RELIABILITY AND FUTURE DEVELOPMENT OF HIGH PRESSURE DIAPHRAGM PUMPS FOR PROCESS SERVICE

by

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Gerhard Vetter, born in 1933 at Stuttgart, FRG, obtained his Dipl.-Ing.-degree in Mechanical Engineering at Techniche Universität Karlsruhe, FRG. After some years as a Research Engineer in turbomachinery and thermal power engineering at the same university, he joined Lewa Herbert Ott GmbH and Company, as head of the Research and Development Department. He became Chief Engineer and, in 1970, Technical Managing Director. In 1981, he accepted the call to a chair (professorship) for Apparatus and Chemical Machinery at Universität Erlangen-Nurnberg, FRG, still remaining Lewa's R & D Consultant.

Dr. Vetter has dedicated more than 25 years to research, development and design of pumps and metering equipment. He has been one of the pioneers in diaphragm pumps development, especially their application for dangerous and abrasive liquids, high pressures, large power, safety systems, metering accuracy and reliability. Many papers, patents and contributions to text books—some dealing with basics like cavitation, fatigue, pulsation, vibrations, and metering accuracy—have established his reputation as a pump specialist. The well-equipped laboratory at Universität Erlangen performs research work on pump and metering subjects such as tribological problems with pumps, kinematics of valve motion at oscillating displacement pumps, numerical computation of pressure vibration, stress and fatigue of high pressure components and diaphragms, and metering of bulk solids.

ABSTRACT

The aim of pump development in recent years has been mainly reliability, safety, low maintenance, and the avoidance of dangerous leakages. For years, more and more rigid regulations against pollution and for better safety have been issued. Many reaction components have been classified toxic or dangerous, with the consequence of a great trend towards leak free systems. For oscillating displacement pumps, the plunger sealing is difficult due to high pressure, pulsating forces and sliding speed. If ever one finds a satisfactory solution for a difficult plunger seal, there remain the disadvantages of unavoidable leakage, necessarily skilled maintenance and low lifetime.

The obvious separation of the pump's liquid end from the drive mechanism by diaphragms has been realized for years in small metering pumps. The development climbed to increasing power of about 10 kW, so that actually, there is a dominance of the diaphragm towards piston pumps.

Recently, the development of very large diaphragm pumps for process service, with power up to 500 kW, shows very reliable performances. Together with the total leak-free design the remarkable increase of reliability compared with conventional multiplex plunger pumps is very important. Many difficult fluids, such as abrasive suspensions, nonlubricating solvents or liquids gases can be pumped troublefree at pressures up to 1000bar. For process service, the application of resistant fluor plastomer or metal diaphragms is a must. The design, calculation, sealing, clamping, fatigue, lifetime, and especially the future development, which is expected to reduce the dimensions, are explained.

Long term applications have shown that diaphragm pumps actually perform to trouble- and maintenance-free service of many months. The most important step towards total safety was the development of sandwich diaphragm rupture monitoring systems.

The hydraulic system and the alternative designs are compared. This has developed towards automatic position-control of the diaphragm. Safety, venting and snifting valves and their control are very important.

Some remarks deal with fatigue and wear problems at high pressure of components like thick walled parts and valves, and with design of the drive units and their control.

An important condition for reliable service of high pressure diaphragm pumps is the restriction of pressure vibrations in the piping system and avoiding any cavitation. There is a significant influence of fluid compressibility on displacement characteristics and stimulation of pipe vibrations. A computation method including dampers is outlined.

INTRODUCTION

Application

In processing and production, there is a need for pumping dangerous, toxic or abrasive fluids against high pressure. As environmental protection and worker safety more heavily influence plant design, there is a trend towards the application of liquefier pumps. The aim is to avoid dynamic fluid sealing at shafts and pistons by means of canned-motor or permanent-magnet centrifugal and oscillating diaphragm-, hose- or bellow-displacement pumps. Thus, the fluid-wetted part (working chamber) of the pump is totally separated from the drive-mechanism and surroundings. In addition to being leakfree, in most cases, remarkable improvements are achieved in maintenance, safety, lifetime and realiability.

Typical applications for diaphragm positive displacement pumps are known from high pressure processes (Table 1), be it for reaction, separation or transport, but there are also many other applications at lower pressures.

In general, "sealless" designed pumps are recommended for the following cases:

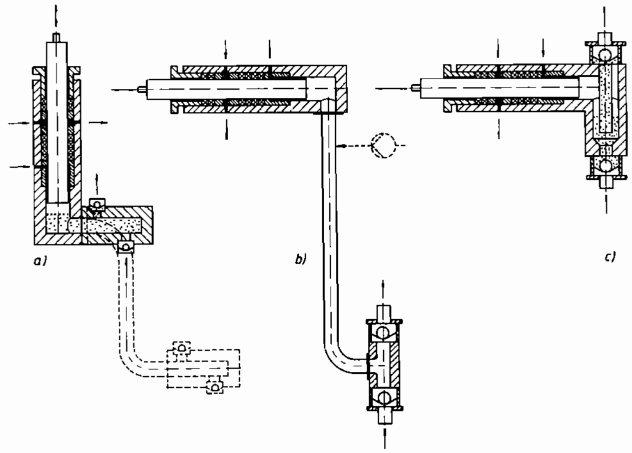


Figure 1. Plunger Pump Design for Abrasive Slurries—a) vertical, b) hydraulic link, c) sedimentation chamber (remote head).

- fluid toxic, explosive, dangerous, radioactive, corrosive, odorous (e.g., chlorine, ammonia, phosgene, plutonium-solution, hydrocarbons, aluminium-alcyls, acids)
- fluid low viscosity (non-lubricating) and with elevated vapor pressure (e.g., liquified gases, hydrocarbons, solvents)
- fluid-stratifying, settling out, abrasive (e.g., resins, potassium-amid/ammonia, aluminium-alcyls, catalyst or ceramic suspensions and other slurries)
- fluid sensitive to humidity and air (e.g., titaniumtrichloride)
- fluid-must remain sterile (food-components, antibiotics).

Much effort in the past has been applied to the development of plunger seals and this continues today. New hard and wear resistent materials and coatings for plungers and bushings, and the application of new seal materials, shapes and designs have demonstrated progress. For abrasive slurries, e.g., coal slurries for hydrogenation plants, the arrangement of vertical plunger, remote head or settiing chamber to keep particles away from the seal have proven effective (Figure 1).

Plunger seals with safe and reliable flushing and pressurized restriction against fluid leakage may be called "leakfree," but they are not "sealless" and need regular maintenance and adjustment in order to achieve safe operation. However, in general maintenance of plunger seals is expensive and many fluids must not be allowed to escape from the fluid circuit, due to legal regulations.

Current Design Survey and Historical Retrospective

Leakfree oscillating diaphragm pumps initially were only used for very critical applications, e.g., for liquid metering in nuclear chemistry [3]. The design was characterized by hydraulically operated stainless steel-diaphragms (Table 2, line 3), very small capacity and stroke length-controlled drive units, as are typical for metering pumps.

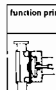
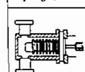
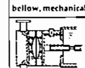
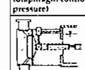
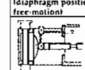
At the same time, mechanically activated diaphragm pumps with elastomer diaphragm (Table 2, line 1) for water treatment chemical metering, and bellow-type metering pumps especially for glass-plants (Table 2, line 2), both limited to low pressure, met and satisfled an urgent need. Actually, many thousands of such leakfree metering pumps, a large number with solenoid drive units, are being installed every year.

The worldwide development of chemical processing resulted in the development of positive displacement hydraulic dia-

Table 1. Typical High Pressure Processes for Reaction, Separation and Transore.

POLIOLEFINES	LDPE	<	3 500 bar
	LLDPE (Ziegler)	<	1 600 bar
	HDPE	<	200 bar
ACETIC ACID		<	700 bar
METHANOL, AMMONIA, UREA, OXO-SYNTHESIS, BUTANOL HYDROGENATION: COAL, HEAVY OILS HYDRO + STEAM-CRACKING HEAVY WATER		<	300 bar
FISCHER-TROPSCH-SYNTH. AL-ALCYLS (Ziegler) ALCOHOLS (ALFOL) GAZ-CLEANING		<	200 bar
LEACHING OF ORES COAL-GAZIFICATION		<	150 bar
SUPERCRIT. EXTRACTION		<	300 bar
REVERSE OSMOSIS		<	1 000 bar
SPRAY-DRYING		<	300 bar
HOMOGENIZATION MIXING, EXTRUSION CELL-CRACKING TOBACCO-IMPREGNATION		<	500 bar
NATURAL-OIL + GAZ-EXPLORATION		<	400 bar
POLYMER-SPINNING and FILTRATION		<	500 bar
SLURRY-TRANSPORT		<	200 bar

Table 2. Survey and Characteristic Parameters of Diaphragm Pump Designs—$\dot{V}$) volume flow, Δp) Discharge pressure, p) power.

function principle	drive mechanism	Limitations $\dot{V}$	Limitations Δp	Limitations p	diaphragm-material	application	
 diaphragm, mechanical (DM)	solenoid linear drive spring action	< 30 l/h	< 25 bar	-	elastomers	metering	low invest, maintenance, leakfree
	cam drive (stroke adjustment)	< 500 l/h	< 10 bar	-	PTFE/elastomer-compound		
	crank drive	< 10 m³/h	< 5 bar	< 3 kW	elastomers	conveying	sanitary
		< 100 m³/h	< 2 bar	< 5 kW			dirty liquids
 bellow, mechanical (BM)	crank drive (stroke adjustment)	< 5 m³/h	< 10 bar	-	PTFE	metering	glas/PTFE-installations
 diaphragm, hydraulical (DHn) (diaphragm control by negalive pressure)	spring action cam drive (stroke adjustment)	< 10 l/h	< 700 bar	-	metals	metering	precise, leakfree low maintenance
	crank drive (stroke/frequency-control)	-	< 3500 bar	-	metals	metering/ conveying	leakfree low maintenance
		-	< 350 bar	< 200 kW	PTFE		
 diaphragm, hydraulical (DHp) (diaphragm position control, free-motion)	crank drive (stroke/frequency-control)	-	< 500 bar	< 200 kW	PTFE (elastomers)	metering, conveying	leakfree low maintenance slurries
	crank drive (triplex, quadruplex, frequency-control)	-	< 500 bar	< 500 kW	PTFE	conveying	
			< 200 bar	< 1500 kW	elastomers	feeding	
 hose, hydraulical (HHp) (position controlled diaphragm)	crank drive (duplex, triplex, quadruplex, frequency-control)	-	< 100 bar	< 200 kW	elastomers	conveying feeding	leakfree low maintenance slurries

