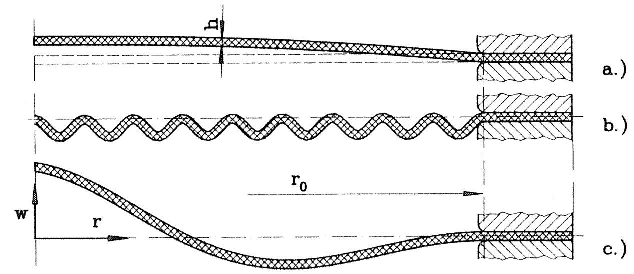


Figure 17. Diaphragm Design Features.

Parallel plane diaphragms are mainly subjected to biaxial membrane stresses. Particulary in the clamping area, however, uniaxial stress due to flexing dominates (Figure 18 (a)).

In undulated, or corrugated, diaphragms the deformation of the diaphragm is transferred to bending of the undulations, which results in lower stress at the outer circumference (Figure 18 (b)).

In case of plus/minus deformation of the diaphragm, the stress at the clamped rim is alternating. Furthermore, all bending

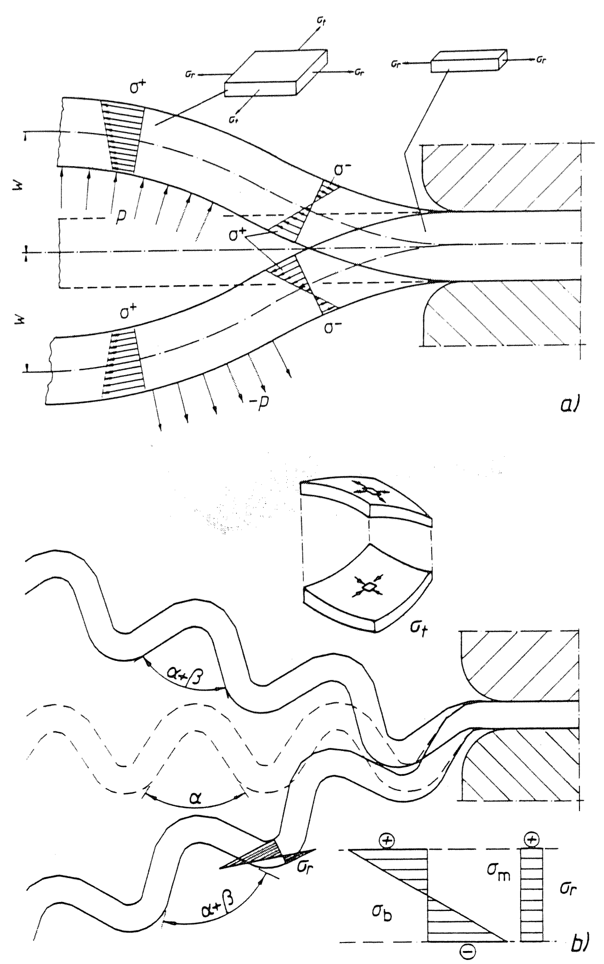


Figure 18. Stressing Planeparallel (A) and Undulated (B) Diaphragms.

stresses are alternating, while the socalled membrane-stresses however remain pulsating only. In undulated diaphragms for plus/minus deformation, the tangential stresses develop alternating additionally.

The "zero transition" of the diaphragm merits special attention. Parrallel plane diaphragms (already conditioned into the domed shape) can pass through this situation with only transitional (partial) buckling (Figure 19). As practice has proven, stress levels large enough to induce ruptures may occasionally occur during these passages. In sandwich type diaphragms, the stress levels are in a somewhat better position for stable zero transitions due to their radial elasticity.

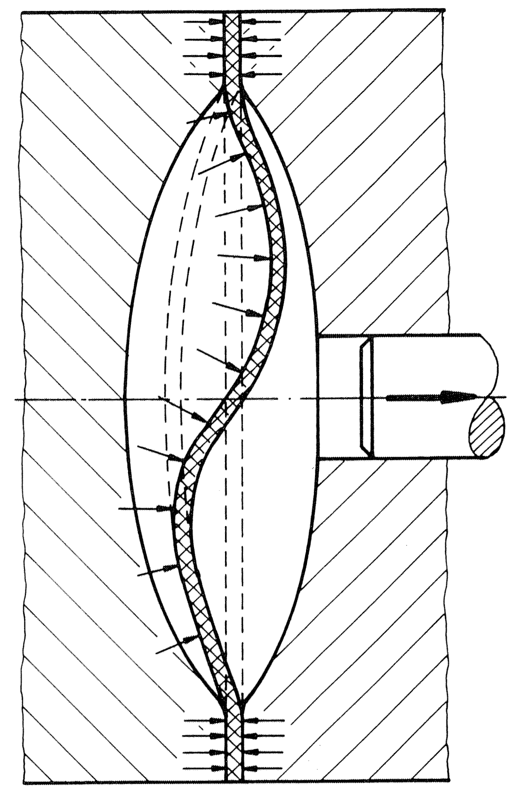


Figure 19. Buckling at Zero Transition (Schematical).

For the design of diaphragms not experiencing fatigue, the following remarks concerning specific properties of the applied materials should be added:

The design of metal diaphragms is based on the "theory of flexible plates" [11], using the characteristic fatigue limits for the metal sheets. In freely flexing diaphragms used for sealless process pumps, high bending stresses ocurring mainly at the rim (Figure 20). During normal operation, frontal supports are shunned, since disturbances may result from adhesion of the diaphragm to the surface. Some potential for future developments still exists in the field of suitable rim area support. In this context, attention has to be paid to local notch stresses caused by impressed particles, as cold rolled sheet metal is quite sensitive against notches (Figure 21, a case of actual damage shown in Figure 22). For metal diaphragms, buckling during zero transition, respectively, at surfaces the diaphragm rests against, should be avoided [12].

Until recently, the design of PTFE diaphragms was only carried out empirically, as permitted by the favorable properties of PTFE. Suitable methods for computation of the occurring

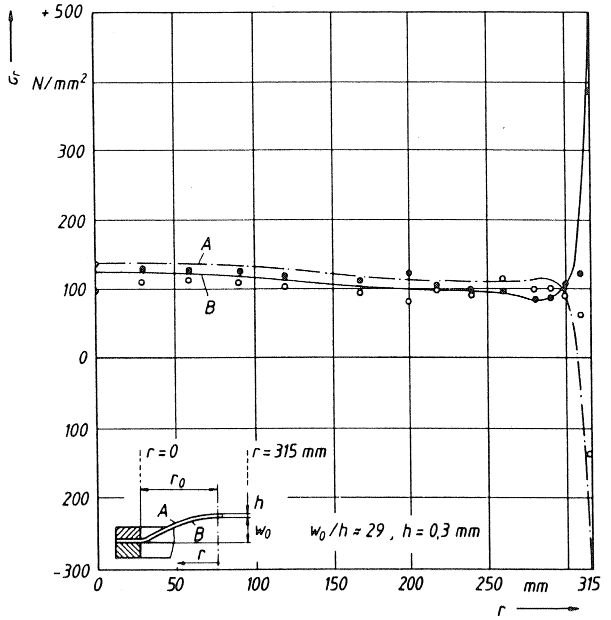


Figure 20. Maximum of the Radial Stress σ_r of a Free Flexing Metal Diaphragm, Calculation (Full Lines), Measurement (Dotted Lines) [11].

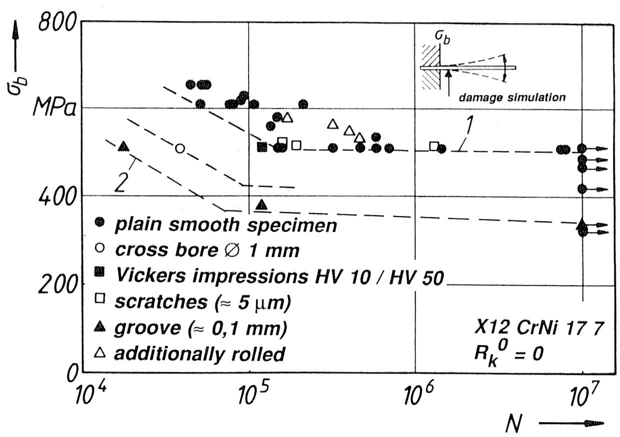


Figure 21. Fatigue of Stainless Steel Sheet Material at Various Notch Effects: 1) smooth, 2) notched, and N) number of cycles.

stresses were not available and knowledge about the values for permanent strength of PTFE were nonexistent.

