

PRESSURE PULSATION DAMPENING METHODS FOR RECIPROCATING PUMPS

by
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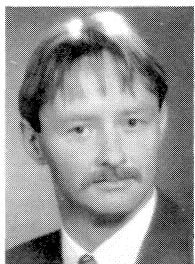
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Gerhard Vetter obtained his Dipl. Ing. degree in Mechanical Engineering at Technische Universität, Karlsruhe, Germany. After some years as a Research Engineer in turbomachinery and thermal power engineering at the same university, he joined Lewa Herbert 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 a chair (professorship) for Apparatus and Chemical Machinery at Universität Erlangen-Nürnberg, Germany, still remaining Lewa's Research and Development 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 textbooks, 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 diaphragm, and metering of bulk solids.



Bernd Seidl obtained his Dipl. Ing. degree in Chemical Engineering at Universität Erlangen-Nürnberg, Germany. In 1988, Dr. Seidl joined the Institute of Apparatus and Chemical Machinery at the same university. As a Research Engineer he performed a thesis about methods of damping for pressure pulsations in piping systems of reciprocating pumps.

ABSTRACT

The safe design of piping installations for reciprocating pumps requires proper knowledge about the pump installation interaction. A compact review shows pump vibration spectra excited by the pumps and methods of numerical computation taking the fluid compressibility and friction into account.

Critical operation occurs in resonance situations where excitement by the pump and the natural frequencies of the compressible continuum in the piping system coincide.

In many cases, resonance effects can be avoided by proper piping geometry, improved pump performance with respect to the volumetric efficiency or higher fluid friction.

If all primary measures have been exhausted, the application of various secondary dampening methods is necessary because the acoustics of the piping system can be influenced. In the range of the excitement frequencies of reciprocating pumps, most of the damper systems are acting as elastic elements (exhibiting spring action) that shift the natural frequency of the hydraulic system away from initially existing resonances.

Several examples for experimental computer simulations of damper action (upstream and downstream) are presented for fluid and gas filled damper versions. The influence of the entry geometry of the dampers is explained, hints on the proper geometry of the entries are given and the effects of various dampers and their influence on the total acoustics are discussed. Special reference is made to the effect of orifices as dissipative and acoustic elements in piping systems, especially where and how to apply orifices to reduce resonance pulsations, change natural frequencies and to achieve helpful filtering effects. The report is supplemented by a case history. The investigation shows the importance of analyzing of the total system of pump piping installation and vessels or reactors and the application of computer simulation for piping design.

INTRODUCTION

Reciprocating plunger, and diaphragm type pumps within their typical application range represent very important components in process technology [1, 2, 3]. The hydraulic power involved extends across a very large range, from small units to ratings of more than 1.0 MW. As reciprocating pumps deliver a pulsating flow, for safe and reliable operation it is very essential to keep the amplitudes and frequencies of pressure pulsations within close limits. In order to ensure smooth operating conditions and to avoid any secondary vibration effects, the design of damping devices requires taking the whole installation of the system into account.

Such an investigation contributes towards understanding how damping systems behave, what interactions between pump and installation system have to be considered, and which design methods have to be preferred.

VIBRATION EXCITATION BY THE PUMP

Reciprocating process pumps are typically characterized by plunger (piston) or diaphragm liquid ends, and speed controlled eccentric crank, or cam and spring return type simplex, or multiplex (mainly triplex) drive units (Figures 1 and 2). For dosing

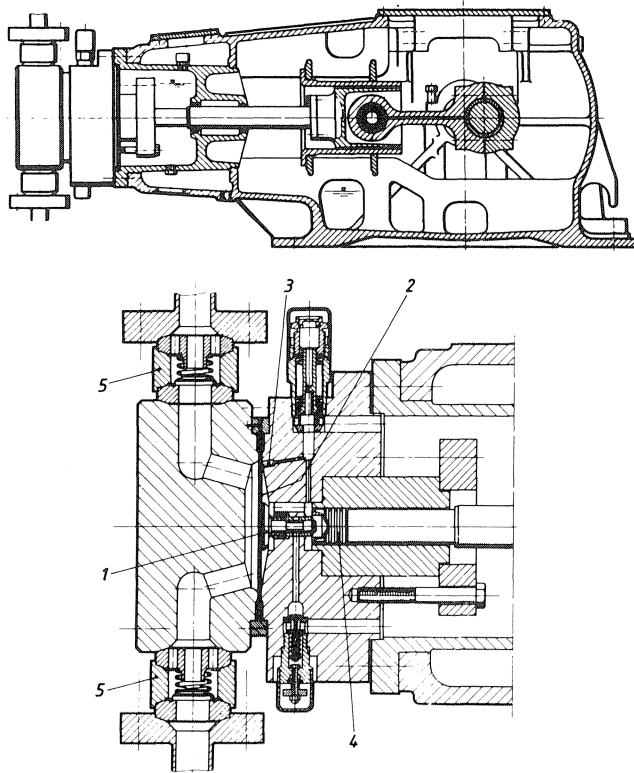


Figure 1. Process Diaphragm Pumps for High Pressure and Toxic Liquids.

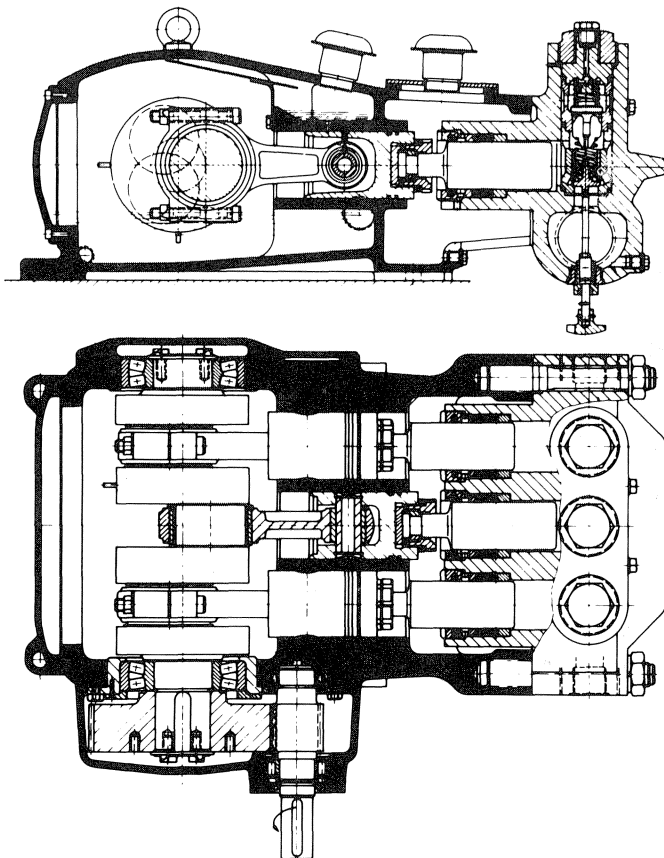


Figure 2. Triplex Plunger Pump.

applications, the stroke of the drive units may be continuously adjustable or via lost motion techniques [4, 5, 6].

Characteristically, the real fluid displacement does not follow exactly the geometrical and kinematic plunger displacement, as there are elasticity and slip (leakage) effects which generate a phase cut φ_2 and, thus, a step Δv in the displacement velocity.

The correlations between the working chamber pressure p_w (indicator diaphragm), plunger stroke h_p , and of the plunger (v_p) and fluid (v) velocity with respect to the crank angle φ (discharge stroke), are shown in Figure 3. The phase cut reduces the effective displacement and at the same time exhibits shock excitation.