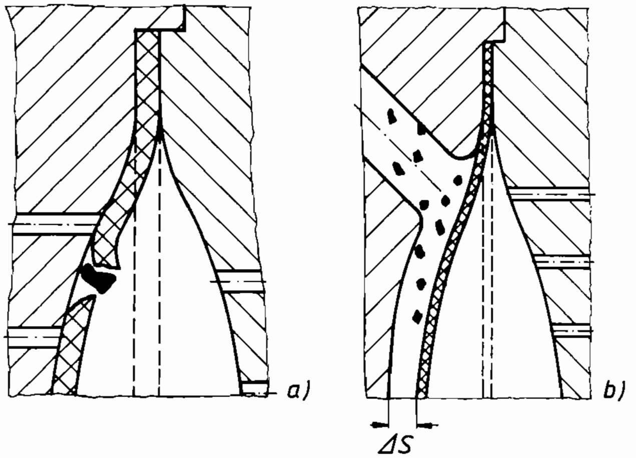


Figure 2. Perforation of Diaphragms by Particles—a) diaphragm motion controlled by limiting curve of cover plate, b) free moving diaphragm by position control, changes safe distance.

phragm pumps, especially for metering applications, mainly with corrosion resistant PTFE-diaphragms, which are resistant to attack, and allow reasonable deflection and long lifetime [4, 5, 6, 7, 8, 9, 10]. The breakthrough was achieved by the introduction of hydraulic control systems which keep the diaphragm away from the working-chamber walls (Table 2, line 4) on the pumped fluid-side, and a reliable sandwich diaphragm rupture monitoring system [11, 12]. The systems will be explained in more detail later, but what happens when particles, dirt or

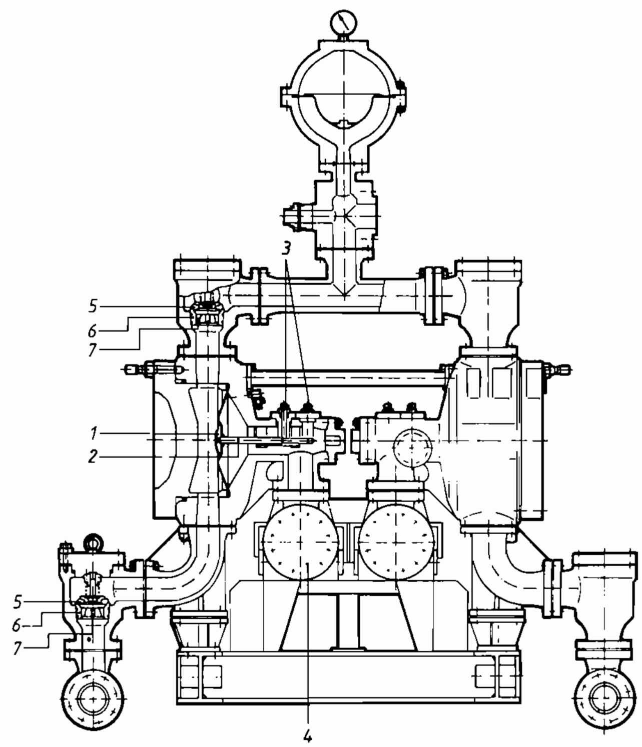


Figure 3. Large Slurry Diaphragm Pump (Four Cylinder Double Acting). 1) diaphragm, 2) rod, 3) position sensors, 4) hydraulic control, 5) — 7) pump valves.

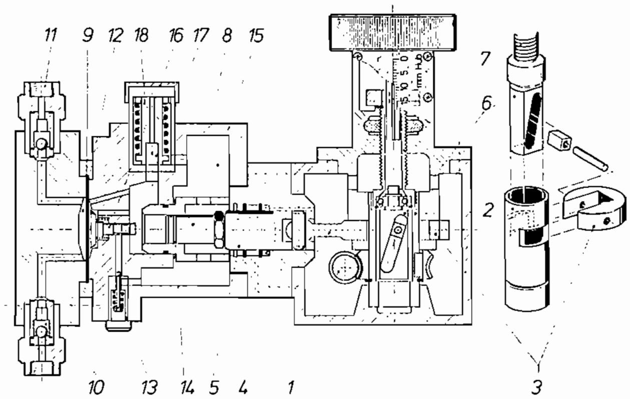


Figure 4. Compact Diaphragm Metering Pump—1) — 7) eccentric stroke adjustable drive unit, 8) — 14) plunger seal, 9) diaphragm, 10) — 11) pump valves, 12), 13), 15) — 18) hydraulic system.

pumped fluid contents, are clamped between diaphragm and wall is dramatically shown in Figure 2.

If we look at the more recent history of the last 10-15 years in the development of diaphragm pumps, we recognize two main courses, which are illustrated in Figure 3 and 4.

- *For the pumping of mainly aqueous slurries* in filtration, pipeline transport and other similar slurry processes (e.g., bauxite and ore-treatment), hydraulic diaphragm pumps up to very large capacities (more than 1000 kW) with rubber-like elastomer diaphragms have been successfully installed. The elastomer-diaphragms, suitable only for low corrosion, moderate pressures (100-150 bar) and limited temperature, allow remarkable deflections (Table 3). Due to large diaphragm-head dimensions and special slurry requirements, the four-cylinder double-acting arrangement is mainly used (Figure 3). Important design criteria is the easy accessability of valves and diaphragms.
- *For hydraulic PTFE-diaphragm pumps*, mainly suitable for process application with various liquids, there was a progressive, step-by-step increase in applicable pressure starting with around 50 bar and ending currently at more than 500 bar (Figure 4).

Since traditionally these pumps have been configured from metering pumps, the drive units are usually stroke controlled

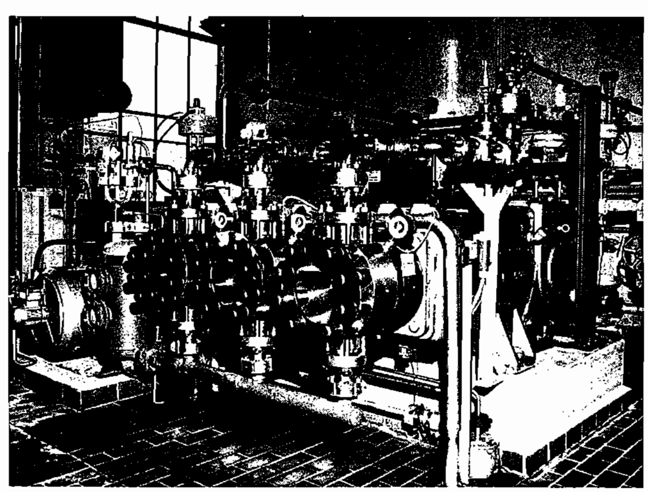


Figure 5. Triplex Diaphragm Pump (320 Bar, 160 kW) for a Dangerous Fluid with Speed Control in a Chemical Plant.

up to powers for triplex-arrangement of around 100 kW. For increasing power demand up to 500 kW, stroke controlled crank drive-units are rather complicated and expensive. Therefore, triplex-crank drive-units with diaphragm heads (Figure 5) including electric speed variation by DC- or frequency-controlled, three-phase AC-motors have been developed, which are, at the same time, a reasonable substitute for triplex plunger pumps.

HYDRAULIC PROCESS DIAPHRAGM PUMPS

Fundamentals

Mass Flow, Volumetric Efficiency

The massflow of a diaphragm pump can be calculated by

$$\dot{m} = \rho \cdot A_K \cdot h_K \cdot n \cdot \eta_E \cdot \eta_C \quad (1)$$

$V_h = A_k \cdot h_k$, representing the stroke volume and, ρ, the fluid's density. The elasticity factor η_E regards elasticity effects of the fluids and working chamber. η_C represents the internal and external leakages through seals and valves. The product $\eta_E \times \eta_C$ is called volumetric efflciency:

$$\eta_V = \eta_E \cdot \eta_C \quad (2)$$

For sroke adjustable stroke units with a displacement characterized by constant mid-position the elasticity factor η_E comes [8] to:

$$\eta_E = 1 - \left\{ \varepsilon_{TF} \cdot \kappa_F + (\varepsilon_{TH} + \frac{1}{2})\kappa_H + \lambda \right\} \cdot \frac{\Delta p}{h_k/h_{k100}} - (\kappa_F - \frac{\kappa_H}{2})\Delta p \quad (3)$$

ε_{TF} and ε_{TH} represent the relative dead spaces on the liquid and hydraulic side

$$\varepsilon = \frac{V_T}{V_h} \quad (4)$$

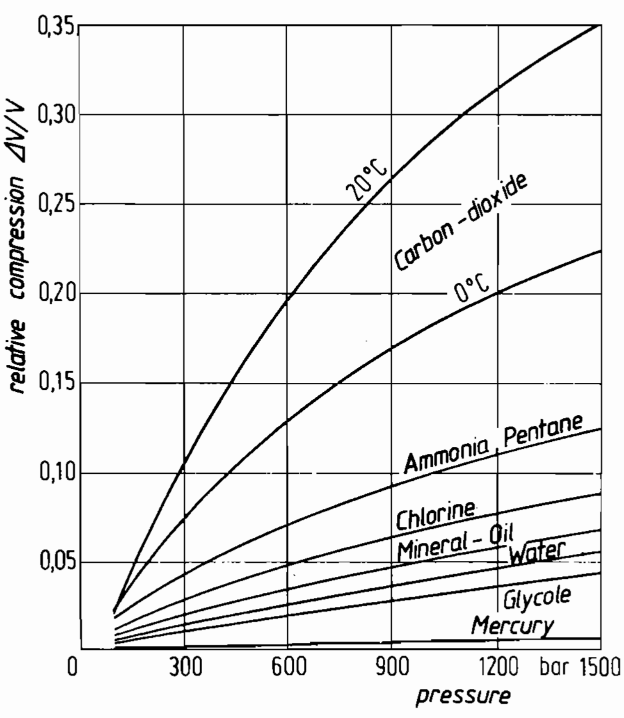


Figure 6. Compressibility of Various Fluids.

The fluid elasticities κ_F and κ_H are connected with usual compressibility data (Figure 6) by

$$\kappa = \frac{\Delta V}{V} \cdot \frac{1}{\rho} \quad (5)$$

In Equation (2), the elasticity of the working chamber is represented by λ assuming a linear correlation. Further, Equation (3) reveals that the pressure difference Δp and the stroke setting ratio are of direct influence on η_E. The elasticity factor is approaching $\eta_E \rightarrow 1$ if:

- $\Delta p \rightarrow 0$
- $\varepsilon_{TF}, \varepsilon_{TH} \rightarrow 0$
- $\kappa_F, \kappa_H \rightarrow 0$
- $h_k/h_{100} \rightarrow 1.0$
- $\lambda \rightarrow 0$

From these relationships the main design aims are determined:

- narrow dead spaces
- low compressibility of hydraulic fluid (e.g., glycols instead of mineral oils)
- rigid working chamber.

For many liquids, especially the "liquid gases," it is more advisable to evaluate the isentropic change in state by a TS-diagram (Figure 7).