According to recent investigations [13], the occurring stresses can be determined by means of the method of Finite Elements (FE) and the use of nonlinear characteristics of the materials (Figure 23).

Some relevant techniques are:

- two-dimensional rotationally symmetric grid structure,
- structuring of the FE-elements in four layers, and
- implementing of the clamping and contacting problems.

Experimental measurements of the deformations occurring verify the results of the computations. Also, according to recent investigations [13, 14] further characteristic data for the "fatigue limit" of PTFE can be determined by means of hysteresis measurement. In this precedure, the stress in a specimen of the

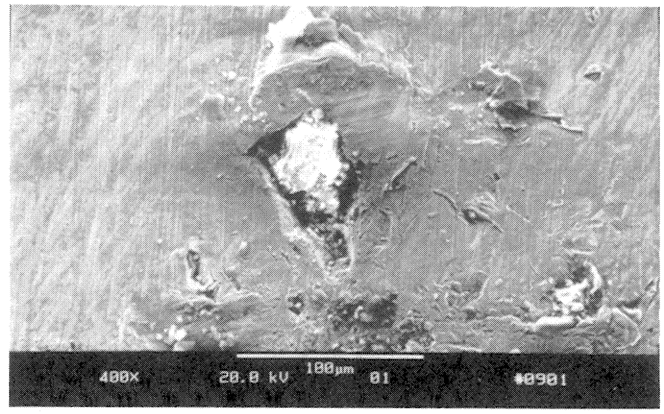


Figure 22. Failure of a SS Diaphragm by Notching Effect through a Sand Particle (Rupture after 2500 h, 600 Bar).

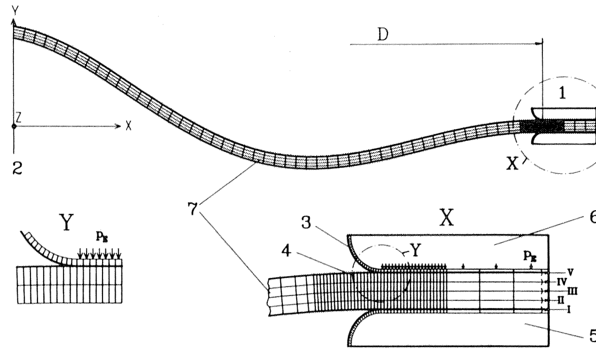


Figure 23. Two-Dimensional FE Network of a Clamped Undulated Diaphragm: 1) clamping ring; 2) diaphragm center; 3) contact line; 4) contact elements; 5/6) clamping comtour; 7) diaphragm; and 8) plane of the nodes.

material is increased in steps, each time following several hundred load cycles (Figure 24 (a)).

The typical phase shift between stress and strain of viscoelastic materials is the basis of the hysteresis method. The stress/strain cycle yields a hysteresis loop (Figure 24 (b)). The maximum stress/strain is derived as shows Figure 24 (c). The "fatigue limit" results from two intersecting straight approximation lines. For a certain PTFE-type, the method yields longterm permissible pulsating stress of about 1015 lb/in^2 (7N/mm^2) and as alternating stress of about 800 lb/in^2 (5-6N/mm^2).

While steels experience fatigue for pulsating and alternating stresses, the fatigue properties of PTFE have proven to be different. As explained earlier, the relaxation properties of PTFE avoid perforation at a specific strain by the self reduction of the stresses. This can be applied to dynamic load also, and explains the experimental experience that pulsating stressing at constant strain, which is characteristical for pump diaphragms, does not exhibit noticable fatigue. With alternating stressing however, the relaxation due to change in sign (+ to -) of strain is compensated during each cycle. Consequently, alternating stressing of PTFE is not time-dependent and, therefore, more critical. Experimentally, this was confirmed with various diaphragm shapes experimentally during fatigue tests.

Bending at the Clamping Rim

The large uniaxial strain, characterized by the rim contour of the clamping components, is normally alleviated by a moderate

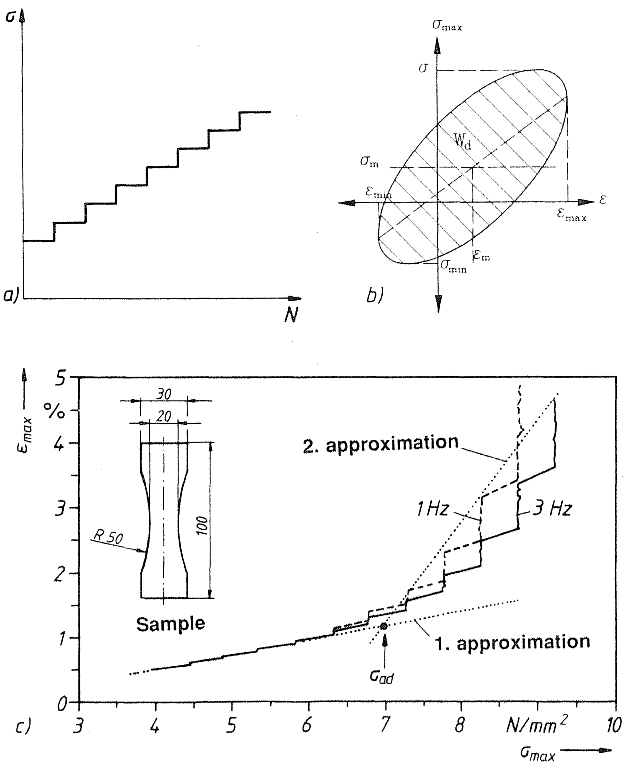


Figure 24. Characterization of PTFE at Dynamic Load Conditions: a) b) hysteresis method; and c) determination of admissible stress σ_{ad}

radius (r = ca 0.040 in/1.0 mm, to avoid particle sedimentation). The strains in that area of the diaphragm, which regularly exceed the plastification range of PTFE, reveal a remarkable behavior of the material; a so-called yield hinge is created and, thus, the high strain at the clamping rim can be handled without fatigue. In the area of the yield hinge the macromolecules assume a special orientation and, consequently, superior dynamic strength, most probably due to the change from a spherolitic to a fibrel structure [15], which can be clearly recognized in Figure 25. This phenomenon obviously is the key for the adequate understanding of the outstanding endurance of plan-parallel PTFE diaphragms.

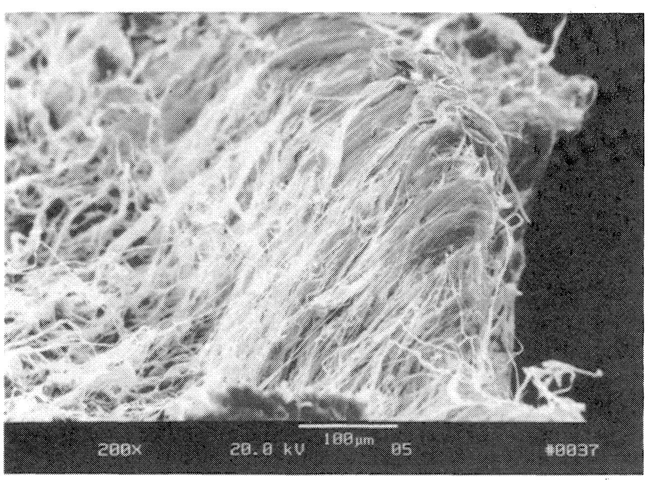


Figure 25. Fibral Structure in the Ruptured Cross Section of a Yield Hinge with PTFE.

OBJECTIVES OF THE DIAPHRAGM SHAPE DEVELOPMENT

The potential for optimization potential seems mainly to be concentrated in the area of the shape design. Plan-parallel types offer only minor possibilities, but undulated types offer a much broader range of possibilities for improvment. Research toward the optimum design should be directed at variation in number and amplitude of the undulations.

With mechanically actuated diaphragms, the support stabilizes the diaphragm shape and a single, large, convolution is most desirable.

The potential for further development centers around the undulation geometry and the clamping areas, especially concerning the rigid central diaphragm support.

With hydraulically actuated diaphragms, local elastic domes are created by the forced deflection towards the end of the diaphragm travel. The deflection resistance of the diaphragm requires a certain deformation pressure. If an undulation exhibits insufficient stiffness, the dome contour is locally buckled (Figure 26). This procedure develops with minimal energy, which means that buckling starts at one location (or several places at the same time) implementing a collapse and the larger danger of perforation. The reason rotationally symmetrical undulated diaphragms buckle, is exessive tangential compression stress. The same effects may occur when nonsymmetrical zero transition combined with increased deformation pressure. The buckling behavior of various diaphragm shapes is definitely influenced by diaphragm thickness, radii of the undulations and their locations, with undulations at larger diameter positions being more endangered.