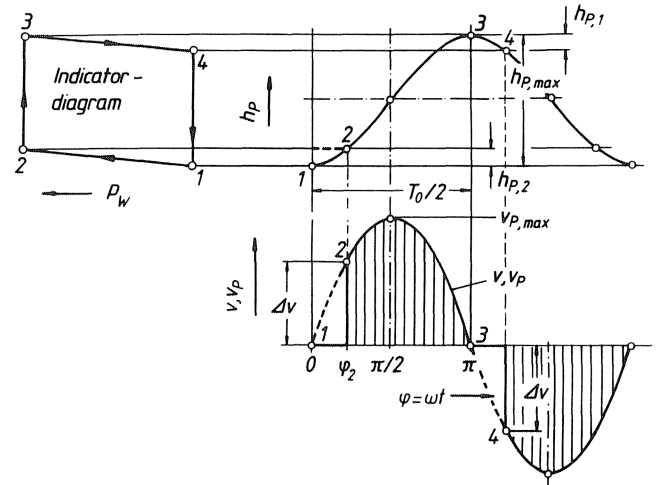


Figure 3. Real Fluid Displacement Characteristic.

The volumetric efficiency η_v reflects the reduction in capacity, including elasticity effects of the fluid and of the working chamber ($\rightarrow \eta_E$) as well as internal and external leakages ($\rightarrow \eta_s$, slip factor):

$$\eta_v = \eta_E \eta_s = \frac{\dot{V}_{eff}}{\dot{V}_{theo}} \quad (1)$$

$$\dot{V}_{theo} = \pi \cdot \frac{d^3}{4} h_p \cdot i \cdot n \quad (2)$$

As the slip factor with well designed valves [7] and plunger seals is normally $\eta_s \approx 1$, the volumetric efficiency is mainly influenced by the elasticity factor, η_E , which can be calculated from Table 1 (Equations (3) through (10)) for various geometrical parameters (relative dead space ε_D), elasticity coefficient of the working chamber, λ , fluid data (compressibility κ), and pressure differential Δp .

Table 1. Equations for the Elasticity Factor $\eta_E \approx \eta_v$ Depending on Pump Features.

Gear System	Type of Pump	
	Plunger Pump ($\kappa_f = \kappa_H = \kappa$, $\varepsilon_D = \varepsilon_{DF} = \varepsilon_{DH}$)	Hydraulic Diaphragm Pump
I) $\delta = 0$	$\eta_E = 1 - (\varepsilon_D \kappa + \lambda) \Delta p \cdot \frac{h_{p,100}}{h_p}$ (3)	$\eta_E = 1 - (\varepsilon_{DF} \kappa_f + \varepsilon_{DH} \kappa_{DH} + \lambda) \Delta p \cdot \frac{h_{p,100}}{h_p} + (\kappa_f - \kappa_H) \Delta p$ (7)
II) $\delta = 1/2$	$\eta_E = 1 - [(\varepsilon_D + \frac{1}{2}) \kappa + \lambda] \Delta p \cdot \frac{h_{p,100}}{h_p} + \frac{\kappa}{2} \Delta p$ (4)	$\eta_E = 1 - [\varepsilon_{DF} \kappa_f + (\varepsilon_{DH} + \frac{1}{2}) \kappa_H + \lambda] \Delta p \cdot \frac{h_{p,100}}{h_p} + (\kappa_f - \frac{\kappa_H}{2}) \Delta p$ (8)
III) $\delta = 1$	$\eta_E = 1 - [(\varepsilon_D + 1) \kappa + \lambda] \Delta p \cdot \frac{h_{p,100}}{h_p} + \kappa \Delta p$ (5)	$\eta_E = 1 - [\varepsilon_{DF} \kappa_f + (\varepsilon_{DH} + 1) \kappa_H + \lambda] \Delta p \cdot \frac{h_{p,100}}{h_p} + \kappa_f \Delta p$ (9)
$h_p = h_{p,100}$ (independent of gear system)	$\eta_E = 1 - (\varepsilon_D \kappa + \lambda) \Delta p$ (6)	$\eta_E = 1 - [(\varepsilon_{DF} - 1) \kappa_f + (\varepsilon_{DH} + 1) \kappa_H + \lambda] \Delta p$ (10)

The elasticity factor η_E is varying with plunger or hydraulic diaphragm liquid end and with the dead center conditions of stroke adjusted drive units [8, 9].

$$\eta_E = 1 - \left(A \frac{\Delta p}{h_p} - B \Delta p \right) \quad (11)$$

$$A = [\varepsilon_{DF} \cdot \kappa_F + (\varepsilon_{DH} + \delta) \kappa_H + \lambda] \cdot h_{p,100} \quad (12)$$

$$B = \kappa_F - (1 - \delta) \cdot \kappa_H \quad (13)$$

The stroke setting ratio $h_p/h_{p,100}$ is important for stroke adjusted drive units ($h_{p,100}$ = maximum stroke, further indices s. nomenclature). The phase cut angle, φ_s , can be calculated from η_E and the plunger stroke kinematics, which are harmonic only for eccentric cam drives and not for crank type systems.

The momentary volume flow is influenced by the volumetric efficiency and the number of superimposed cylinders. Based on the well known crank kinematics (λ_r : rod ratio), the pulsations increase with lower volumetric efficiency and decrease with for higher numbers of cylinders.

It becomes evident that for volumetric efficiencies $\eta_v < 0.8$ to 0.9, which are characteristic for compressible fluids and high pressures, the amplitude of the pulsations, simultaneously containing strong shock type episodes, increases dramatically (Figure 4). The triplex pump ($i = 3$), for example, exhibits at $\eta_v \approx 0.8$ totally "gapping" characteristics. In this connection, the advantage of quintuplex pumps as compared to triplex arrangement becomes evident.

Fourier analysis of excitation caused by the pump furnishes the amplitude/frequency spectra (Figure 5 for a distinct configuration) and demonstrates the strong effect of volumetric efficiency. As the

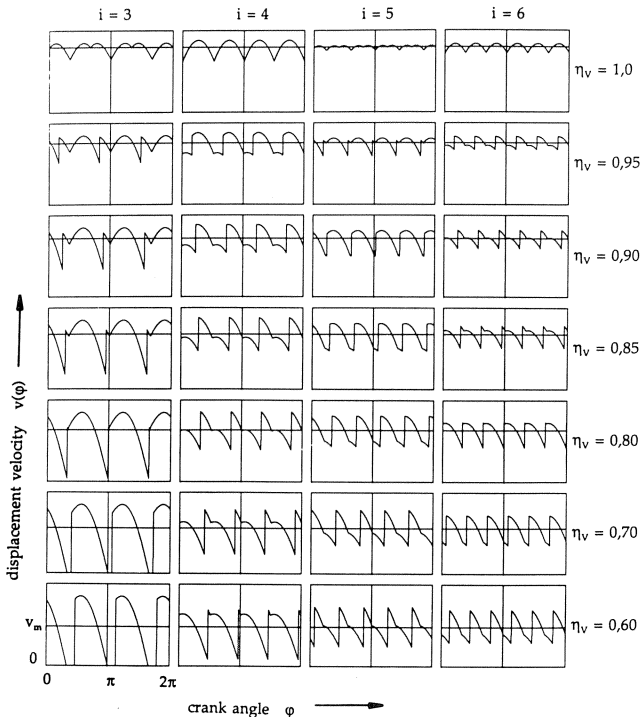


Figure 4. Flow Patterns of Multiplex Pumps (Crank Drive $\lambda_r = 0,2$) Depending on the Volumetric Efficiency η_v and Number of Cylinders i Superimposed [10].