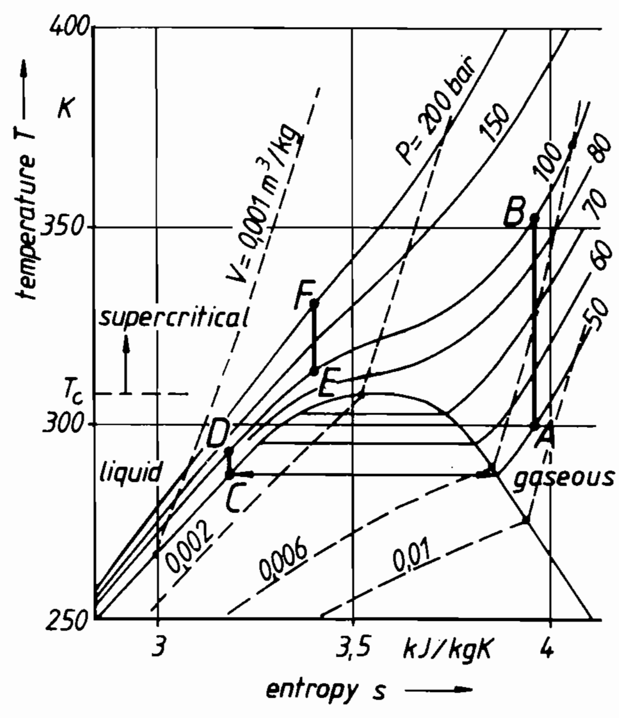
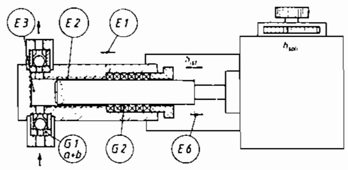
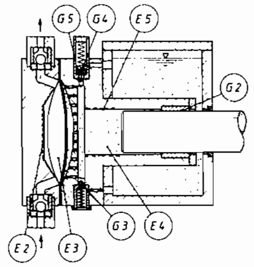


Figure 7. Isentropic Change in State (CO_2)—A-B) gas compression, C-D) liquid compression, E-F) supercritical fluid, T_c) critical temperature, p) pressure, v) specific volume, s) entropy, t) temperature.

If, at high discharge pressures, the compressibility for a liquid κ_F is extraordinary, a reduction of ϵ_{TF} should be especially emphasized.

Regarding Equation (2), the tightness factor η_C is correcting for leakage influences through pump check valves and plunger





	Parameter	design action
E.1	plunger seal	rigid, small volume
E.2	working chamber (fluid)	rigid, small dead space
E.3	fluid	small dead space
E.4	hydraulic fluid	small dead sparce, low compressibility
E.5	working chamber (hydraulic fluid)	rigid, small dead space
E.6	elastic deformation	rigid, no play
G.1a	valve leakage	no wear, tight
G.1b	valve closing delay	strocke frequency, spring loaded
G.2	plunger seal	tight, no wear
G.3	snifting valve	tight
G.4	relief valve	tight
G.5	venting valve	volumetric principle

Figure 8. Regular Parameters Influencing Mass Flow.

seals on the hydraulic side. Without referring to more details *Pump Valves* it should be pointed out that with non-delayed closing pump valves and tight plunger seal, the tightness factor should be better than $\eta_G > 0.98$, including the hydraulic side venting system.

The different influences on η_E and η_G are summarized in Figure 8.

A typical volumetric balance for a 350 bar diaphragm pump of the 150 kW level with water as pumped fluid:

tightness factor	η_G =	0.98
venting valve		0.005
pump valves		0.01
plunger seal		0.005
elasticity factor	η_E =	0.89
fluids		0.095
elasticity of working chamber		0.015
total volumetric efficiency	η_V =	

For other fluids, η_E would change: water 0.87, ammonia 0.83, CO_2 0.71. Normally, the tightness factor—regarding pump valve tightness—changes with decreasing fluid viscosity v as the leakflow V_L through an annular clearance would be approximately calculated by (clearance dimension s).

$$\dot{V}_L = K \cdot \frac{s^3 \cdot \Delta\rho}{v} \tag{6}$$

When considering leakflows through clearances at high pressures, the remarkable increase of fluid viscosity with the pressure should be noted.

Pulsations

Compared with plunger pumps, the volumetric effIciency of diaphragm pumps is signiflcantiy lower, due to the necessary larger dead spaces. The actual displacement of the fluid (Figure 9) for any oscillating displacement pump is delayed by the compression period. The indicator-diagram [Figure 9(a; 1-2-3-4); 9(b) and 9(c; 1-2, 2′)] is good information about the working period, and volumetric efficiency can easily be evaluated from that diagram. The special shape of the actual discharge diagram (e.g., 1, 2′, 2, 3, 1) involves a shock situation at 2, 2′, in addition to the basic pulsation.

Superimposition of several cylinders at different volumetric efficiencies η_V shows the growing shock-contents of the pulsation input of the pump (Figure 10). Referring to η_V data in the example above mass flow, one can determine the displacement characteristics at $\eta_V = 0.7$ respectively, 0.9. Pulsations between $\eta_V = 0.9$ and 1.0 would more likely represent plunger pump behavior.

The installation of high pressure diaphragm pumps needs careful pipeline layout and pressure pulsation computation [13] [14]. There is a difference in pulsation excitation between stroke or speed control of the pump (Figure 11).

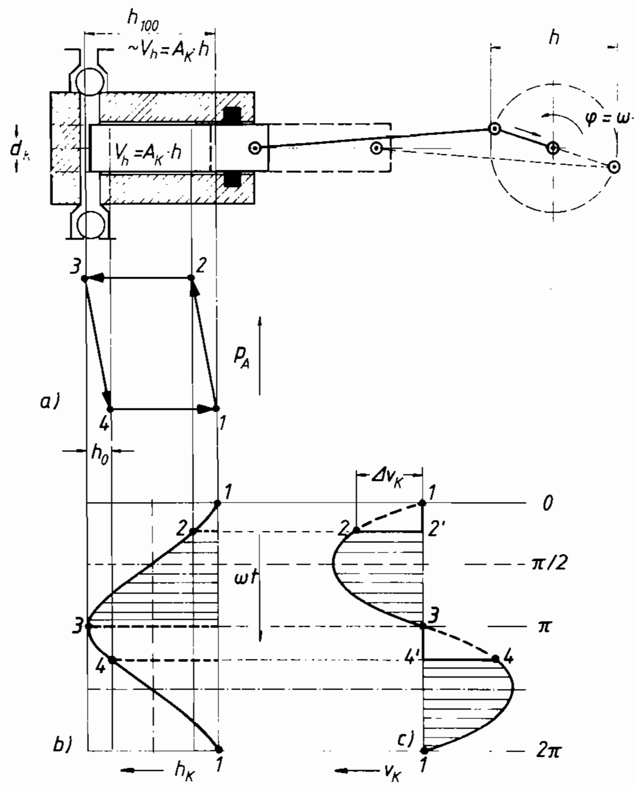


Figure 9. Fluid Compression During Displacement—h_k) plunger stroke, v_k) plunger velocity, vk) plunger velocity, ω.t) crank angle φ.

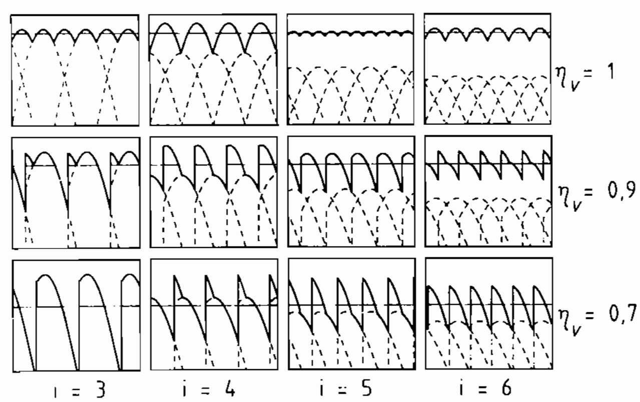


Figure 10. Pulsation of Volume Flow for Multiplex Oscillating Displacement Pumps—η_v volumetric efficiency, i) number of cylinders.

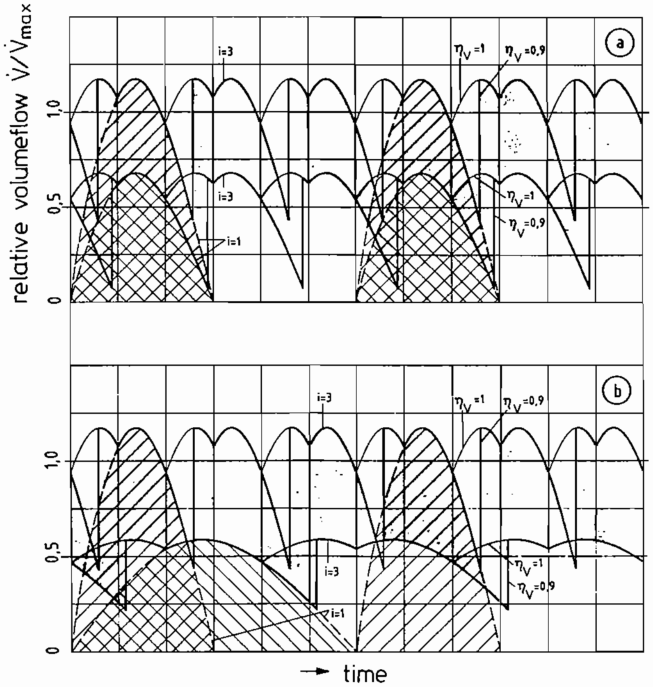


Figure 11. Pulsation of a triplex pump ($\eta_v = 0.9$)—above) stroke control, below) speed control.

Characteristics

As is usual for all positive displacement pumps [15], the massflow has a linear relationship, per Equation (1), based on the control parameter, stroke length or frequency (Figure 12). The elasticity factor η_E is dependent on stroke setting ratio, h_k/h_{k100}, so that at h_{k0}/h_{k100}, there exists a stroke setting dependent on discharge pressure Δp where m = 0. For speed control, the $\dot{m}$ = f (n/n_{100}) correlation is approximately linear and proportional.

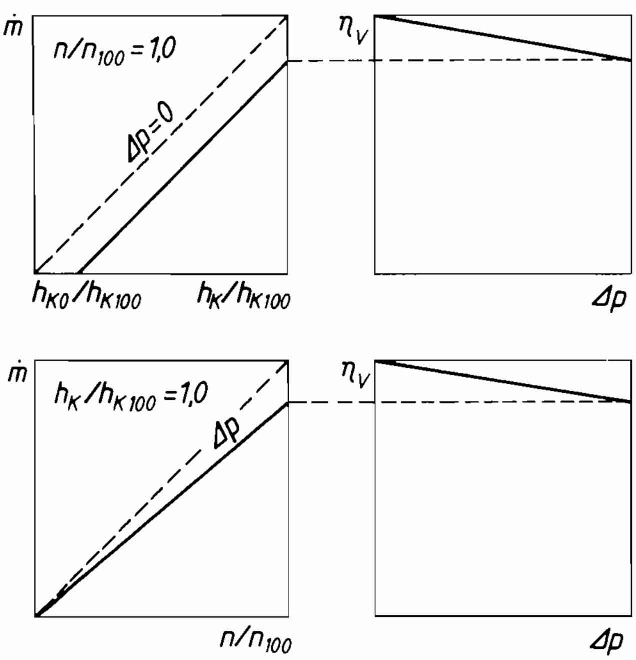


Figure 12. Characteristics of Stroke and Speed-Controlled Oscillation Displacement Pumps. $\dot{m}$) mass flow, Δp) discharge pressure, h_K/h_K100) stroke setting, n/n_{100}) speed setting, η_v) volumetric efficiency.