As a consequence, nonbuckling and safe diaphragms for hydraulical operation require a moderate number and amplitude

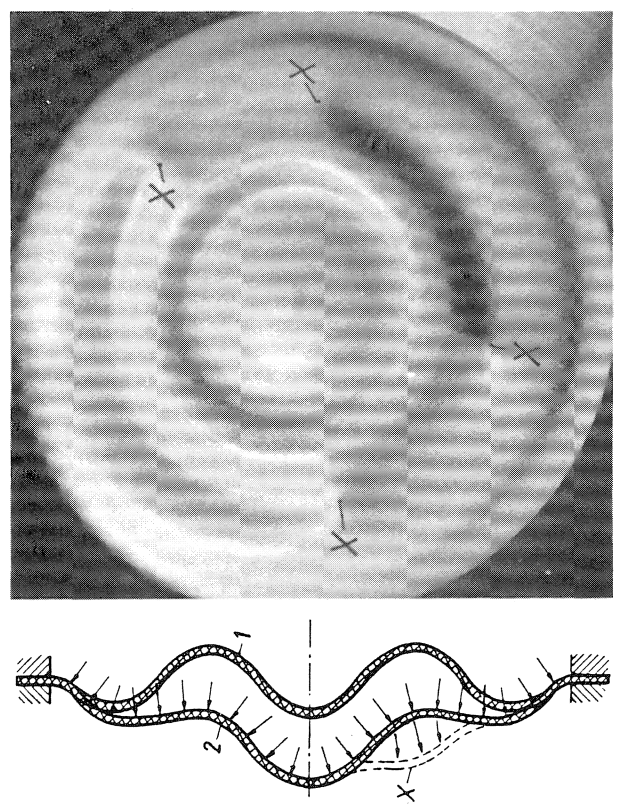


Figure 26. Local Buckling of Diaphragms. 1) inital; 2) deflected, and X) buckling locations.

modifications to valves (including changes in lift, spring preload, spring construction and stiffness, valve disc geometries, etc.) will be examined using data from actual systems.

OVERVIEW OF PUMP VALVE RELATED PROBLEMS

Many problems with reciprocating pumps have been shown to be due to improper valve operation. In an optimum system, the suction and discharge valves would open and close at the precise instant to facilitate the suction or discharge stroke. However, the plunger can change velocity faster than the valves can respond, resulting in "valve lag." If a discharge valve lags on opening, higher than normal pressures are created in the pump cylinder (working barrel) since the plunger continues to compress the liquid in the cylinder until the valve opens. When the valve finally opens, the pressure in the cylinder is higher than the discharge line pressure. The rapid buildup and release of this pressure at the beginning of the discharge stroke is called an over pressure spike. Similarly, if the suction valve lags on opening, the pressure in the cylinder will be rapidly reduced until the suction valve opens. This rapid reduction and return of pressure at the beginning of the suction stroke is called an under pressure spike. If the pressure falls below the vapor pressure, cavitation can result. As the suction pressure returns to normal, vapor bubbles formed during cavitation will collapse, creating high amplitude "cavitation spikes." Even with properly operating valves, lag of a few degrees is inevitable. However, factors such as spring preload, stiffness, valve disc mass, valve lift, and pump speed can significantly increase the valve lag.

Pressure time data taken in the working barrel from a triplex pump that had an over pressure spike problem are shown in Figure 1. The over pressure spikes act on the plungers and the resulting forces are transmitted to the rods, crankshaft, bearings and frame. This additional dynamic pressure therefore increases the load induced stresses on the power end and the liquid end components and can contribute to fatigue failure of these parts.

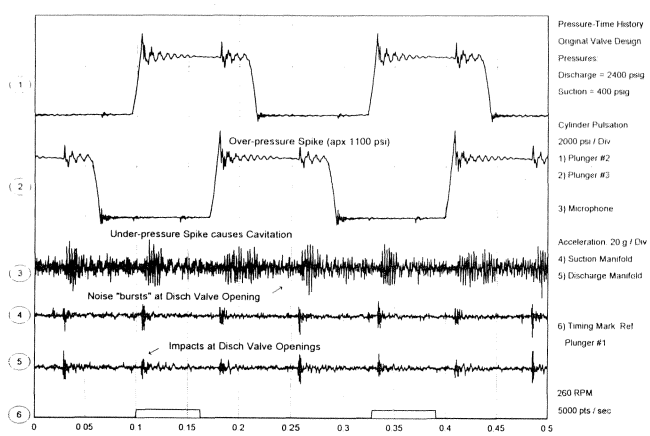


Figure 1. Pressure Time Histories—Triplex Pump With Over Pressure Spike.

The over pressure spike is the resultant of several combined effects including:

- Viscous adhesion (sticktion) of the valves to the seat (which depends upon sealing area, surface finish, disc flexibility, and fluid properties)
- Plunger side/line side dynamic pressures
- Differential area (unbalanced valve area)
- Acceleration of valve disc (due to changes in running speed)
- Spring preload and stiffness
- Valve mass

Data obtained from numerous field measurements by EDI have indicated that, when significant over pressure spikes occur, they are most often due primarily to the sticktion effect. In addition to potentially causing damaging forces and cavitation, the sticktion effect creates a reduced pressure area near the center of the sealing surface. At the reduced pressure location, cavitation may result on both the suction and discharge valves. The resulting cavitation pits on the discs and seats are usually concentrated near the center of the sealing surface. These pits are often mistaken for foreign object damage (Figure 2).

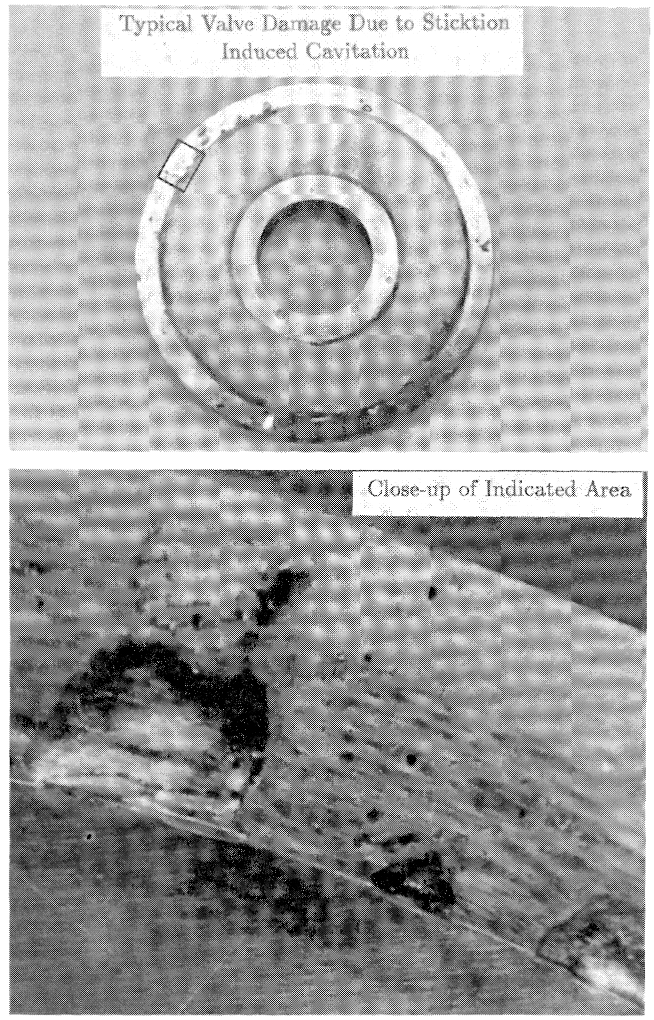


Figure 2. Typical Stricktion Induced Valve Disc Damage.

FIELD TESTING OF RECIPROCATING PUMPS

Field data are the primary diagnostic tools for analysis of pump valve behavior. The data can ultimately be used to provide the basis upon which analytical calculations depend. However, the data must be of a high quality to be useful.

A general technique of field data acquisition has been developed that has been used extensively to obtain the required data for use in analyzing valve problems. Instrumentation, acquisition software, and data analysis techniques have been combined to deliver the desired characteristics. Each of these aspects are discussed in the following sections.