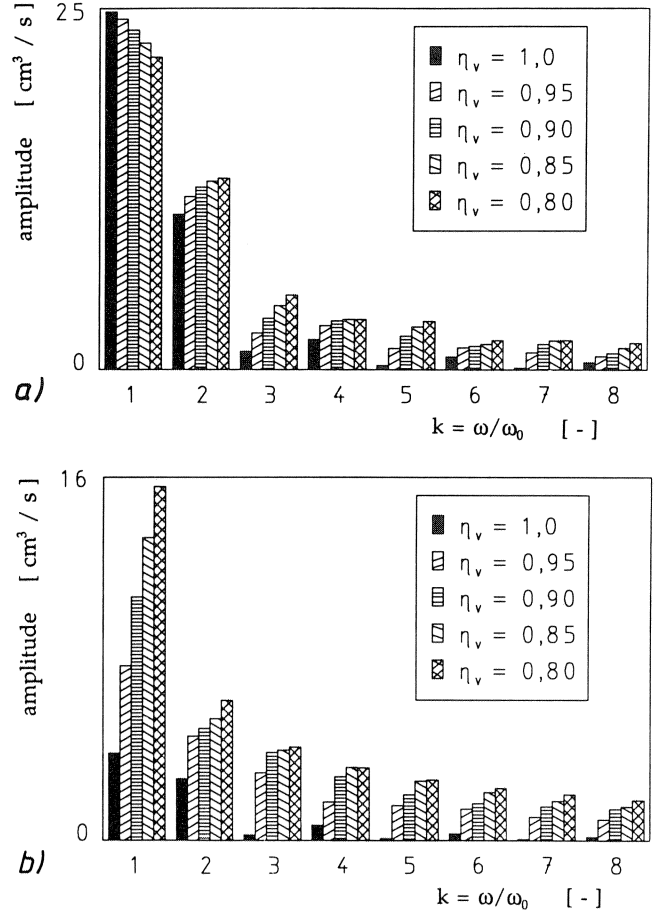


Figure 5. Harmonic analysis of the periodic pump displacement velocity as a function of the volumetric efficiencies ($l_r = 0,2$; $d_p = 20$ mm; $h_p = 30$ mm; $n = 200$ min⁻¹) [10]; a) Simplex Pump; and b) Triplex Pump.

behavior explained is common for all configurations, improved volumetric efficiency by reducing dead spaces should be an important design aspect.

THE RESPONSE OF THE PIPING SYSTEM ON PRESSURE PULSATIONS

The aims for safe piping design are to avoid:

- Cavitation in the pump or system.
- Superflow at the pump.
- Superload in the pump or system.
- Non admissible pressure pulsations or shaking forces.
- Frequencies which may initiate hydraulic or mechanical resonances.

The criteria consequently to be applied to safeguard the pump and the entire installation are:

- Cavitation is avoided if the minimum local pressure (e.g. in the working chamber) always exceeds the vapor pressure of the liquid:

$$P_{w,min} > P_t \quad (14)$$

- Superflow is avoided if the momentary suction pressure is always kept below the discharge pressure:

$$P_{SF} < P_{DF} \quad (15)$$

• Superload is avoided if the discharge pressure is kept below the maximum permissible value:

$$P_{w,max} < P_{ad} \quad (16)$$

• Reduction of the shaking forces to a safe level is generally achieved by limiting the relative pressure amplitude in the pipe system to the admissible value ($x < 2 \div 10$ percent) which is determined from field experience:

$$\frac{\Delta p}{2p_m} < x\% \quad (17)$$

With respect to the pump/installation interaction, the fluid filled system is responding on the operation of the pump with pressure pulsations. Within this context, critical resonance conditions can be observed if any natural frequencies of the fluid filled compressible continuum coincide with distinct and strong excitation frequencies. Since the liquid column may exhibit an almost infinite number of natural frequencies, depending on the pipe length and the velocity of sound, there is good chance for resonances occurring in the system, if reciprocating pumps capable to cause excitation are connected.

The response of the piping system, therefore, consists of either forced and/or resonating pressure pulsations.

A typical pump installation with short suction and long discharge piping is shown in Figure 6. Since both sections are only slightly interfering and are behaving very similarly, it is sufficient for the researchers to concentrate on the generally much more severe vibration problems on the discharge side.

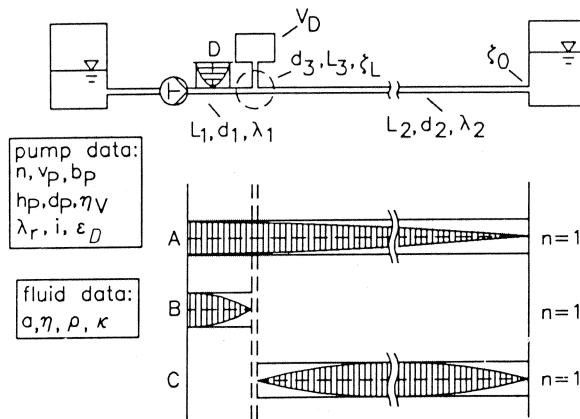


Figure 6. Installation System; Occuring Natural Shapes of Vibration ($n = 1$).

For the basic understanding (Figure 7) of the piping pulsation response, the system can be modelled as serially acting springs, gas/liquid filled damper, plus elastic continuum (Figure 7 (b)) in combination with local (ξ) or continuous (λ_R) fluid friction.

The four sections of the discharge piping system may develop separate but superimposed pulsation responses (shown in Figure 6 in the first order of resonance).

- A = natural frequency of the system (first order: spring/mass system)
- B, C = natural frequencies of the downstream/upstream piping sections
- D = internal natural frequency of the damper connection

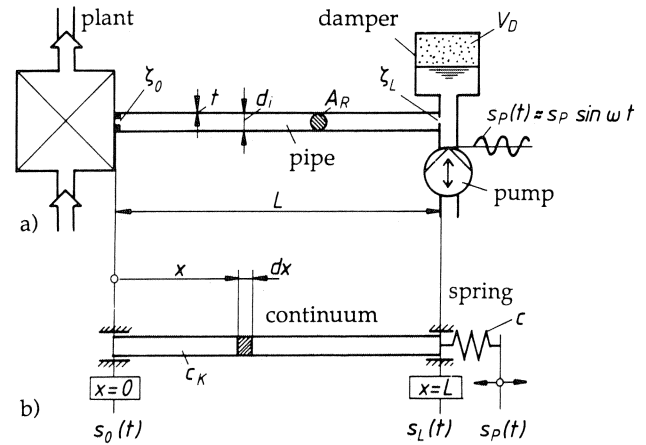
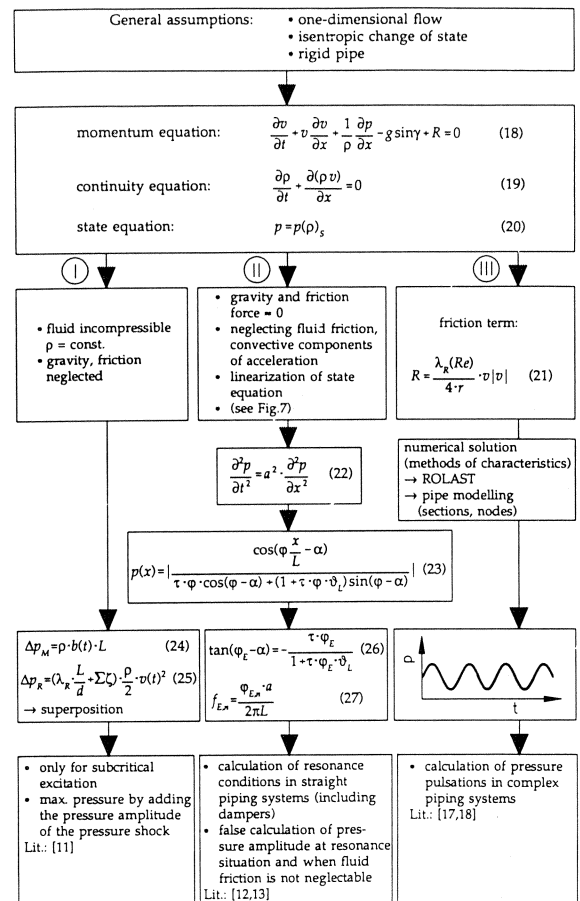


Figure 7. Vibrating System [12,13]. a) installation; b) model.