It is evident that metering accuracy increases if the stroke adjustment approaches the limitation h_{k0}/h_{k100}[15].

The regulation range is limited (e.g., 1:10) for speed controls mainly by the available torque and the motor cooling systems (AC frequency controlled or DC motors). The average diaphragm pump power requirement is:

$$P = \dot{V} \cdot \Delta p \tag{7}$$

where the average volume flow V is:

$$\dot{V} = \int \dot{V}(t)\, dt \tag{8}$$

If the temporary volume flow V(t) (Figure 11) is pulsating, torque is pulsating also (ω angular velocity)

$$P(t) = T(t)\ \omega \tag{9}$$

Driving motors and couplings have to be designed for pulsating load.

The breakaway torque (when starting a pump with full discharge pressure) may equal double normal torque. It is recommended that a special startup arrangement for large pumps be utilized.

The fluid side startup arrangement is well known for plunger pumps, and may also be used in conjunction with a bypass control (Figure 13). The diaphragm pump makes a hydraulic side startup arrangement feasible (Figure 14). Before starting the pump, the sliding valve (1) is in the upper position by the spring load. During the startup period, the increasing pressure through check valve (6) shifts the piston to the closed position.

Fatigue of Pressurized Parts

As the pressure in the working chamber pulsates to the full amplitude of the discharge pressure, all pressurized parts are endangered by fatigue. The stress situation for single stage diaphragm pumps is usually more severe than for multistage piston compressors (Figure 15).

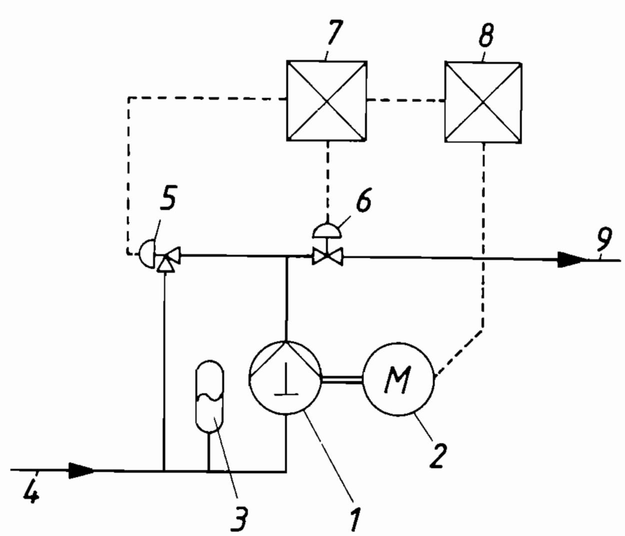


Figure 13. Startup through By-pass [7]—1) pump, 2) motor, 3) pulsation damper, 4) suction, 5) — 6) control valve, 7) — 8) control system, 9) discharge.

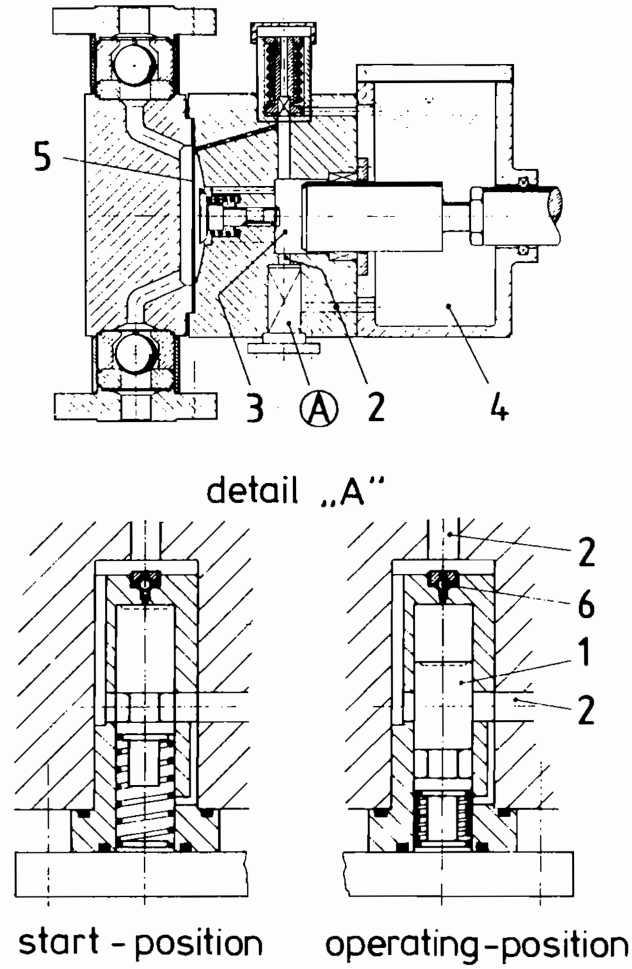


Figure 14. Hydraulic Startup System of a Diaphragm Pump [7].

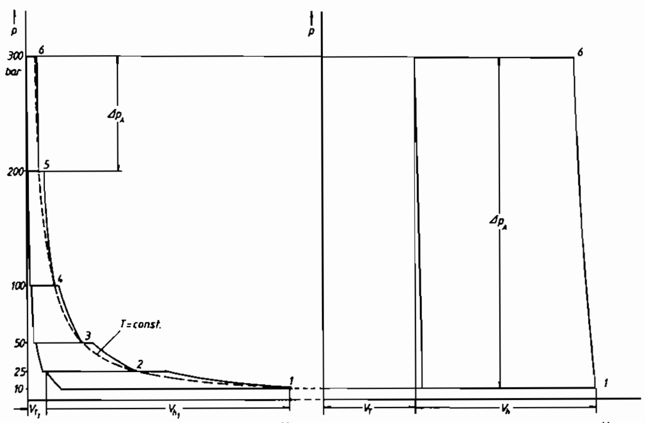


Figure 15. Pressure Amplitude for Multistage Gas Compressors (left) and Single Stage Diaphragm Pumps.

Many fatigue problems can be solved by proper design, stress analysis, mirror like surface finish and well rounded notch free geometry. It is recommended that comparable fatigue tests with similar specimens under identical conditions (e.g., fluids) be performed. An example is shown in Figure 16 of the result of a thickwalled circular hydraulic cylinder under pulsating pressure [16] yielding the maximum applicable pressure amplitude $130p_{max}$ Since most applications require more or less corrosive liquids to be pumped, the use of chrome-nickel steels of the duplex or soft martensitic type, is recommended for their high strength and good toughness.

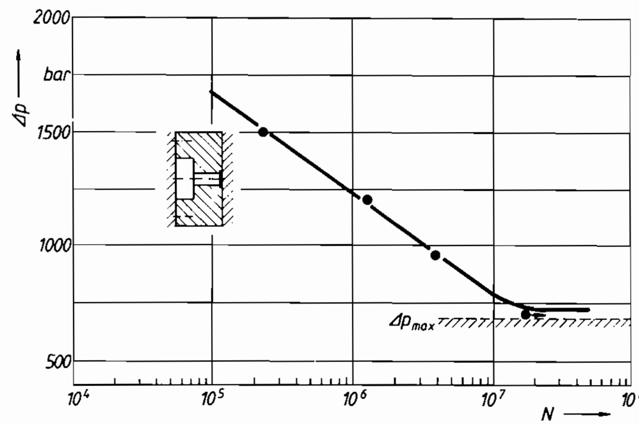


Figure 16. Fatigue of a Thickwalled Hydraulic Cylinder Exposed to Pulsating Pressures—Δp) pressure amplitude, N) number of cycles.

DESIGN OF PROCESS DIAPHRAGM PUMPS

The following explanations represent a short survey of the actual design, the result of long term development and experience in plants.

Types of Drives

The economic limit for stroke controlled drive units today, mainly eccentric systems from different manufacturers (Figure 17) with manual, pneumatic, or electric stroke control, is a maximum of 50 kW per element (150 kW per triplex arrangement). As stroke control requires operation of the stroke and crank mechanism under full and pulsating load, it is obvious that

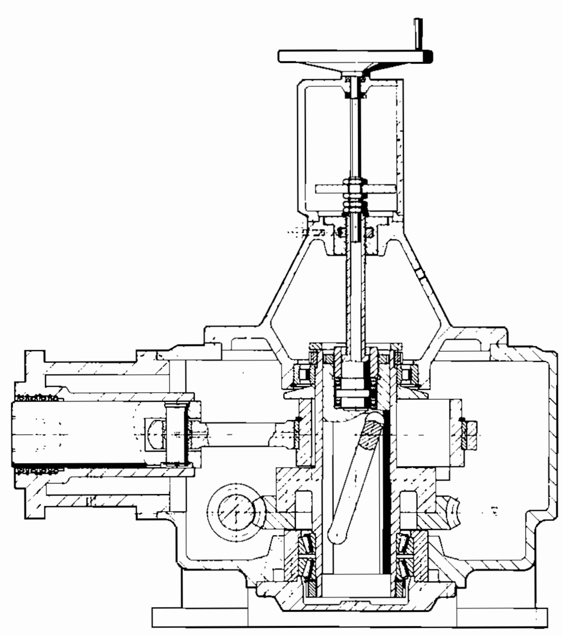


Figure 17. Stroke Controlled Eccentric Drive Unit.

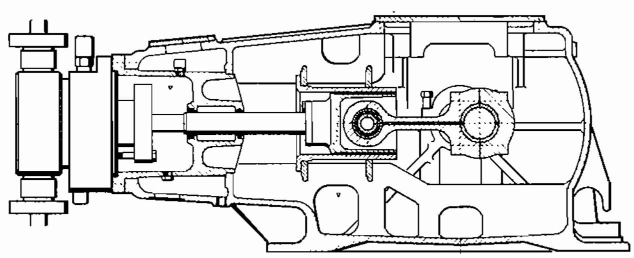


Figure 18. Cross-section of a Triplex Crank Drive Unit (power up to 500 kW, see Figure 5).