Instrumentation

Several different types of instruments are needed to quantify pump behavior. A typical installation on a vertical triplex pump is illustrated in Figure 3.

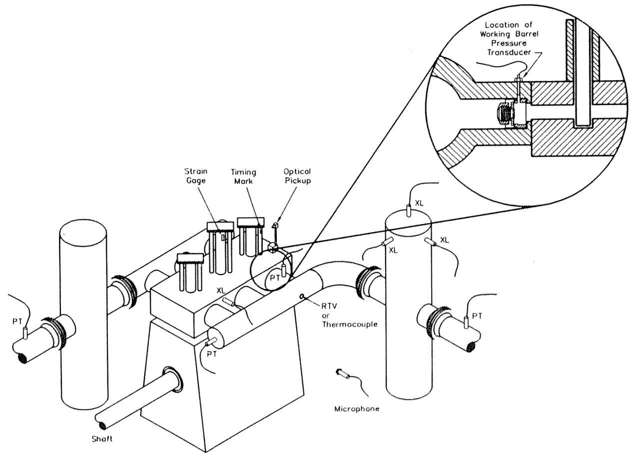


Figure 3. Typical Reciprocating Pump Instrumentation.

Pressure Transducers (PT). Pressure time data should be obtained in one or more pump cylinders (working barrels), in the suction and discharge manifolds, and at locations in the piping where high vibration levels occur. These data are usually obtained with dynamic pressure transducers. Sensitivities of 5.0 to 10 mV/psi are usually the best choices for pulsation measurements in the piping and manifolds. Since the change in cylinder pressure from suction to discharge pressure fluctuates over a wide range (typically 3000 psi), low sensitivity (1.0 mV/psi) transducers should be used for measurements in the cylinders. Static pressure measurements are sometimes needed to evaluate the system with regard to cavitation, or to quantify certain process conditions that may be important to pump operation. Strain-gage type transducers are most often used for this purpose. These transducers require more elaborate signal conditioning and calibration than piezo-electric transducers. The static transducers can also be used for dynamic measurements, if the conditioning amplifiers have sufficient frequency response.

Accelerometers (XL). With appropriate signal conditioning, accelerometers can be used to measure low-frequency piping vibration. However, it is important to measure high-frequency energy, too. Cavitation, over pressure spikes, and other "impact" energies often result in high-frequency vibration in the pump. Accelerometers mounted on the pump manifolds are useful for measuring this energy. These acceleration data can be used to identify problems with specific cylinders or valves. To obtain the required high frequency response, the accelerometers should be rigidly mounted (i.e., not attached with magnets).

Strain Gages. Where failures have occurred, it is sometimes useful to install strain gages to measure the strain levels at the location of the failures. Although these instruments require a good deal of effort to install, the resulting data are usually worth the additional effort. Acquiring strain data on reciprocating or rotating components may require that the data be telemetered (although some success has been achieved with direct wiring of gages to reciprocating parts).

Strain gages attached to the plunger can also be used to infer cylinder pressures when it is not possible to install cylinder pressure transducers. It should be noted, however, that the data can be distorted due to excitation of plunger lateral natural frequencies, friction, and inertial effects.

Microphones. Some problems are discovered because operations personnel hear atypical noises emanating from the pump. Microphones installed in the near-field can be useful in correlating perceived noise with other phenomenon that might be occurring.

Thermocouples/RTDs. For problems involving cavitation, it is often important to measure the fluid temperature at the pump. Thermocouples and RTDs can be easily installed to provide this information.

Timing Mark. Much of the data acquired will be analyzed in the time domain. A timing mark is required to help identify specific portions of the pumping cycle and to identify improperly operating valves. Magnetic pickups, optical pickups, or proximity probes can be used to provide a once per revolution pulse. The pickups can be installed to sense a keyway or reflective tape on an exposed shaft. An optical sensor is often used to sense the passage of a reflective tape on the side rods (Figure 3).

Tape Recorder. A multichannel FM or digital recorder should be used to ensure that data can be recorded for later playback. This capability is invaluable for capturing data from a startup, a shutdown, and "trip" situations. The recorder should be adjusted to provide at least a 5.0 kHz bandwidth.

Data Acquisition

The data acquisition system should be capable of acquiring data in both the time and frequency domains. Time domain data are used to evaluate the operation of the pump (i.e., valve lag, pressure buildup, etc.). Frequency domain data are usually used to evaluate system resonances. Given the ability of portable computers to manipulate and store information, time domain data are most efficiently acquired using analog-to-digital (A/D) conversion hardware coupled with a computer.

Most reciprocating pumps operate at speeds below 300 rpm, which would seem to indicate that only low-frequency data are required. However, the rapid pressure buildup and reactant flow "bursts" from each cylinder can generate energy to frequencies of 3000 Hz or more. Therefore, sampling rates as high as 6000 Hz per channel may be required. A typical test will usually involve 10 to 15 channels of data that should all be acquired simultaneously for comparison of data. Since most A/D boards multiplex their inputs, the sampling rate requirements for 15 channels of data could be as high as 90,000 Hz. (Multiplexing is a technique whereby a single A/D converter can be used for many channels. This technique reduces the expense and complexity of the A/D hardware with only minor tradeoffs in acquisition speed and quality.) Typically, data will be acquired for periods of one sec up to one minute. The acquisition system could be required to manipulate as much as 10 Mbytes of data (15 channels at 6000 Hz sampling rate per channel, one minute of acquisition) per captured event.

Data Analysis

Analysis of the acquired data can be a time consuming task. Acquisition software capabilities can lessen the burden of manipulating the large amounts of data that result. A system has been developed at EDI that has been shown to be effective at acquiring the necessary data. System capabilities are outlined below.

Channel Management. A facility is provided to manage and display transducer sensitivities, descriptive text, and channel numbers.

Acquisition Control. Parameters such as sampling rates, and acquisition time are easily controlled.

Acquisition Triggers. Acquisition of data are sometimes desired only when specific events occur. Typical events include a pulse from the timing mark, a peak—peak dynamic signal level or a certain static pressure. A pretrigger (beginning acquisition *before* the trigger occurs) is provided to add flexibility to the triggering.

Filtering. Sometimes, the signals include unwanted "noise." Digital filter algorithms are used to remove the unwanted noise. Low-pass, high-pass, band-pass, and band-stop filtering have all been required.

Data Display. Some or all traces may be presented simultaneously on the "page." The orientation and placement of each trace is easily adjustable. Capabilities for detailed documentation and annotation of data (to note specific test conditions, time and date, etc.) are also provided. Hard copy may be obtained by a variety of devices, including both raster and vector devices.

Data Storage/Retrieval. Acquired data may be stored for later recall. Facilities are available to import/export data from/to other software packages. This capability allows further manipulation of data (e.g., to compute a single channel of principal strain data from a rosette of three strain gages).

SOLVING PUMP-VALVE RELATED PROBLEMS

Using the previously described field data acquisition techniques, many different types of pump and system problems have been addressed. In some cases, it is necessary to utilize analytical capabilities to fully understand a problem. Case histories of actual pump installation are described in the following sections.

Pump Component Failures Due to Over pressure Spikes

Several triplex pumps experienced drive train and fluid-end component failures that, when the pumps were inspected, appeared to be the result of excessive loads. When operating, high amplitude impact noises could be heard, as if internal components were "knocking." Ground-borne vibration could be felt by personnel standing near to the pump. Typically, valve life was short. After only a few hours of operation, the valve discs and/or seats became pitted as if foreign object damage had occurred (Figure 2). Often times, the valve disc or seat developed a series of pits at the center of the sealing surface that joined together to form a ring that appeared to have been machined into the surface.

One of the triplex pumps was instrumented as described earlier with pressure transducers in the pump cylinders and manifolds, and with accelerometers mounted on the pump manifold. A near-field microphone was also installed. Initial data (Figure 1) showed that high amplitude (40 percent to 50 percent) over pressure spikes were occurring at the beginning of the discharge stroke.

Cavitation due to under pressure spikes was also observed at the instant the suction valves opened. Sound measurements showed noise bursts at the instant of discharge valves opening (Figure 1). Accelerometers mounted on the pump manifold also showed impact energy which correlated with the opening of the discharge valves, and sometimes with the opening of a suction valve.