The pulsation analysis has to cover the whole system at any location. The necessary data are evident from Figure 6 or the NOMENCLATURE. In daily practice, data are sometimes hard to obtain on the velocity of sound in the fluid along the system, the volumetric efficiency of the pump, the elasticity constant of the damper, the pressure losses, the fluid compressibility, and the friction occurring.

Multiple pumps in a single system can have a dramatic effect on the system response and must be evaluated in all running modes.

Table 2. Different Models for the Calculation of the Pressure Pulsations.



It should be clearly stated that, based on the knowledge of pressure pulsations the shaking forces, and, thus, the piping structure vibrations can be evaluated. However, this is not included herein. Furthermore, damping methods have to be applied so that energy dissipation remains low and economical.

It turned out that the estimation of the natural frequencies with the analytical Model II, neglecting friction, favorably supports the numerical Model III. Model II identifies the critical points and furnishes values applicable towards the pulsation analysis by use of Model III at these points.

In addition, it is helpful to realize the pressure shocks excitation as given by the Joukowsky shock theory (Figure 3):

$$\Delta p_j = \Delta v \cdot a \cdot Q \quad (28)$$

By evaluation of Δv for harmonic reciprocating drives (v_m average flow velocity) the shock pressure peaks for reciprocating pumps are:

PRESSURE PULSATIONS THEORY

There is extensive literature available on pulsation theory so that only some remarks for the basic methodical understanding seem to be necessary [10, 11, 12, 13, 14, 15, 16].

The pulsation analysis is actually based on computer simulation, which requires the solutions for the fluid mechanical equations of momentum, continuity, and thermodynamic state for the liquid and yields the local pressure pulsation in the time or frequency domain. Equations (18) through (27) in Table 2 are used to show a survey of three commonly applied calculation models, the terms disregarded, the limitations for the application, and selected literature references. Modern pulsation analysis covering resonance conditions, in fact, almost always require numerical computation methods for reliable results (Model III Table 2). Herein, modelling via computer has been implemented by a computer program based on the characteristics method that supplies the pulsation data in the time domain [17, 18].

$$\Delta p_j = Q \cdot a \cdot v_m \cdot \frac{2\pi}{i} \sqrt{\frac{1}{\eta_v} - 1} \quad (29)$$

METHODS FOR PULSATION DAMPING

Available methods in general are:

- a) Reduction of the excitation by the pump and selection of larger pipe sizes
- b) Avoiding of resonance conditions by adequate piping geometry
- c) Application of damping systems with accumulative, dissipative, interference or filtering effects and shifting of the natural frequencies in the anticipated direction.

Usually, when pulsation analysis has to be performed, there is no chance left for *primary measures* like a) and b) because pumps and pipings have already been selected. Accordingly, the only remaining applications are *secondary measures* like pulsation dampers and/or orifice plates.

From numerous pulsation studies, it seems necessary to establish a good knowledge about physical effects involved in connection with various design aspects.

Therefore, this study is based on the analysis of a number of typical installation configurations with computer modelling and experimental verification.

The test rig (Figure 8) was equipped with a small triplex plunger (EH3) pump, which could run as a simplex pump (EH1) alternatively. The volumetric efficiency could be adjusted by a variable dead space elasticity implemented by means of a piston loaded by compressed air and connected to the working chambers.

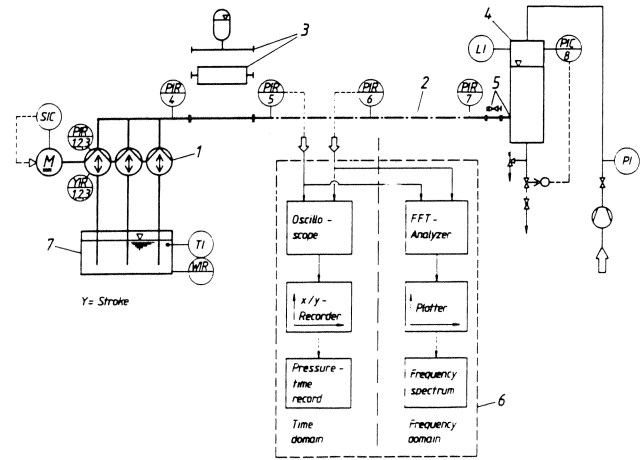


Figure 8. Test Rig [17]. 1) triplex (simplex) pump, 2) discharge piping, 3) damper systems, 4) vessel with pressure control, 5) boundary condition (e.g., orifice), 6) data processing, 7) tank.

For the easy comparability of all data, tests and computer simulation have been performed for standardized installation conditions (Figure 9 (a)), characterized by an extra low volumetric efficiency of $\eta_v = 0.6$, a pipe length of around 100 m with definite reflection conditions at the pressure reference vessel (80 bar) and various pulsation damping devices.

These installation conditions represent the “worst case” including many resonances, small fluid friction, and strong excitation by the pump. The advantage of this method of investigation as compared to other investigations is the coherence of all data, the transferability of design strategies and the practical understanding of parameter sensibility.

A survey of the various installation configurations which have been investigated is shown in Table 3, representing primary and secondary damping methods. In the various figures the installation

Table 3. List of the Installations.

Case	Installation Description	Table	Diagram
Case 1	Fluid friction, pump excitation	Table 1	I
Case 2	Pulsation dampers (gas-filled, liquid-filled)	Table 2	III
Case 2.1	Effectiveness of pulsation dampers	Table 2.1	III
Case 2.2	Pressure pulsations - upstream	Table 2.2	X1, X2, X3, X4
Case 2.3	Variation of the connection device	Table 2.3	IV
Case 3	Orifice	Table 3	V

data and the resonance situations have been tabulated ($f_0 \hat{=} \text{basic excitation frequency}$, $EF \hat{=} \text{natural frequency}$).

With respect to the design of pulsation dampers the attention was mainly directed to their accumulative, resistive, dissipative and natural frequency shifting effects as a component of the whole system. There was no emphasis on detailed design features, because they have been empirically investigated already by other authors [19, 20, 21,22].

PRIMARY DAMPING MEASURES

Volumetric Efficiency

Although in many practical cases, the pumps are already selected and not available for alterations, Figure 9, nevertheless, explains the large potential for improvement determined by the volumetric efficiency. The resonance conditions, remaining basically unchanged, are being reduced drastically for example from $\eta_v = 0.6$ (Figure 9 (a)) to $\eta_v = 1.0$ (Figure 9 (b)), the numerical computation delivering a close fit to experimental verification (Figure 10) [17].