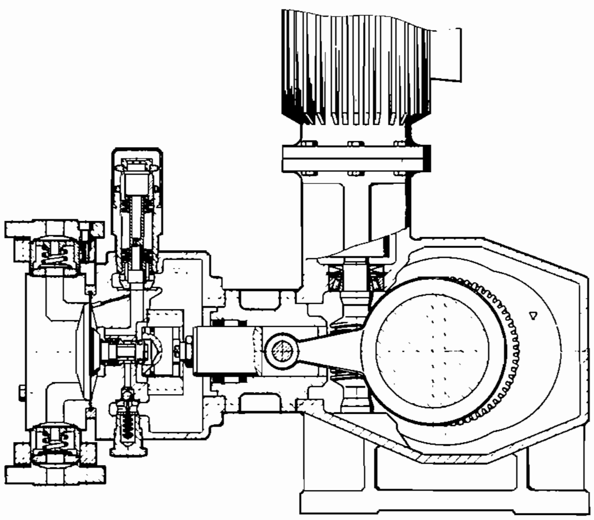


Figure 19. Cross-Section of a Smaller Triplex Diaphragm Pump.

triplex crank drive units (Figures 18 and 19) for the full power range (10-500 kW) are mechanically more reliable. As diaphragm heads are of relatively large diameter with respect to the distance between their centrer lines on multiplexed pumps, the largest drive units require intermediate bearings on the eccentric shaft.

Hydraulic System

The hydraulic system has to fullflll the following functions:

- diaphragm displacement by deflection
- control of the hydraulic fluid volume between plunger and diaphragm
- venting of air-bubbles
- safety or relief function
- plunger sealing

The gate-valve system (Figure 20) for example, works as follows: The plunger (1), sealed by lip-seals, piston rings or ground bushings, displaces hydraulic fluid and thus deflects the diaphragm (2), the deflection volume (stroke volume) being about 15 percent smaller than the fluid side working chamber volume. The diaphragm performs a plus/minus motion, and if the hydraulic flll decreases by leakage the diaphragm, touches and moves the balanced gate (sliding)-valve (3) rearward. In the rear gate valve position a connection between the hydraulic chamber (10), the snifting (refill) valve (4) and the reservoir (8) is established, permitting volumetric suction of hydraulic fluid replacement. The system's "intelligence" is based on the two necessary conditions for snifting:

- diaphragm at rear position, which indicates lack of proper hydraulic fllling
- negative pressure to open snifting valve

The diaphragm motion is, therefore, forced from rear to positive position, never touching the working chamber's upper cover-plate (Figure 2).

The hydraulic relief-function is attained by the spring-loaded valve (5), which contains a venting valve, operated by the pulsating hydraulic pressure.

There are several known venting systems. Some examples are shown in Figure 21. For large diaphragm pumps, bleed valves, in combination with a throttling system and check valve or double-seat ball valve, are utilized both showing a venting-flow dependent on discharge pressure. Some experience is also available with timer-controlled solenoid venting systems. For smaller diaphragm pumps, volumetric venting by piston valves is necessary, which are not dependent on discharge pressure.

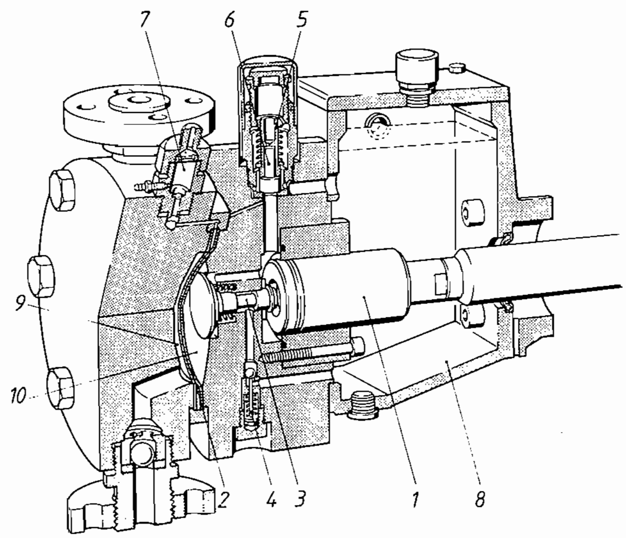


Figure 20. Diaphragm Position Control by Gate Valve-System (free moving diaphragm). 1) plunger, 2) diaphragm (sandwich), 3) gate valve, 4) snifting valve, 5) relief valve, 6) venting valve, 7) rupture monitoring system, 8) reservoir, 9) working chamber, 10) hydraulic chamber.

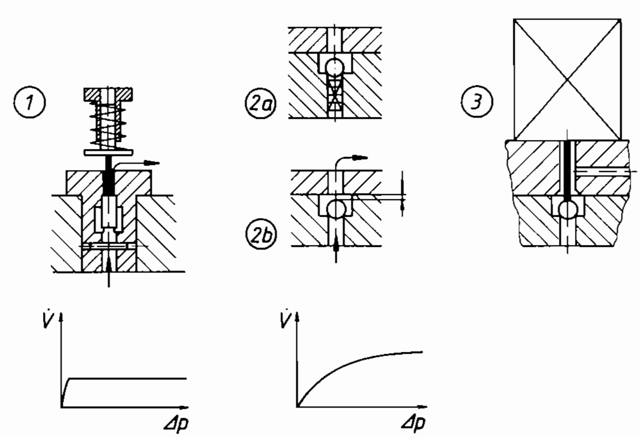


Figure 21. Venting Systems for Hydraulic Diaphragm Pumps. 1) volumetric piston venting valve, 2a) bleeding valve (throttling system, check valve), 2b) bleeding valve (double seat ball valve), 3) solenoid bleeding valve.

The continual venting of the hydraulic system by a hydraulic fluid stream of around one-half percent of the pump capacity is necessary as the formation of gas bubbles in unavoidable from localized pressure gradient or temperature increases (plunger seals). The general design should prevent any formation or sucking of air; an oil-flooded plunger rod (Figure 20, reservoir 8) is recommended. Furthermore, diaphragm pumps usually claim elevated suction conditions.

Quite a number of hydraulic systems similar to that in Figure 20 for diaphragm pumps are proven [18, 19, 20, 21]. It is a fact that a certain resistance to diaphragm perforation at the cover side exists due to unexpected long-term vacuum on the suction (strainer clogged, etc.), and the hydraulic components which control snifting according to diaphragm position may be not totally tight. Large diaphragm pumps should, therefore, be installed with sufflcient suction pressure (several bar), or a booster pump arrangement and monitoring system for minimum suction pressure. It is the state-of-the-art for large slurry diaphragm pumps (Figure 3), where clogging may occur frequentiy, to control the diaphragm position in both directions by sensors and hydraulic valves. For that purpose, the diaphragm moves an attached rod [22]. The future trend of development for process diaphragm pumps will make use of totally safe electronic position control of the diaphragms.

The plunger sealing with viscous hydraulic fluids (hydrocarbon oils, glycols) needs experience. Up to 300-500 bar, the application of piston rings (Figure 22) is reliable. For higher pressures, the application of ground bushings, specially coated with very narrow clearance, gives better results (Figure 23).

Diaphragms

The diaphragm design is mainly deflned by the material. Four groups should be considered:

• *Metallic materials.* They are necessary for certain fluid and temperature requirements. The application of cold-rolled metal sheet (austenitic steel, Ni- alloys, Titanium) with 0.2 to 1.0 mm thickness is proven.

• *Plastomers*: The majority of all process diaphragm pumps utilize PTFE, FEP or similar plastomers, which represent the standard for corrosion, thermal, and fatigue resistent material.

• *Elastomers*: Their remarkable elasticity allows greater thickness, which is very advantageous for the clamping. For applications in the process industry, a number of different elastomers is required (e.g., butyl and silicon rubber, fluor-elastomers, etc.) for chemical resistance. Most elastomers can be used for different shaped diaphragms such as hoses and profiled disc diaphragms.

• *Compound Materials.* The application of PTFE-coated elastomer diaphragms is proven only for rather low pressures and, therefore, not a current solution for process diaphragm pumps. The main problem is the different modulus of elasticity and the permanent tightness of the thin coating layer.

Some characteristics of frequentiy used diaphragm features are summurized in Table 3.

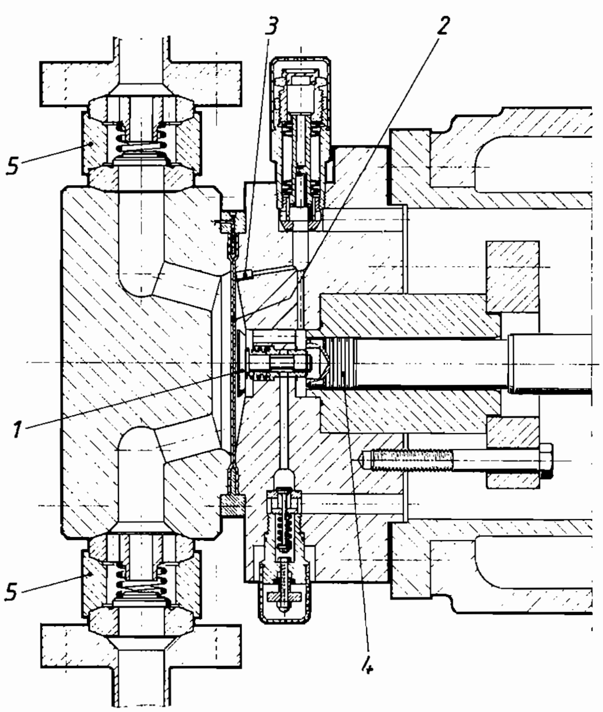


Figure 22. Diaphragm Head for a Large Triplex Process Pump (320 bar, see Figure 5) — 1) gate valve, 2) sandwich diaphragm, 3) venting bore, 4) piston ring seal, 5) pump valves (plate springloaded).

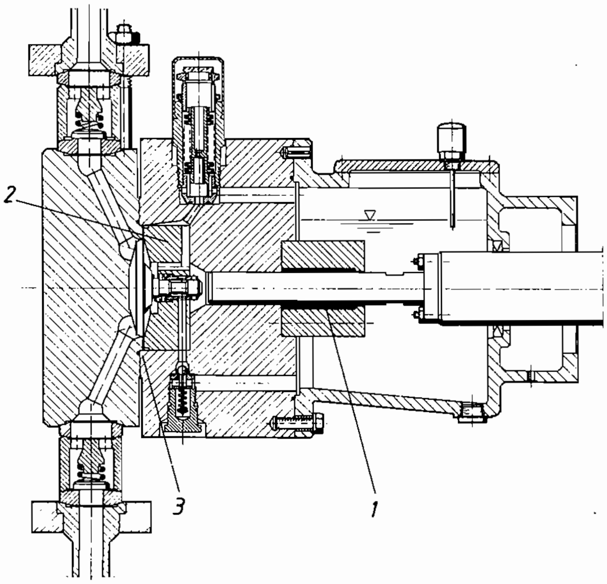
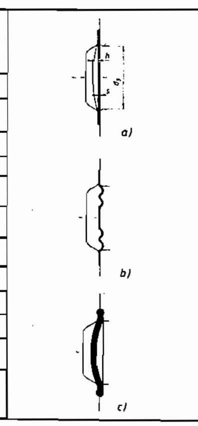


Figure 23. Diaphragm Head for Pressure up to 500 Bar. 1) ground bushing, 2) insert, 3) diaphragm pressure balance.