The over pressure spikes generated impact forces. Knocking noises occurred as the impact force traveled through the power end. These impact forces were apparently responsible for the damage that had been experienced. In the suction side, cavitation resulted when the under pressure spike caused the suction pressure to fall below the vapor pressure. It was thought that the delay in opening of the suction and discharge valves could be due to sticktion.

As discussed earlier, sticktion is a phenomenon that produces a force that holds the valve disc onto the seat and retards the opening of the valves. The magnitude of the sticktion force is a function of the width of the sealing surface of the valve disc. A wide seal will produce a much higher force than a narrow seal. Therefore, to reduce the sticktion force, the valve *sealing* surface area must be reduced (considering only the potential for sticktion, the ideal seal surface would be a knife edge). However, if the valve *seating* surface area is too small, impact stresses in the valve disc and seat will be too high, resulting in valve damage. The authors have determined that a groove pattern cut into the valve disc or seat may be used to reduce the sealing area, which can sometimes reduce the sticktion force without significantly affecting the seating area. A typical groove pattern for a disc is shown in Figure 4.

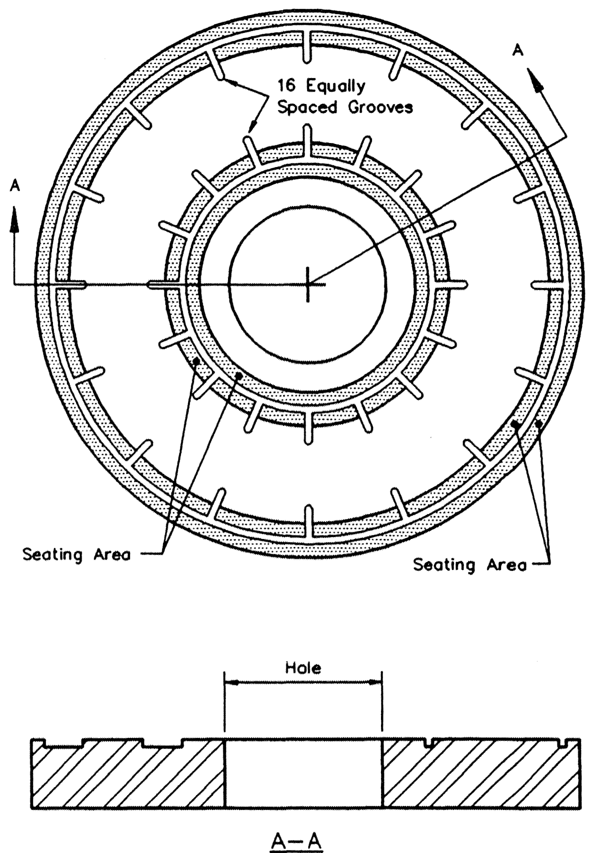


Figure 4. Groove Pattern to Reduce Sticktion.

A set of valve seats were machined with the groove pattern shown in Figure 4 and were installed in a pump. It was decided to machine the grooves into the valve seat rather than the disc, because it was felt that the modification would be more permanent. When the pump was started, knocking noises were no longer present and personnel standing near the pump also indicated that the ground-borne vibration could no longer be felt. Data acquired in the pump cylinder showed that the over pressure spike had been nearly eliminated (Figure 5). Cavitation spikes in the suction side were reduced. Impact energy (acceleration) measured at the pump manifolds was also significantly reduced.

Flowrates with the modified valve seats were 6.5 percent higher than with the original valve seats. This initially surprising result was found to be due to a decrease in valve lag from 16.4 to 11.7 degrees. Discharge valve lag causes "backflow" from the discharge into the working barrel at the beginning of the suction stroke. Similarly, suction valve lag causes backflow from the

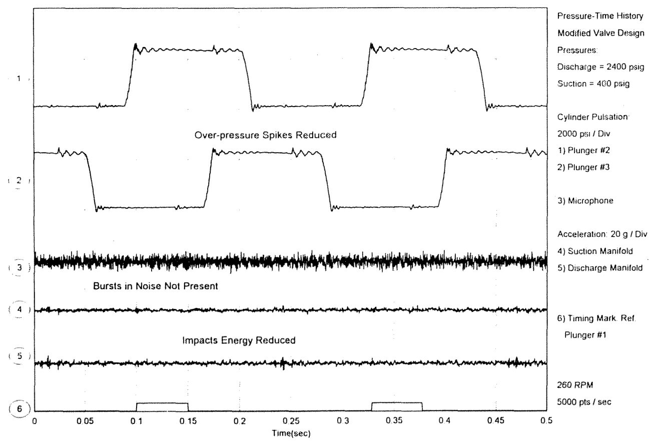


Figure 5. Pressure Time Histories—Over Pressure Spike Reduced.

cylinder back into the suction as the discharge stroke begins. The suction and discharge valve lag reduce volumetric efficiency—i.e., a decrease in flow.

The pumps were operated for approximately one year with the valve seat modification. No additional failures were reported. However, when the valves were inspected, the valve discs appeared to be "forging" themselves into the valve seat. Although the valves were operating satisfactorily, it was felt that the effective groove pattern was reduced. Therefore, the discs were replaced with new valve discs. Even though the one year valve life was much improved over the original design, alternative valve disc materials are currently being investigated to extend their lives.

Use of Valve Dynamics Analysis

It has been shown to be possible to reduce the over/under pressure spikes by cutting grooves in the valve. It is also possible to alter other valve design parameters (i.e., fluid viscosity, valve disc mass, spring rates, etc.) to affect over/under pressure spikes. While the specific modifications may be arrived at by trial and error, it can be more effective to use an analytical tool (valve dynamics analysis) to evaluate the effects of different designs before installing them. As will be demonstrated, field data, combined with analytical analysis, can be used to achieve better results in actual operation.

Valve Dynamics Simulation Techniques

A computer based dynamic simulation technique has been used to determine the sensitivity of the valve displacement time history and the cylinder pressure time history to the various pump and valve parameters. The valve motion and cylinder pressure are determined by numerical integration of the governing differential equation of motion. The discontinuities due to valve impact on the seat or stop are handled by integrating between "smooth" portions of the solution using restarts with new initial conditions to switch to the next piece-wise smooth portion of the solution.

A sketch showing pertinent parameters of the pump cylinder-valve model is given in Figure 6. A plunger (piston) of diameter D is driven by a crankshaft of radius, *r*, and a connecting rod of length *l*. The clearance volume of liquid changes with time, as does its pressure. The valve discs are assumed to be rigid bodies, and the impacts are assumed to be perfectly plastic (no rebound).

The valve disc mass is acted upon by the forces of the spring (K), and by two damping components, C^1 and C^2, which are coefficients for damping force proportional to the first and

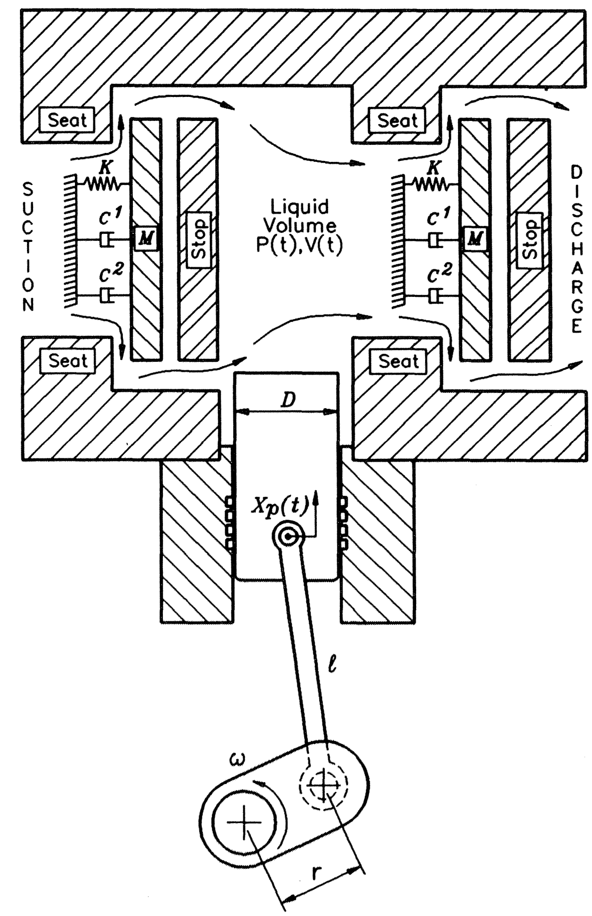


Figure 6. Pump Cylinder Valve Model Parameters.