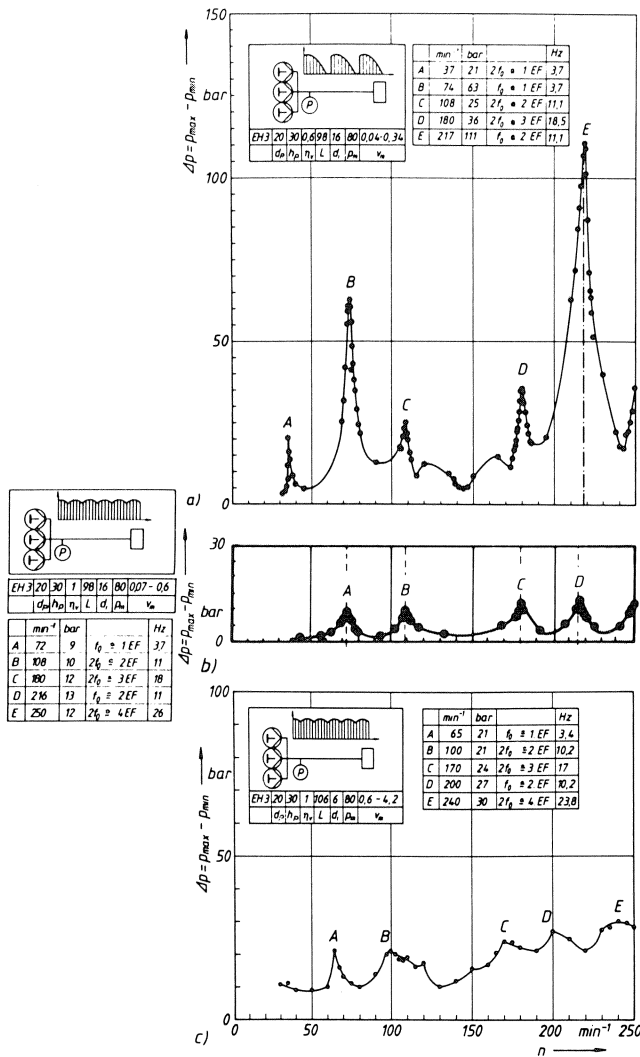


Figure 9. Pressure Pulsations Vs Pump Speed (Triplex Pump) at Different Volumetric Efficiencies and Inside Diameter of the Pipe [17]. a) $\eta_v = 0.6$; $d_i = 16 \text{ mm}$; b) $\eta_v = 1.0$; $d_i = 16 \text{ mm}$; c) $\eta_v = 1.0$; $d_i = 6 \text{ mm}$.

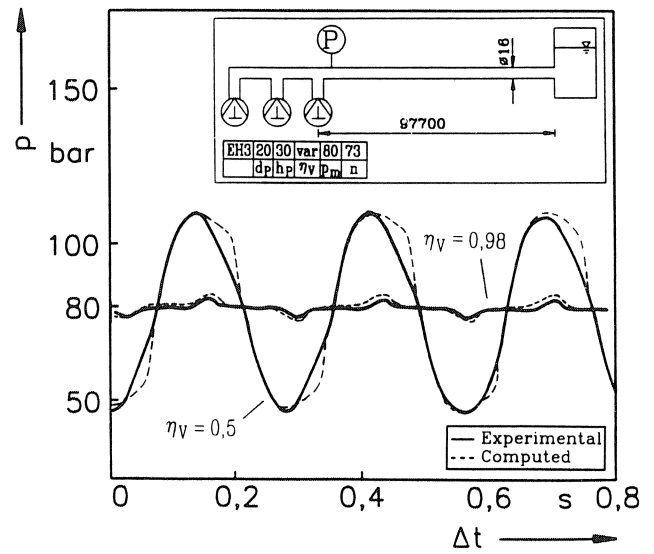


Figure 10. Pressure Pulsations as a Function of the Volumetric Efficiency; Critical Excitation [17].

Obviously, improving volumetric efficiency for simplex pumps at supercritical excitation turns out not to be an efficient means to reduce pulsations (Figure 11 (a): $\eta_v = 1.0$, Figure 11 (b) $\eta_v = 0.6$); this should be considered for pulsation study practice.

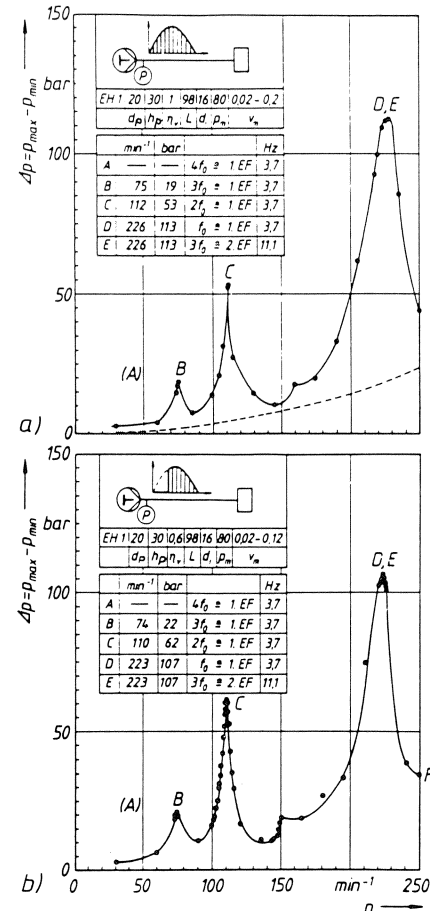


Figure 11. Pressure Pulsations vs. Pump Speed (Simplex Pump) at Different Volumetric Efficiencies. a) $\eta_v = 1.0$; b) $\eta_v = 0.6$.

At subcritical excitation on the other hand, increasing the volumetric efficiency for a simplex pump is definitely a very efficient measure because the Joukowski shocks dominate the excitation (Figure 12), with computer simulation proving this in all cases.

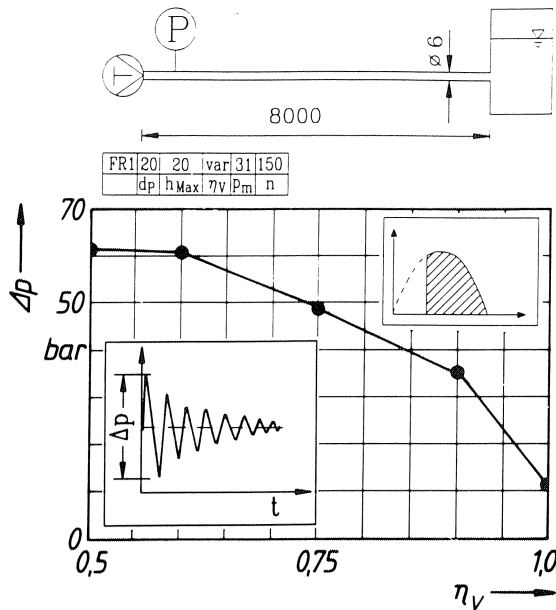


Figure 12. Pressure Pulsations as a Function of the Volumetric Efficiency; Subcritical Excitation.

As a consequence, the improvement of volumetric efficiency is a simple method to reduce pulsation for multiplex reciprocating pumps. The pump manufactures should realize that dead space reduction, especially for compressible fluids (liquified gases) and high pressures, is an easy method to improve pump quality in general.

Fluid Friction

Increased fluid friction by viscosity develops remarkable damping effects especially at resonance conditions (Figure 13 (b)), well predicted by numerical computation (Figure 13 (c)).

The reduction of the pipe diameters to increase fluid friction, on the other hand, is not yielding advantageous results (Figure 9 (c)), since the excitation by the pump is now even stronger, due to the larger ratio of cross sectional areas between the plunger and the pipe (shocks are transduced by d_p^2/d_i^2).