Table 3. Characteristics of Different Diaphragm Types.

	disc-shaped plain-parallel (a)	disc-shaped plain-parallel (a)	disc-shaped corrugated (b)	disc-shaped calotte-shape or undulated (c)
expences	low	low	high	moderate
manufacturing quality	good	very good	moderate	very good
support	favourable	favourable	unfavourable	favourable
pores	moderate	negligible	moderate	negligible
sandwich-diaphragm	yes	possible	yes	–
surface sensitivity	unimportant	negligible	unimportant	negligible
clamping	sensitive	simple	sensitive, fair	simple
material	PTFE...	BUNA-types	PTFE...	BUNA-types
p bar/t °C	500/150	200/100	350/150	200/100
thickness mm	0,5 - 1	3 - 6	1 - 2	3 - 6
deflection h_M/d_B	< ± 0,2	< ± 0,15	< ± 0,08 ÷ 0,12	
pre deformation profile	calotte (automatic)	calotte (automatic)	corrugated	calotte (h_M) or undulated profile



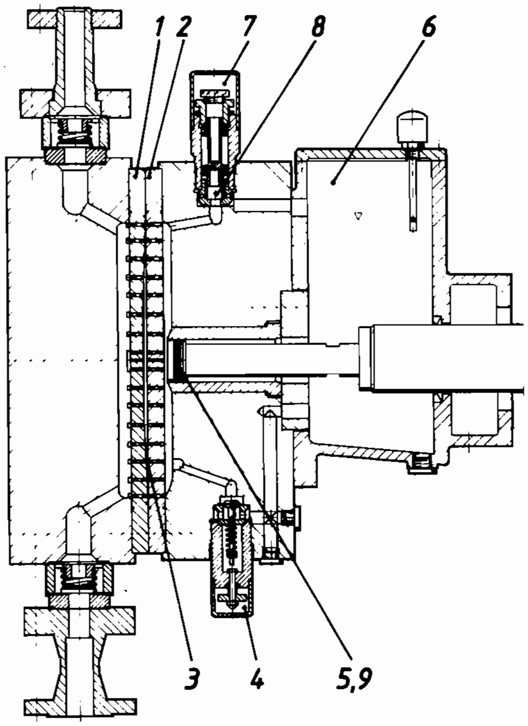


Figure 24. Diagram Head with Metal Diaphragm (350 bar). 1) — 2) hole plates, 3) diaphragm, 4) snifting valve, 5) — 9) plunger seal, 6) reservoir, 7) — 8) relief/venting-valve.

Metal diaphragms. For metallic materials, only plain-parallel circular disc diaphragms can be used together with hydraulic systems (Figure 24) whose hydraulic fllling is controlled by negative pressure (snifting valve 4) and concave upper limitation plate (1).

The diaphragm design, which is very similar to diaphragm compressors, can be performed by strain gage stress analysis and numerical computation (Figure 25). Metal diaphragms permit only small deflections. As they are very sensitive to dirt particles (local notches), strainers are required on the fluid and hydraulic side. The flow must be symmetric, local adhering effects and plus/minus motion should be avoided. The bore-holes (several millimeters) and the screen-profile in the plates are tiny and expensive. Since the tight clamping of metal diaphragms needs high compression, very smooth and precise sealing geometry, the application of elastomer O-ring seals is an improvement.

Figure 25. Testing Diaphragm Stress with Strain Gages.

There is no question that metal diaphragm pumps are the ultimate measure to solve a difflcult pumping problem. Nevertheless, multicylinder pumps (up to 150 kW) have proven reliable with many years of operation (e.g., 300 bar ammonia + potassium amide). Experience shows that diaphragm life exceeds 3000-5000 hr at stroke frequencies below 150 to 200 min^{-1}.

Plastometer diaphragms (PTFE). Historically considered, the progress from metal to PTFE diaphragms was the breakthrough for extended diaphragm pump application. As the possible deflection is extraordinary and still improving, the dimensions of high pressure plunger and diaphragm pumps are more and more approaching one another, although diaphragm pumps are still clearly more expensive. Plastomer diaphragms are used in plain disc or corrugated shape (Figure 26); the ideal design is not yet determined.

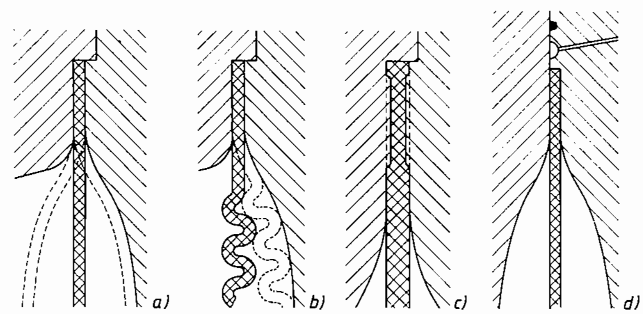


Figure 26. Plastomer Diaphragm. a) plain parallel, b) corrugated, c) restricted compression (grooves), d) pressure balanced.

The plain diaphragm is simple (Figure 26 (a)), and can be flexed to the rear limitation (high suction pressure) without problems; but in order to achieve large deflection, thicknesses must be small (0.5 to 1.5 mm), which requires very high precision at the clamping zone and long-life pore-free material. The corrugated diaphragm (Figure 26 (b)), suitable for essentially the same plus/minus deflection, may be thicker (1.0 to 2.0 mm), maintains symmetric shape during operation, and stress can easily be restricted to elasticity; but it is more expensive, difficult to manufacture, and not suitable for high support compression (e.g., high system pressure). Currently, both types of diaphragms are being used, the majority being plain, and have proven to be good.

The clamping, which has to hold and seal the diaphragm, must be performed by the "restricted compression" principle, which implies a geometrically exact defined clamping volume a certain percentage smaller than the original diaphragm volume in the clamping zone. It has proven good to design the clamping zone with grooves according to Figure 26 (c), the grooves acting in some respect as lip seals. For pressures exceeding 300 bar, it is possible to relieve the diaphragm stress in the clamping zone by a pressure balance from the hydraulic system (Figure 26 (d)) [23]. By this measure, the clamping is separated from the sealing function, and the design has proven reliable up to 1000 bar for elastomer and plastomer diaphragms.

The experience from many pumps with PTFE diaphragms indicates diaphragm life-time of 10,000 to 20,000 hr at most operating conditions ($p < 350$ bar, $t < 150 <$ °C) and normal stroke frequency of 150 to 250 min-1. The experience includes fluids like ammonia, different liquid gases, solvents, catalyst suspensions, organic and anorganic acids, water and chlorine.

Compared with plunger pumps and their plunger seal problems, the diaphragm pumps prove a real advance in reliability and maintenance. Continuing progress and reduced costs can be expected through further development, especially in diaphragm shape and size.

Elastomer diaphragms. There is a continuing challenge to apply elastomer diaphragms due to their larger deflection (Table 3) and thickness (4.0 to 6.0 mm).

For the high pressure slurry application (Figure 3), the elastomer diaphragm is "soft corrugated" and deflected by around

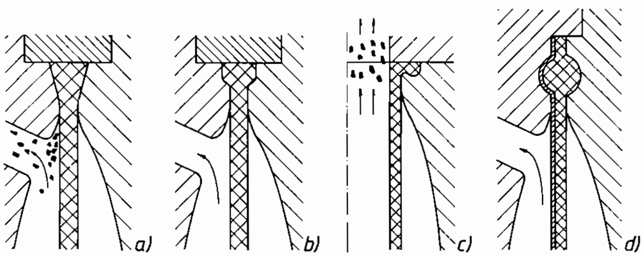


Figure 27. Elastomer Diaphragms and Hoses. a) clamping zone conical, b) clamping zone groove profile, c) clamping of an elastomer hose, d) O-seal shaped diaphragm (coated).

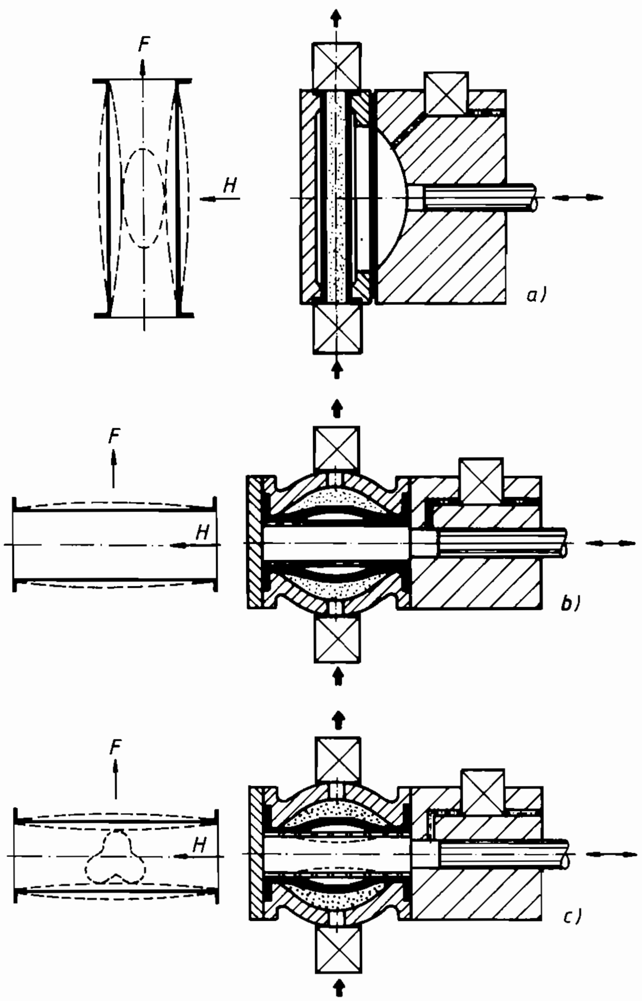


Figure 28. Different Principles of Hydraulic Hose-pumps. a) disc diaphragm (control) combined with hose, b) direct displacement (control) through the hose, c) as b) special shaped support pipe increase deflection volume.

± 20 percent of the clamping diameter. At the rear position, the central cone is supported at the central plunger bore seat, so that the pressure differential is well supported by a hydraulic fluid layer.

The clamping of elastomer diaphragms, also performed by the restriced compression principle, is much easier because, due to more compression volume, the required precision is not especially high. At the clamping zone (Figure 27) the diaphragms are usually profiled (O-or cone-shape, Figure 27 (a and b). Experience shows that the clamping deformation should be separated from the stress due to diaphragm deflection. The optimum diaphragm shape obviously should be an undulated circular disc, prodominantiy stressed by bending, and showing good deflection intensity and lifetime.

The hydraulic hose (tube) pumps (Figure 28) have not really succeeded in process and slurry applications, except in filtration with pressures below 50 bar, although (Figure 27 (c)) the "hose-diaphragm" shows a bending zone on the hydraulic fluid side, preventing any danger of particle clamping. One reason is that it is difficult to control the hose deflection.