second powers of the velocity of the valve disc. A pressure loss factor for the valve is determined based on empirical data, with the volumetric flow through the valve.

$$Q = A\sqrt{\frac{2\Delta P}{K\rho}} \tag{1}$$

where:

- Q = volumetric flow through valve
- A = time variable area
- ΔP = pressure drop across valve
- K = valve loss coefficient
- ρ = fluid mass density

The governing differential equations for the system are:

$$\frac{dP_{cyl}}{dt} = \left\{Q_{SV} - Q_{DV} + \left(\dot{x}_p \cdot \frac{\pi D^2}{4}\right)\right\} / C_v \tag{2}$$

$$M_{SV}\ddot{x}_{SV} + C^1_{SV}\dot{x}_{SV} + C^2_{SV}|\dot{x}_{SV}|\dot{x}_{SV} + K_{SV}x_{SV} = \sum F_{SV} \tag{3}$$

$$M_{DV}\ddot{x}_{DV} + C^1_{DV}\dot{x}_{DV} + C^2_{DV}|\dot{x}_{DV}|\dot{x}_{DV} + K_{DV}x_{DV} = \sum F_{DV} \tag{4}$$

where:

C_v = volume compliance
B = liquid effective bulk modulus
$\dot{x}_p$ = velocity of plunger
D = diameter of plunger
Q_{SV} = volumetric flow rate — suction
Q_{DV} = volumetric flow rate — discharge
P_{cyl} = (uniform) pressure in clearance volume
M_{SV} = disc mass—suction valve
M_{DV} = disc mass—discharge valve
K_{SV} = spring rate—suction valve
K_{DV} = spring rate—discharge valve
C^1_{SV} = damping coefficient 1—suction valve
C^1_{DV} = damping coefficient 1—discharge valve
C^2_{SV} = damping coefficient 2—suction valve
C^2_{DV} = damping coefficient 2—discharge valve
x_{SV} = displacement of suction valve from seat
x_{DV} = displacement of discharge valve from seat with subscript SV specifying the suction valve, and subscript DV specifying the discharge valve.

The right-hand side terms ΣF_{SV} and ΣF_{DV} represent forces acting on the suction and discharge valve discs due to fluid pressure, and include:

- Viscous adhesive (sticktion) force acting on valve due to velocity of separation of disc from the seat.
- Differential pressure across valve acting on effective area of valve

The time domain solution proceeds with assumed initial conditions at time t = 0. The initial plunger position corresponds to the maximum chamber volume, so that the plunger begins compressing the liquid in the clearance volume. The differential equations are integrated numerically in time with all of the pertinent pressure, displacements, flows and forces calculated at each time step. Cylinder pressure and valve displacement and velocity values as a function of crank angle are written to disk files for post-processing.

Sticktion effects can be modeled by considering the case of two parallel surfaces immersed in a liquid (Figure 7). The viscous adhesive force is [1]:

$$F_s = -\frac{\mu b^3}{(y + e_o)^3} \cdot \frac{de}{dt} \cdot L, \tag{5}$$

where:

F_s = sticktion force acting on seat
μ = liquid absolute viscosity
b = width of seat
e = film thickness
e_o = initial film thickness
y = displacement of valve disc from seat
L = circumferential length of seat.

This effective sticktion force is due to the pressure profile created over the width of the seat as the fluid fills the void created by the separation velocity. This effect is also referred to as the Bernoulli effect.

From this equation, it can be seen that the valve seat width *b* has a strong influence on the sticktion force. Therefore, this dimension is an extremely important design parameter of the valve assembly. The sticktion force is also proportional to the fluid viscosity, and the force is strongly dependent on the initial effective film thickness, e_o, which is influenced by the surface finish of the disc and seat, and the degree to which the surfaces are in intimate contact. The, e_o, parameter can change as the valve wears and with pressure differential. Different valve disc materials may conform more or less to the seat, changing the effective e_o even though the design may be dimensionally identical. Therefore, it is extremely difficult in practice to evaluate e_o accurately, making model normalization using measured data a prerequisite to obtaining valid simulation results. Note also that since the separation velocity appears in the equation, it would be expected that higher pump speeds would result in higher sticktion induced pressure overspikes. Field data have indeed shown this to be the case.

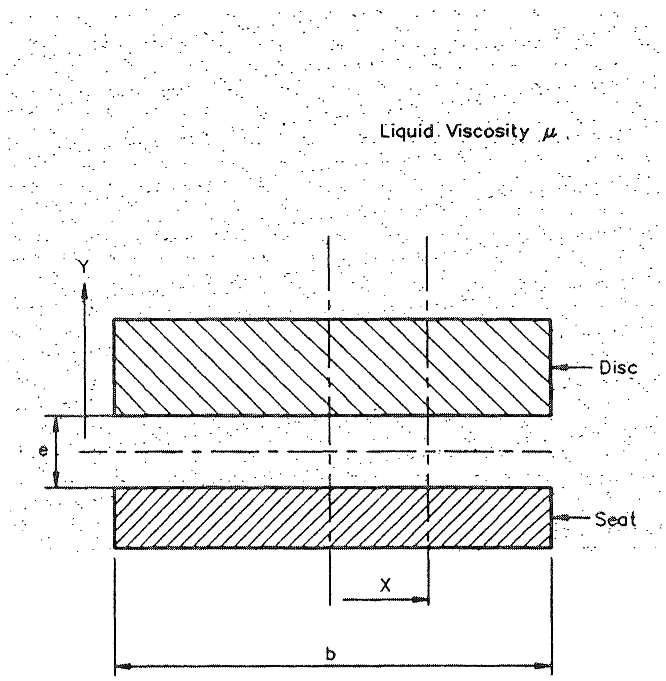


Figure 7. Model for Sticktion for Immersed Surfaces.

Coupling Analysis with Field Data

A crude oil pump system in operation for a short period of time experienced fatigue failures of small bore piping (vents and drains) attached to the main pipe. Due to the critical service of the pump, a field study was done to determine the cause(s) of the piping failures. Instrumentation was installed similar to that shown in Figure 3. After the piping was repaired, strain gages were attached at the locations of the failures and the pump was operated. The data indicated that the failures were the result of excessive vibration of the piping "stubs" at their structural natural frequencies (which were in excess of 200 Hz). This fact seemed unusual, since there is normally little energy generated by the pump at these frequencies. Further data acquisition and analysis showed that the energy exciting the resonant vibration was not from pulsation, but was an impact energy which was being mechanically transmitted from the pumps through the piping and support structures.

The source of the high frequency energy was determined to be impact forces generated by high amplitude over pressure spikes in the cylinders (Figure 8). One obvious solution to eliminate the piping failures was to add braces to reduce the differential vibration between the vent/drains and the main piping; however, there were many similar fittings in the piping system which made it impractical to brace all of the fittings. Therefore, it was

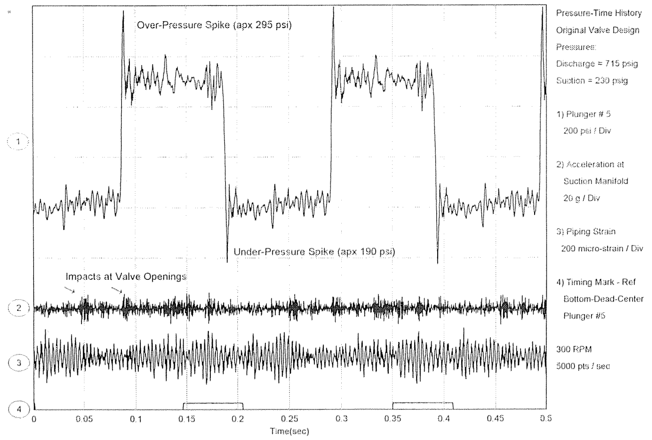


Figure 8. Comparisons of Over Pressure Spike, Impact Acceleration, and Piping Strain.

decided to reduce the over pressure spikes by modifying the valves, which, it was hoped, would also reduce the high-frequency vibration of the stub piping.

Initially, the groove pattern shown in Figure 4 was cut into the valve discs. When the pump was operated, reductions in over/under pressure spike amplitudes were not significant. Rather than proceed on a "trial and error" basis, it was decided to utilize the valve dynamic analysis tools to predict a groove pattern that would be effective.

Numerous valve designs were analyzed. A sketch of the valve disc is shown in Figure 9. The pump characteristics are outlined in the table below.