As fluid friction is normally not a relevant parameter, there are no practical but more scientific consequences. Nevertheless, piping systems with viscous fluids develop lesser pulsations.

Length of Piping

Adjusting of the piping length with constant speed pumps as the primary method to avoid resonances just might be mentioned, but can be implemented only rarely.

SECONDARY DAMPING METHODS

It is common that at the time of pulsation analysis neither pump nor piping layout may be subjected to major changes. Damping devices only gas/liquid filled volumes or orifices plates can be selected at that time to meet pulsation limitations.

Most of the published papers about design, action, and computation of damping systems concentrate on the damping device

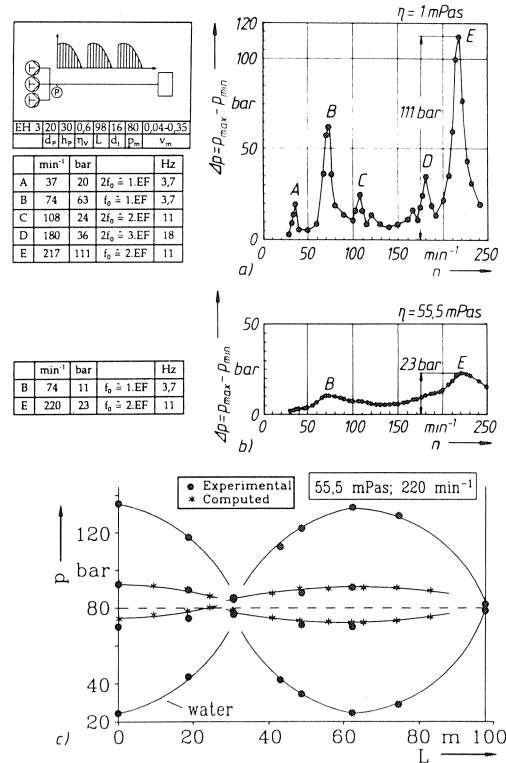


Figure 13. Influence of the Fluid Friction (Pressure Pulsations vs Pump Speed). a) fluid with low viscosity (water; η = 1 mPas); b) fluid with higher viscosity (η = 55,5 mPas); c) Pressure amplitudes along the pipe.

itself and not on the interactions within the whole system. Daily practice, therefore, shows that understanding of how pulsation dampers work and interact is still lacking.

PULSATION DAMPERS

Pulsation dampers per usual definition represent gas or fluid filled volumes (pressure vessels) that develop different actions or combinations of different actions. Their design has to meet the necessary pulsation reduction along with operational requirements like residual time, no sedimentation, safety, and reliability.

SHIFTING OF THE NATURAL FREQUENCY

In the normal range of pump excitation frequencies (up to 50 Hz), the gas or liquid filled dampers primarily act due to their elasticity or spring function. One consequence of the use of such dampers is the immediate change (lowering) of the system's natural frequency.

Small or large gas/liquid dampers perform rather similarly (Figures 14 and 15) at the first view, independently from their geometrical shape, in total accordance to Model II (Table 2, Figure 6). Starting from nondampened installation (Figure 14 (a)), small gas volumes (Figure 14 (b)), and relatively larger liquid volumes (Figure 14 (c) and (d)), are shifting natural frequencies relative to the resonance conditions (e.g., Figure 14 at n ≈ 220 rpm). This damping effect by elasticity can be easily applied for constant speed pumps. The natural frequencies can be calculated by Equations (26) and (27) using the expressions for the elasticity constants in Table 4. There is no need for numerical modelling of that problem.

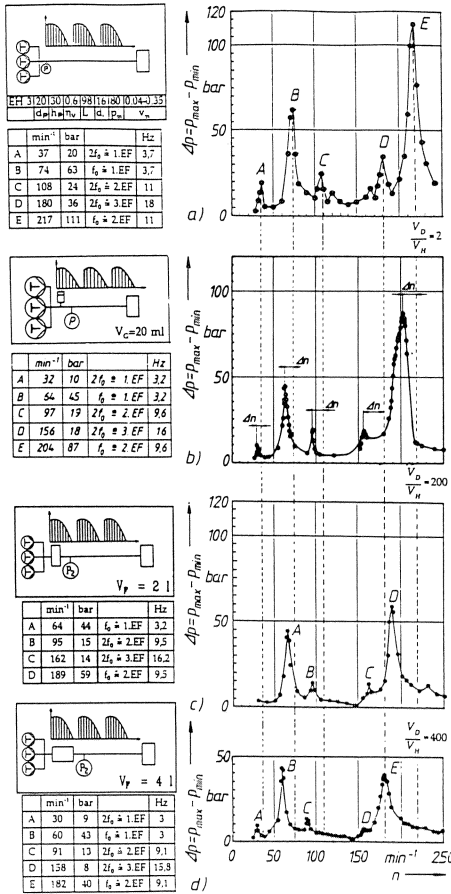


Figure 14. Pressure Pulsations Vs Pump Speed. a) installation without damper; b) small gas filled damper; c) liquid filled damper (2 l); d) liquid filled damper (4 l)

ELASTIC AND ACCUMULATIVE DAMPER EFFECT

With growing damper elasticity, the basic natural system frequency is shifting below the pump's first and strongest excitation frequency for a certain speed range (i.e., Figure 15(b): $f_0 \approx 1. \text{EF} \approx 1,2 \text{ Hz}$, Figure 15(c): $f_0 \approx 1. \text{EF} \approx 1,0 \text{ Hz}$), thus introducing a major damping effect.

As a "rule of thumb," a liquid filled damper, as compared with a gas (air) filled damper, needs, at pressure level of about 100 bar,

Table 4. Elasticity Constants.

	elasticity constant
continuum	$c_K = \frac{K \cdot A_R}{L} \quad (30)$
gas-filled damper	$c = \frac{\kappa_G \cdot A_R^2 \cdot p_m}{V_D} + c_B \quad (31)$
liquid-filled damper	$c = \frac{K_D \cdot A_R^2}{V_D} \quad (32)$

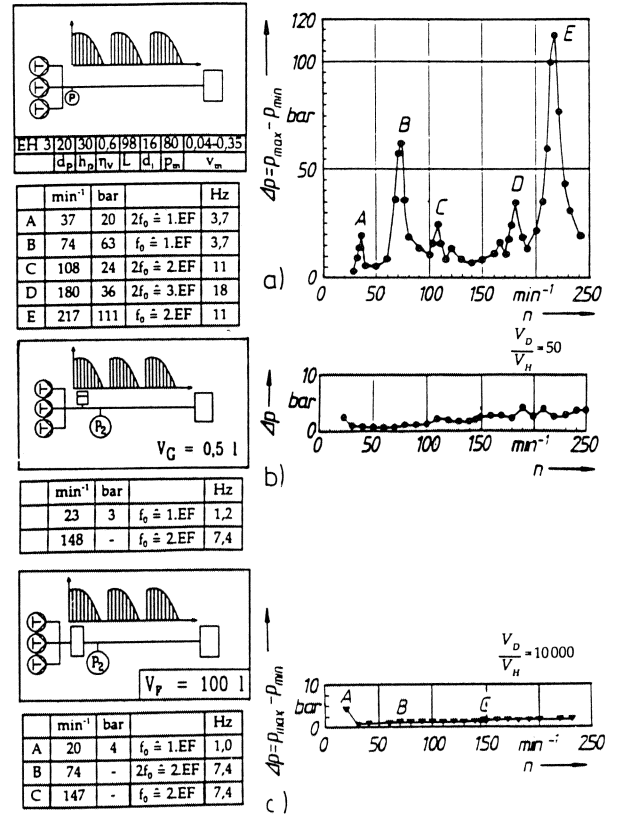


Figure 15. Pressure Pulsations Vs Pump Speed. a) installation without damper; b) large gas filled damper ($V_G = 0,5 \text{ l}$); c) large liquid filled damper (100 l).

around 150 times the volume to provide a comparable elasticity coefficient.