Elastomer diaphragms are rarely used in process industries, due to the lower chemical and thermal resistance of elastomers. Nevertheless, these diaphragms are robust and seldom give clamping problems if compression is restricted (10 to 15 percent). Although current plastomer and elastomer diaphragm designs promise lifetimes of above 10000 hr, there is a requirement for rupture monitoring. The best solution currently is the hydraulically coupled sandwich-diaphragm, which marks the breakthrough for safe diaphragm pumps [11], schematically explained in Figure 29. Two diaphragms (1), (2) are clamped together with an intermediate ring (3), which has a connection to the monitoring system [check valve (4), pressure transducer (5)]. The layer between two diaphragm filled with a suitable fluid (6), then compressed, e.g., by the first discharge stroke, in order to squeeze out the excess coupling fluid through the check valve (4). Now the diaphragms are coupled by adhesion. When a rupture occurs on either diaphragm, the respective fluid penetrates and increases the monitoring pressure, which had been ambient. The detail of the design needs much experience and a sense for high precision, which is illustrated in Figure 30.

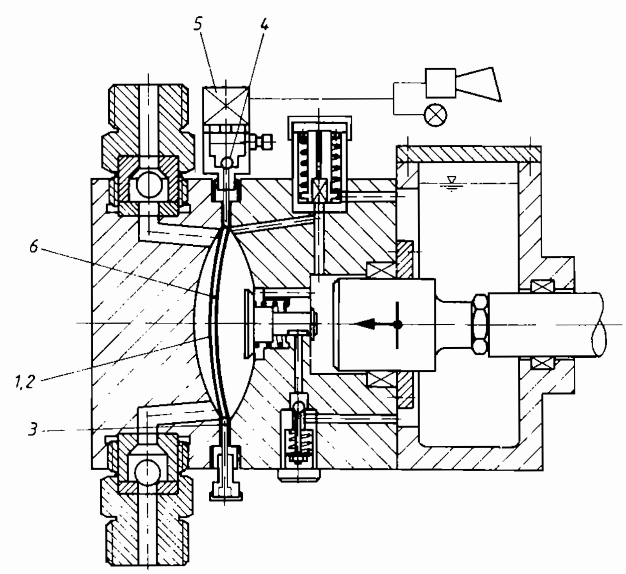


Figure 29. Sandwich Diaphragm Rupture Monitoring System. 1) — 2) diaphragms, 3) intermediate ring, 4) — 5) monitoring system 6) intermediate fluid.

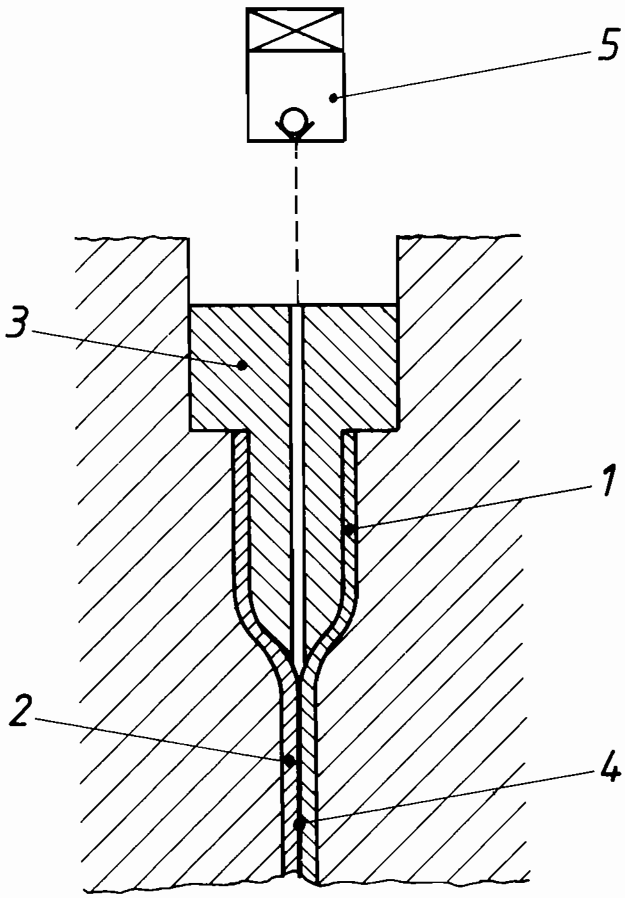


Figure 30. Detail Design of Sandwich-diaphragm (schematically) — 1), 2) diaphragms, 3) ring, 4) coupling fluid, 5) monitor.

Another approach is diaphragm coupling by vacuum [9], which should be less reliable, due to the demand for permanent vacuum tightness. For diaphragm pumps conveying aqueous slurries, the rupture monitoring can also be solved by conductive or capacitive sensors.

The diaphragm design, shape, clamping, and rupture monitoring, is the individual manufacturer's carefully protected secret, because a large amount of detail development is necessary to arrive at reliable performance.

High Suction Pressure

At high suction pressure (Figure 22) the diaphragm is forced to the rear limitation surface at every stroke. If there is a smooth and uninterrupted support area, the diaphragm can withstand the pressure difference between suction pressure and the approximately ambient snifting pressure. This can only be achieved by a very narrow clearance between gate-valve plate (Figure 22 (1 and 2)) and bore. Furthermore, the upper venting bore (Figure 22 (3)) must be very narrow or partially closed by an insert piece.

The suction pressure may then run up to several hundred bar, and normal high vapor pressure fluids cause no problems. For the rather rare application of metal diaphragms which are usually supported by perforated plates at the rear side, higher pressure differential requires a booster supported hydraulic system (Figure 31). The hydraulic working chamber is pressurized by booster pump (1), check valve (2), control system (3) and reservoir (4). At high suction pressure, the booster pressure differential at the diaphragm support is then limited to less than 20 bar.

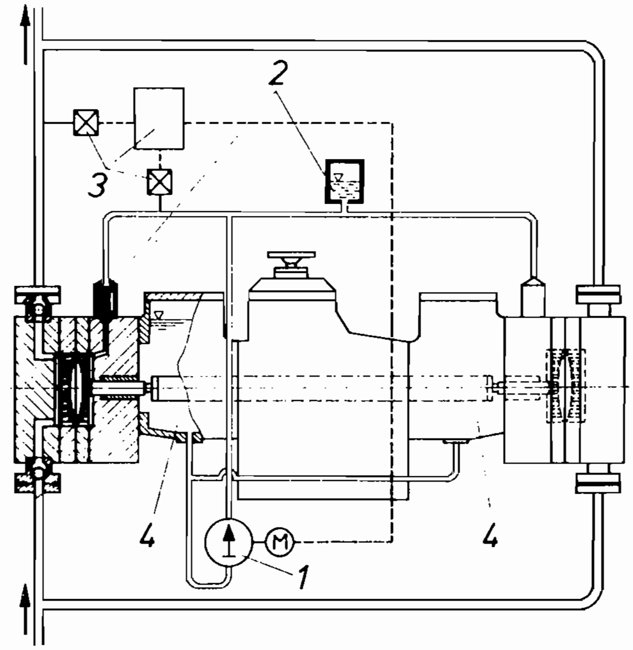


Figure 31. Booster-system for High Suction Pressure Diaphragm Pump. 1) booster pump, 2) check valves, 3) control system, 4) reservoir.

Pump Valves

The automatic motion of the pump check valves is affected by the balance of forces—weight, spring, inertia, flow, and friction, which are acting according to the kinematics of the pump displacement. The periodic motion of the moving valve closing component, cone, plate, and ball, can be computed if the flow resistance factor depending on the Re number is evaluated, e.g., from measurement. For high pressure process diaphragm pumps, it is very important to avoid delayed valve closing because this reduces the tightness factor η_G under *Mass Flow Volumetric Efficiency,* and, thus, implies additional pulsation (Figure 10). Discharge pressure (working chamber) p_A and valve motion h_V for discharge (DV) and suction (SV) valve, if the fluids are incompressible are shown in Figure 32. The valve closing delay (e.g., crank angle $\Delta\varphi_s$ causes a loss in effective stroke Δh_k This effect is further increased by fluid compressibility (indicator diagram Figure 32, right hand/below). For smooth valve motion, which is normally associated with low closing velocity,

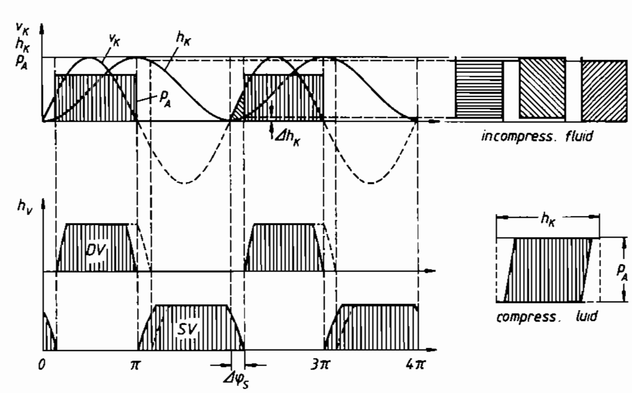


Figure 32. Valve Kinematics. h_k plunger stroke, v_k) plunger velocity, P_A) pressure in working chamber, h_V) valve stroke, $\Delta\varphi_s$) closing delay (crank angle).

the delay for the valve closing should be negligible, which usually can be achieved be by proper springload.

For *homogeneous (clean) fluids* springloaded plate valves (Figure 33 (a)) from different materials (hardened chrome-steels, soft-martensitic chrom-nickel-steels, stellite-coated) for the wearing parts (plate, seat) have proven good for long life. The material choice has to keep in mind—toughness, hardness and corrosion resistance. For large flow, double flow ring-plate valves have to be used. For *heterogeneous fluids* (suspensions, slurries) the tribological system including the moving valve component, the seat and the particles have to be analyzed [24]. The main design criteria are as follows:

• The wear rate increases rapidly with particle hardness. If particle hardness Hp exceeds material hardness $H_M(H_p/H_M > 1)$ the wear rate "jumps" from low to high level. According to experimental results, the ratio should be $H_p/H_M < 0.9$ (Figure 34).

• The particle concentration influences the wear rate, starting very strongly with low concentrations (Figure 35). Tiny dirt contents in fluids, therefore, may force the use of rather hard wear parts.

• Life endurance is better for cone (Figure 33 (b), up to 300 bar) than for plate valves. A certain relative motion between mobile parts (ball, cone) and seat is important, which is optimum for ball valves. Ball valves with elastomer seats are suitable for pressures below 50 bar (Figure 33 (c)).

• As long as wear due to beam like backflow can be avoided, the instaneous wear rate is rather constant. If appropriate, elastomer seal rings, especially mobile with the valve cone, should be used (Figure 33 (c), up to 200 bar).