Parameter	Value
Operating Speed	150 & 300 rpm
Liquid Density	61 lb/ft^3
Liquid Viscosity	30 μreyns
Stroke	7.0 in
Bore	4.75 in
Suction Pressure	230 psia
Discharge Pressure	715 psia
Suction and Discharge Valve Disc Weights	1.7 lbs
Suction and Discharge Spring Constant	112 lb/in
Suction and Discharge Valve Lift	0.46 in
Valve Spring Preload	54 lbs
Valve Disc Material	Delrin

The results of the computer analysis are presented in time domain graphs of the plunger pressure, suction valve displacement and velocity, and discharge valve displacement and velocity (Figure 10). The simulation results at 300 rpm predict an over pressure spike of approximately 285 psi (1000 psi spike minus 715 psi static discharge pressure). Under pressure spikes of 175 psi caused the pressure in the cylinder to fall to the vapor pressure, initiating cavitation spikes. The valve displacement graphs indicate that the valve discs would impact the valve stops during opening. This agreed with the damage that had been experienced on the back side of the valve discs and the broken springs which occurred after a short operating time.

For reduced speed operation of 150 rpm, the peak pressure was reduced to approximately 825 psia which is equivalent to an over pressure of 175 psi. The under pressure spike was computed to be 70 psi above vapor pressure. Field data obtained for these two cases correlated well with the simulation results (the 300 rpm case is shown in Figure 8).

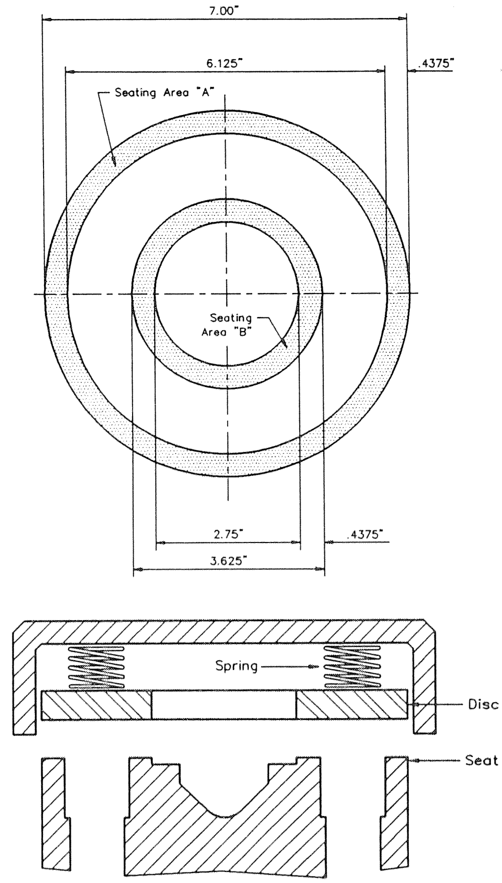


Figure 9. Typical Pump Valve Disc and Seat.

To reduce the over pressure spike, it was proposed that the discharge valve discs be modified, as shown in Figure 11. To reduce the under pressure spike, the suction valves were similarly modified. The concept of the modification was to reduce the width b of the sealing surface which, as shown in Equation (5), directly influences the sticktion force by its value cubed. The results of the computer simulation of these modification are shown in Figure 12. Both the over pressure and under pressure spikes were significantly reduced at 300 rpm. In addition, the valve vertical displacement (lift) was no longer hard against the stop. This would reduce the impact force against the valve stop and would reduce the damage to the springs. Similar improvements were obtained at 150 rpm.

Field data acquired after the valves were installed are shown in Figure 13. As shown in the pressure time histories, the amplitudes of the over pressure and under pressure spikes were significantly reduced. Strain gage data also showed a significant decrease in strain amplitudes at the stub piping. A summary of the measured and computed over/under pressure spike amplitudes at 300 rpm is shown in the table below.

	Original Design		Modified Design	
Data	Over-pressure	Under-pressure	Over-pressure	Under-pressure
Field	295 psi	190 psi	120 psi	100 psi
Analytical	285 psi	175 psi	96 psi	88 psi

Other designs (Figure 14) were analyzed that showed a further reduction in over pressure spike amplitude. Although the computer analyses indicated that additional improvement could be

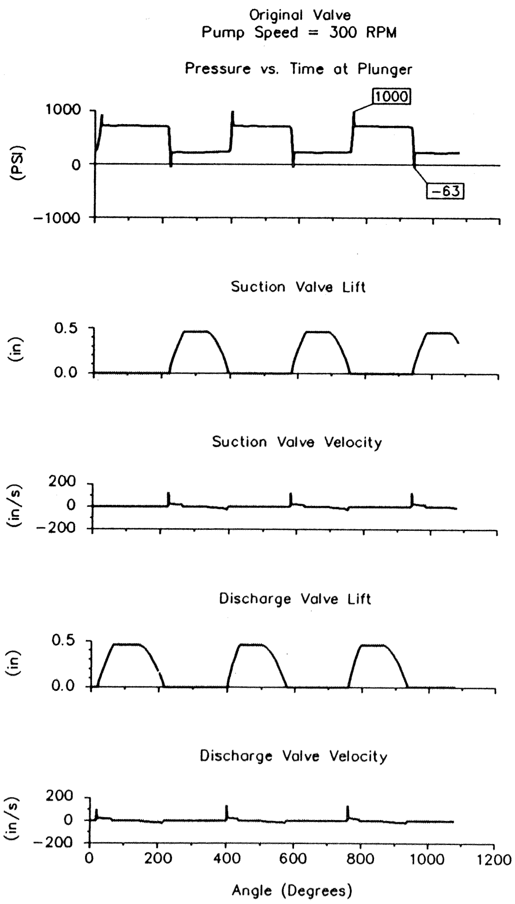


Figure 10. Computed Valve Dynamic Time Histories—Original Design.

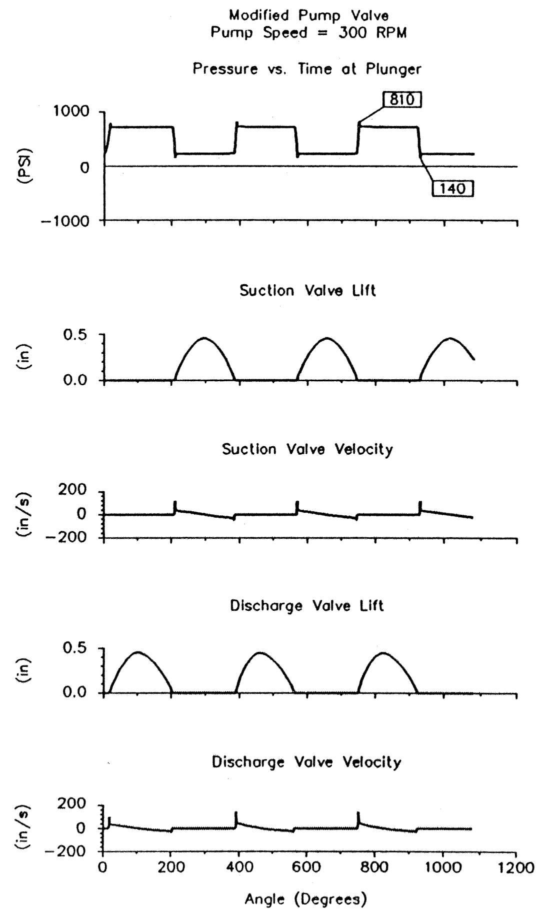


Figure 12. Computed Valve Dynamic Time Histories—Modified Design.

obtained with the other groove patterns, these designs were not tested, because there was concern that the remaining seat area was insufficient to provide adequate load bearing for the pressures experienced. Additionally, the analytical and field data showed that at some point, further reductions in seat area did not

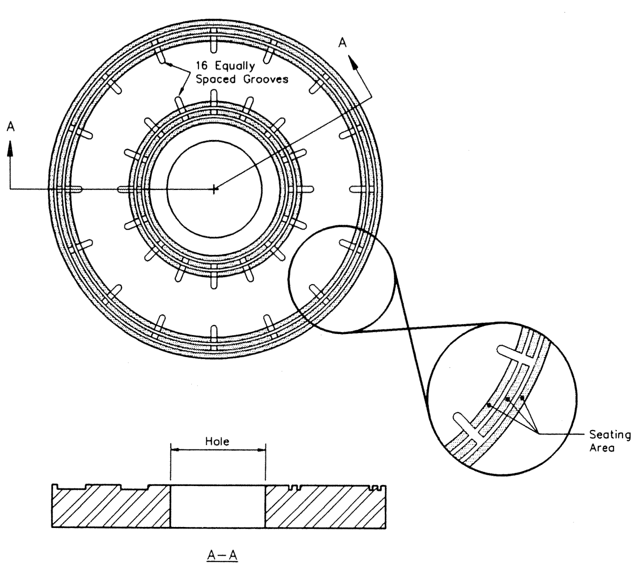


Figure 11. Double-Row Groove Pattern.