As a consequence, frequency shifting may be achieved in any case. Liquid filled dampers are relatively large and expensive, but may become operationally necessary in cases where fluid contact with a damping gas or a elastomere diaphragm or bladder is not acceptable.

Damping volumes are always exhibiting accumulative effects due to their elasticity, absorbing and releasing pulsation volumes, so that they are damping apart from resonance conditions too.

DISSIPATIVE EFFECTS

Pulsation dampers always exhibit their action by superimposed elastic, accumulative, and dissipative effects due to fluid friction, but regular pulsation damping is optimal when distinct local friction (i.e., by orifice), which does not change system acoustics, is implemented.

Experience shows that damper manufactures often do not mention the fact that their dampers already contain a throttling device, although this is extremely important for modelling.

Starting again is demonstrated at nondamped conditions (Figure 16 (a)), for the example of liquid filled dampers (cylinder/ball shaped), the frequency shifting (Figure 16 (b) and (c)) and the additional damping effect of throttling orifices (Figure 16 (d) and (e)), which increase accumulative action and dissipative friction, especially at resonance locations.

Dissipative effects represent expensive pressure losses, require a certain level (for triplex pumps with $\eta_v > 0,85$, normally 2-5 bar would be sufficient) and need numerical computation methods for design.

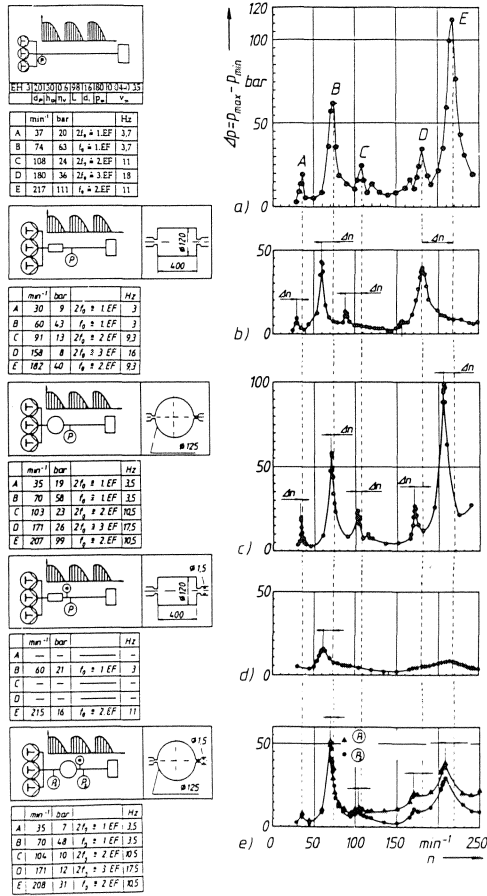


Figure 16. Pressure Pulsations Vs Pump Speed [17]. a) installation without damper; b) cylindrical liquid filled damper; c) spherical liquid filled damper; d) cylindrical liquid filled damper with throttling device; e) spherical liquid filled damper with throttling device.

For all the configurations explained in Figures 14, 15, and 16, computer simulation and modelling (Model III, Table 1) worked well, compared to experimental verification (Figure 17).

PRESSURE PULSATIONS UPSTREAM OF THE DAMPER

The connecting piping between pump and damper is excited especially by the shock contents usually in the subcritical mode. The maximum pressure amplitude can reach the Joukowski shock (Equations (28) and (29)) if the time interval for the velocity step is shorter than the time required for the pressure wave to proceed to the reflecting location and back. Characteristic conditions for simple pumps with $\eta_v = 0.6$ are explained in Figures 18 and 19. The maximum pressure amplitude directly behind the pump reaches the Joukowski conditions (Figure 18 (a) and (b)), if the manifold piping is long enough. It becomes evident that larger pipe diameters (Figure 18 (b)) reduce the pulsation amplitudes there, the frequencies during discharge and suction stroke appearing different due to different elastic volumes (including working chamber) involved (see: discharge stroke 90 Hz, suction stroke 130 Hz). The reduction of connecting piping (Figure 19) length should be realized as much as possible, because this decreases the pressure pulsations.

The consequences for the design are to select short and thick connecting pipings. The pressure pulsation can be approximately

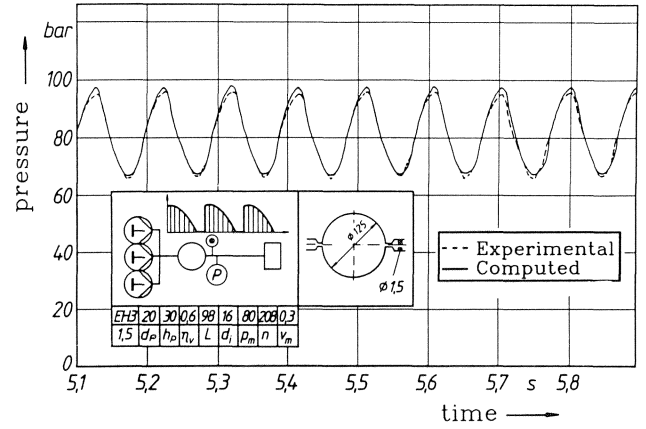


Figure 17. Pressure Time Plot; Comparison Computation Experiment [17].

estimated on the safe side on the base of the Joukowski formula. From there the important parameters involved are evident.

INFLUENCE OF THE DAMPER CONNECTION

Especially for the higher pulsation frequencies, the influence of the connecting geometry is becoming more and more important. The hydromechanical equivalent schematic diagram (Figure 20) is used to explain the importance of resistive and dissipative effects

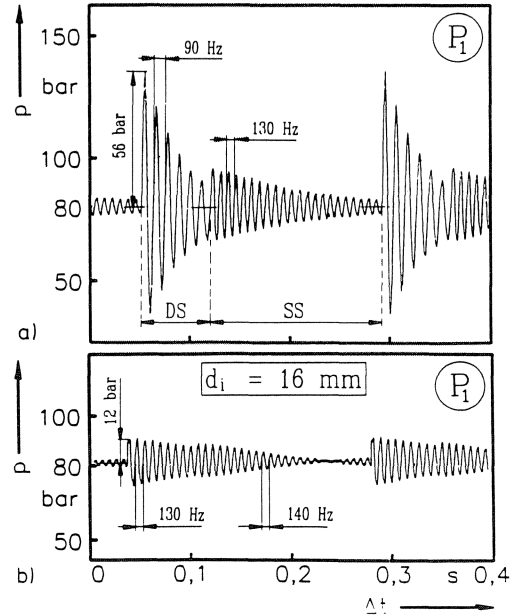
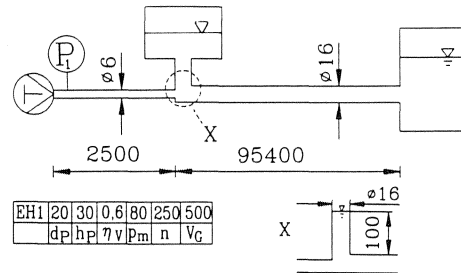


Figure 18. Upstream Pulsations. a) Maximum pressure shock; b) Reduced pressure shock due to increased upstream pipe diameter.