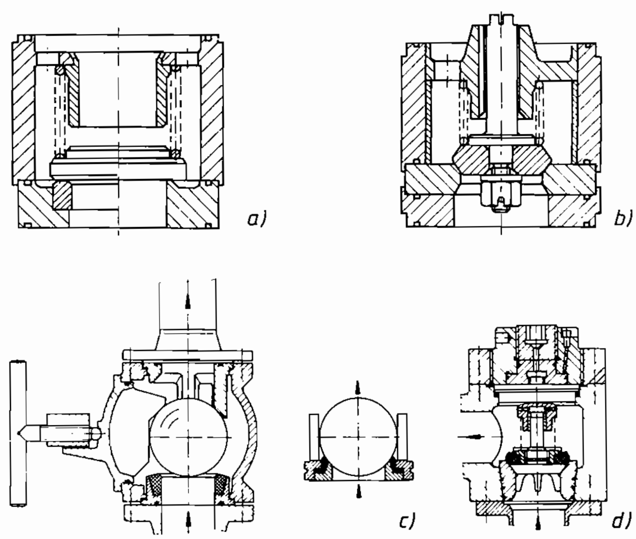


Figure 33. Typical Pump Valve Design. a) plate valve for clean liquids (up to 500 bar, metal wear parts, spring loaded), b) cone valve for abrasive suspensions (up to 300 bar, hard metal wear parts, spring loaded), c) ball valve for slurries (up to 50 bar, elastomer seat), d) cone valve for slurries (up to 200 bar, elastomer ring-seal.

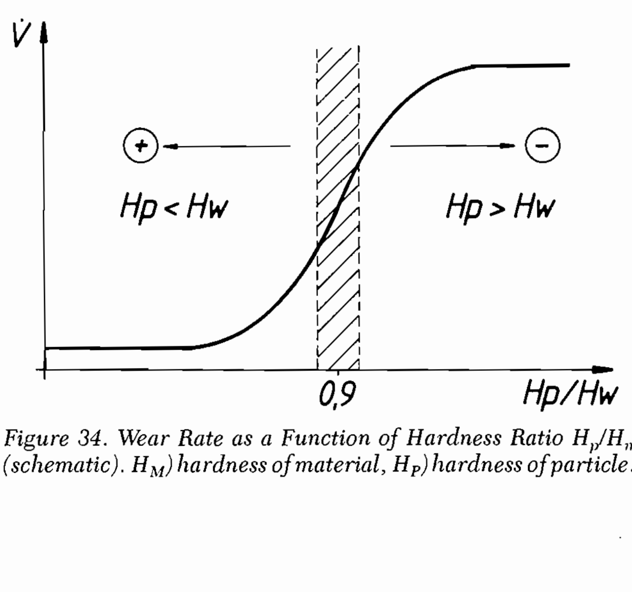


Figure 34. Wear Rate as a Function of Hardness Ratio H_p/H_m (schematic). H_M) hardness of material, H_P) hardness of particle.

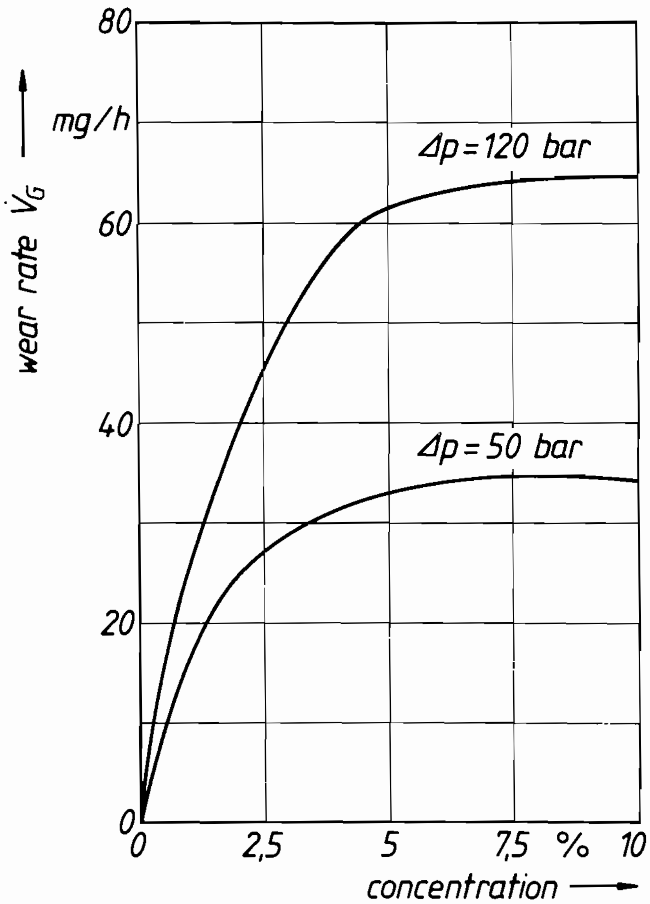


Figure 35. Influence of Particles Concentration on Wear Rate (schematic).

There is extended experience available on high pressure diaphragm pumps for abrasive catalyst suspensions and other slurries. The most favorable valve design is the cone valve (Figure 33 (b)), using very hard materials (e.g., tungsten hard metals) for the wear parts, when high pressure (up to 300 bar), elevated temperature (> 100°C), or aggressive fluids (e.g., solvents), are applied. The cone valve with elastomer sealing-ring (Figure 33 (d)) is the best proven design for noncorressive suspensions, low temperature (< 80°C) and lower pressure (<100 to 200 bar).

The lifetime can be predicted from wear tests which have to be performed either with test pumps or with special tribometers. An example is presented in Figure 36 of the tribological chart of a cone valve, which has run in a diaphragm pump with different aqueous suspensions (10 percent weight). It is obvious that wear is low if material exceeds particle hardness (e.g., GTC/

quartz). Field tests with catalyst suspensions yielded a lifetime of valve wear parts at high pressure (100 to 300 bar) of more than 5000 hr with hard metal GTC. The lifetime prediction from laboratory tests correlated well. The Miller test an important tool for comparing the wear resistance of valve materials and the abrasivity of particles [25].

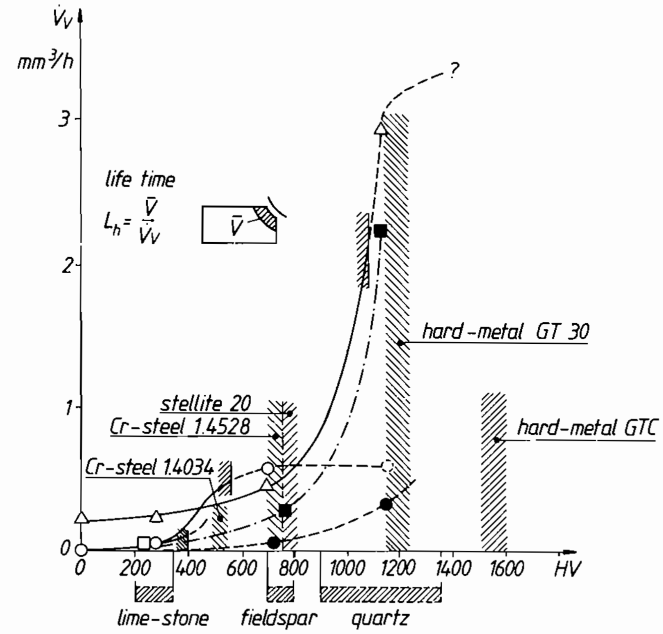


Figure 36. Wear Chart for a Cone Valve (similar to Figure 33 (b) [24] DN). 32 mm, 120 bar, 100 min^{-1}, V_V) wear rate (volume), H_V) Vickers hardness of material and particle, $\bar{V}$) maximum tolerable wear volume, L_h) life time, GTC) 89 percent tungsten carbide hard metal (CrNi-matrix), GT30) 85 percent tungsten carbide hard metal (Co-matrix), Stellite 20) (33 percent Cr, 46 percent Co, 2.5 percent C, 18.5 percent TiC), Cr-Steel 1.4528) (17 percent Cr, 1.0 percent C), Cr-Steel 1.4034) (13 percent Cr, 0.5 percent C).

Some Remarks to Diaphragm Pump Installation

There is no fundamental difference in pipe dimensioning and computation between plunger and diaphragm pumps. As diaphragm pumps usually show higher pulsations (see *Pulsations*), their stronger vibration excitement of the piping system needs careful analysis and damping by friction or hydraulic damps. Numerical and analytical computation methods, which have been experimentally verified [7, 13, 14], are available. It should be pointed out that for many process installations and speed controlled triplex pumps, the resonance between vibration excitement (higher harmonics) and the characteristic values of the piping system cannot be avoided. Therefore, it is sometimes necessary to increase hydraulic friction in order to reduce the vibration amplitudes.

The calculation methods for NPSH to avoid cavitation are identical for plunger and diaphragm pumps. Diaphragm pumps normally require elevated suction pressure due to aeration, which is dependent on the hydraulic system. For the majority of hydraulic diaphragm pumps, it is necessary to sense and warn of minimum longterm suction pressure (e.g., 1.5 bar) in order to prevent diaphragm perforation.

DISCUSSION

Diaphragm pump development of the last 20 years has finally brought forth a superior, reliable substitute to plunger pumps for difficult applications and especially for high pressure. The most important steps in favor of this progress have been:

- the application of PTFE diaphragms
- the development of hydraulic systems for free moving diaphragms
- the invention of hydraulic sandwich diaphragms
- the extension to larger capacity (500 kW).

Compared to sophisticated plunger sealing and leakage restriction systems, the reliability of diaphragm pumps is currently absolutely superior. Lifetime of diaphragms up to 20,000 hr is normal, which among other things, means less maintenance expenses.

However, the investment for diaphragm pumps is definitely higher, so that the economical application is still mainly for dangerous, hazardous and abrasive fluids. The safety argument is very strong, because with modern sandwich diaphragms any contamination can be prevented. Diaphragm pumps with PTFE-diaphragms currently have their limitation at 1000 bar and around 500 kW hydraulic power.

According to pump valve design and pipe dimensioning, there are no fundamental differences between plunger and diaphragm pumps. It should be pointed out that diaphragm pumps are more pulsating and require more attention for careful vibration and damping analysis.

CONCLUSION

Future Trends

The increasing attention on pollution protection will shift the economic breakeven point towards diaphragm pump application. In order to achieve that aim, it will be necessary to develop diaphragms for still more deflection, which would reduce the pump dimensions and costs. There are fair chances for successful development of corrugated PTFE-diaphragm with up to 50 percent more deflection than is currently available, which would help to reduce investment. The hydraulic systems should make use of totally safe position sensing, e.g., by electronic means, in order to reduce sensitivity at extreme conditions and suction pressure requirements.

In considering the historical development of diaphragm pump percentage in production for metering pumps, which has now reached more than 50 percent, the future development for process diaphragm pumps as a substitute for plunger pumps can be predicted rather positively.

NOMENCLATURE

A_K	plunger cross-sectional area
d_K	plunger diameter
h_K	plunger stroke
h_k/h_{k00}	relative stroke setting
$H_{P,M,V}$	hardness (particle, material, Vickers)
K	constant
L_h	lifetime
n	stroke
n/n_{100}	relative speed setting
$\dot{m}$	mass flow
P	power
p_A, Δ_A	pressure working chamber; amplitude

s	clearance, entropy
t	time
T	torque, absolute temperature

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