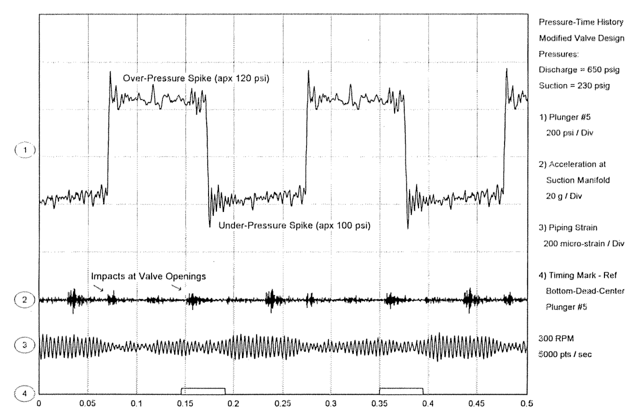
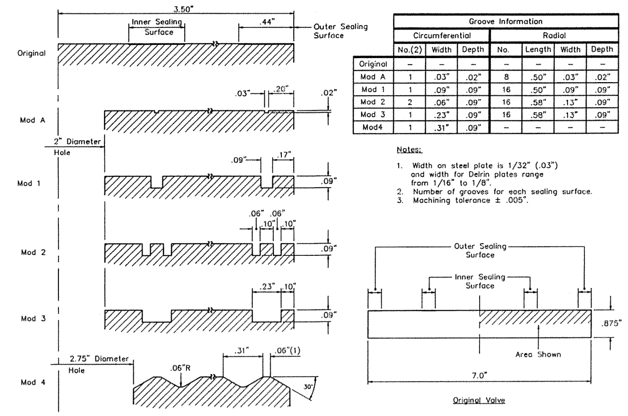


Figure 13. Over Pressure Spike, Impact Acceleration, and Piping Strain after Valve Modification.

cause further reductions in over pressure spike amplitudes, indicating that other effects were predominating the over pressure spike generation. Therefore, another type of valve would probably need to be considered to provide further improvements.

Operational and Design Parameters Affecting Sticktion

Other pumps were tested that had similar over/under pressure spike problems due to sticktion forces. As has been shown,



	Groove Information						
	Circumferential			Radial			
	No.(2)	Width	Depth	No.	Length	Width	Depth
Original	–	–	–	–	–	–	–
Mod A	1	.03"	.02"	8	.50"	.03"	.02"
Mod 1	1	.09"	.09"	16	.50"	.09"	.09"
Mod 2	2	.06"	.09"	16	.58"	.13"	.09"
Mod 3	1	.23"	.09"	16	.58"	.13"	.09"
Mod4	1	.31"	.09"	–	–	–	–

Figure 14. Other Valve Disc/Seat Groove Patterns.

cutting grooves into the valve disc or seat will not always eliminate the spikes. Several tests were conducted which showed that pump operational parameters and valve design parameters other than seat width could also affect the sticktion forces.

Valve Disc Material

A pump with flat, dual-ported steel valve discs was experiencing over pressure spikes of approximately 50 percent of discharge pressure. It was decided by the user to change valve disc material (for reasons unrelated to the over pressure spike) to PEEK plastic. No changes were made in the valve disc or seat designs. With the PEEK valve discs, over pressure spikes were reduced to 16 per cent of developed pressure.

Cutting grooves into the valve disc further reduced over pressure spike amplitudes to 11 percent, which was not as dramatic a reduction as had been previously experienced at some other installations.

Computer analysis showed that the reduction in mass of the PEEK valve disc could cause a reduction in over pressure spikes, but by an amount smaller than what was actually experienced. One possibility for the discrepancy is that the valve disc was probably not in complete contact with the seat, due to the increased flexibility of the PEEK material (Figure 15). The imperfect contact would have effectively reduced the sealing area, which in turn reduced the sticktion effect. This phenomenon would also account for the grooves being less effective.

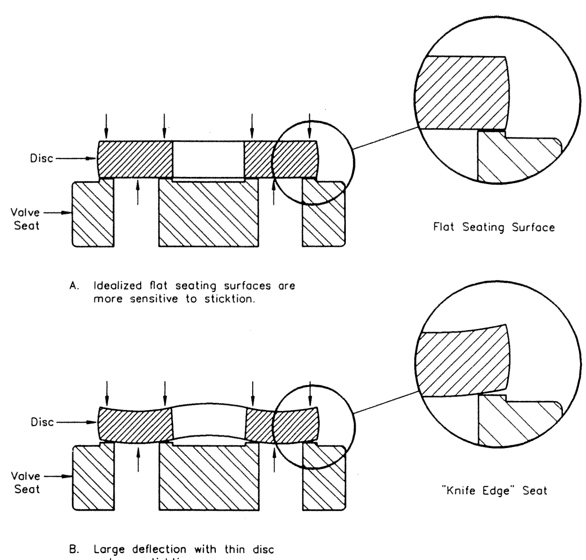


Figure 15. Imperfect Valve Seating Due to Disc Flexibility.

Pump Speed

Theory predicts that the valve disc velocity also affects the magnitude of the sticktion force. Since velocity of the disc is directly related to pump speed, the sticktion forces and resulting over/under pressure spikes should be higher. When data were acquired on variable speed pumps, this was indeed what was found. For example, the over pressure spike amplitudes were increased from 20 percent to 45 percent of discharge pressure when the pump speed was increased from 150 to 300 rpm.

Fluid Differences

It was found that virtually identical pump valve designs operating with similar pressures and speeds, but different fluids, could have dramatically different sticktion characteristics. Field data has shown that sticktion forces are greater for fluids having higher viscosity and molecular cohesion. (Molecular cohesion is the phenomenon that gives rise to viscosity. For Newtonian liquids, viscosity is essentially the resistance to shear caused by a velocity gradient. The amount of resistance depends both on the molecular cohesion and the shear. Without the presence of a velocity gradient, the molecular cohesion alone results in a "stickiness" [2].) For instance, a pump that operated with water experienced much less over pressure than a similar design operating with amine (a substance somewhat like automotive antifreeze).

CONCLUSIONS

Extensive field testing of reciprocating pumps that have experienced failures in the working barrels, valve and piping have shown that the valve behavior strongly influences the vibration and failures. Over pressure spikes have previously been attributed to area ratio, valve disc mass, and preload. Perhaps a more important factor is sticktion (Bernoulli effect), which is primarily related to seat area. Fluid effects such as viscosity and molecular cohesion, pump speed, valve disc and seat surface finish, and valve disc stiffness also affect sticktion, but to a lesser degree.

Sticktion delays the valve opening which results in over pressure spikes, valve disc impacts at valve opening, and local damage (cavitation pits) to valve seats. High amplitude impact noises are often an identifying characteristic of over pressure spike problems. These over pressure spikes can also cause excessive loads to be transmitted to pump components, which can result in drive train component failures and working barrel failures. In some cases, the high frequency mechanical impact energy was shown to be structurally transmitted throughout the pump and piping system exciting structural resonances that ultimately caused piping failures.

In the design phase, it is useful to perform detailed valve dynamic analyses to assist with valve geometry selection. When problems occur, use of instrumentation and data analysis hardware/software will lead to an understanding of pump problems. This information can be used alone, or combined with valve dynamical analyses to produce solutions to the pump problems. The groove pattern on the valve discs and seats has been shown to be effective in reducing the sticktion effects and associated problems.

REFERENCES

1. Streeter, V. L., and Wylie, E. B., *Fluid Mechanics*, Seventh Edition, New York, New York: McGraw-Hill Book Company (1979).

2. Bauer, F., "The Influence of Liquids on Compressor Valves," International Compressor Engineering Conference at Purdue University, West Lafayette, Indiana (1990).

ACKNOWLEDGEMENTS

The authors wish to thank the staff of EDI and especially Mark Broom for his assistance in producing many of the figures.