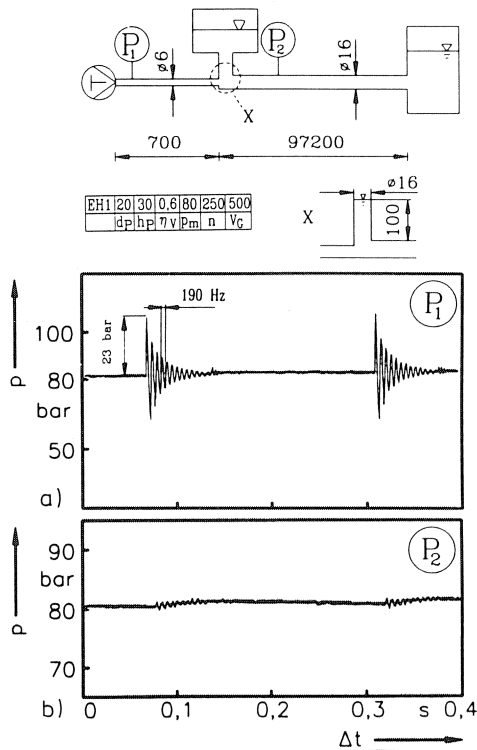


Figure 19. Pressure Pulsations; Simplex Pump; Gas-Filled Damper.; a) upstream; b) downstream.

on the efficient temporary elasticity involved in the system [24], yielding that for good action the ratio of complex resistances z_2/z_3 (Equation (33), Figure 20) should be large.

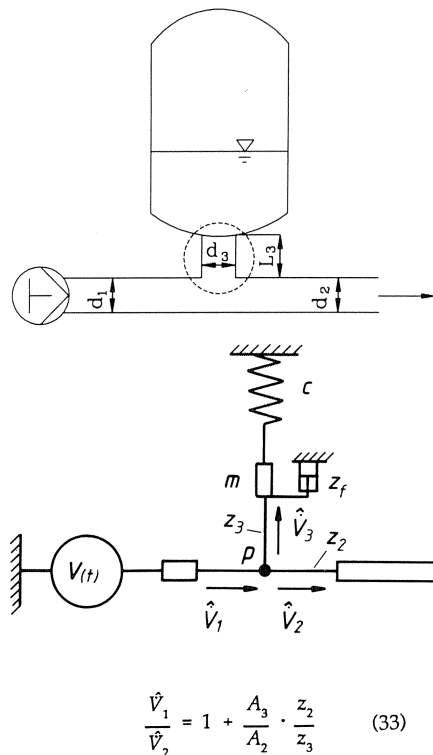


Figure 20. Model of Gas-Filled Damper [23].

To reduce z_3 , the friction z_f and the participating masses (fluid, bladder if appropriate) must be minimized. The complex resistance z_2 is composed of mass acceleration and fluid friction, which can be influenced by pipe length, Reynolds number and local flow resistances (i.e., orifices, elbows, throttling valves), characterized by resistance factors.

Some experiments and computer simulations proved the general parameter influence, well supported by experience and shown in Figure 21 for various systematically chosen connecting geometries (pipe length L_3 , pipe diameter d_3). For small resistance z_3 (Figure 21 (a)) the damping effect is relatively perfect, compared to configurations with larger z_3 . The fluid mass involved being the same, the difference between Figure 21 (a) and (c) can be attributed to the different values for friction and inertial force. The precision of the computer simulation used in this investigation exhibits growing deviations (on the safe side!) from experimental results with increasing flow resistance at the connection which may be due to difficult modelling of the excitation of shocks upstream and the connection flow resistance situation, which have not been investigated further.

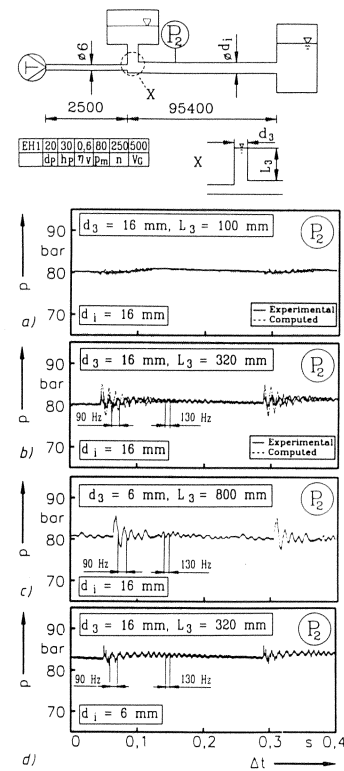


Figure 21. Downstream Pressure Pulsations at Different Connecting Geometry of the Gas Filled Damper. a) $d_3 = 16 \text{ mm}$, $L_3 = 100 \text{ mm}$, $d_i = 16 \text{ mm}$; b) $d_3 = 16 \text{ mm}$, $L_3 = 320 \text{ mm}$, $d_i = 16 \text{ mm}$; c) $d_3 = 6 \text{ mm}$, $L_3 = 800 \text{ mm}$, $d_i = 16 \text{ mm}$; d) $d_3 = 16 \text{ mm}$, $L_3 = 320 \text{ mm}$, $d_i = 6 \text{ mm}$.

The implementation of additional fluid friction (causing a pressure drop of several bars), use of narrower pipe (Figure 21 (d)) or installation of a throttling orifice downstream behind damper, improves the damping effect.

The optimum is achieved by avoiding connection resistance altogether by flowthrough design (Figure 22). For the "appendage design" (Figure 22 (A)), the damper is still bypassed by high frequent pulsations which totally disappear for the flow through design (Figure 22 (F)).

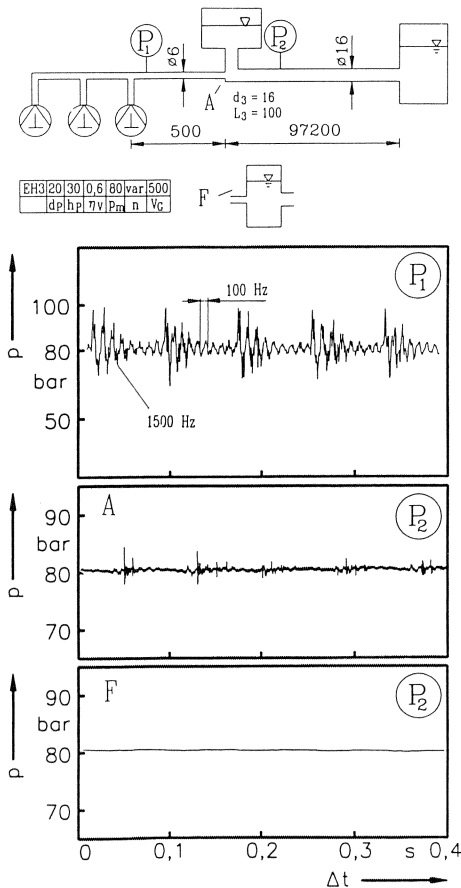


Figure 22. Up- and Downstream Pressure Pulsations. A: gas filled appendage damper; F: flow through damper.

As a consequence, manufacturers should characterize and optimize damper configurations with respect to the connection resistance. Resistance is well reduced with a short and thick connecting pipe or with inline or flowthrough dampers, which are available as gas filled volumes with bladders, diaphragms or direct gas/liquid contact.

ATTENTION TO DISTINCT RESONANCES

Gas filled dampers develop the normally low frequent singular resonance condition of the spring-mass-system, which can be calculated analytically [22] or by numerical computation together with the piping system. In addition to that, the system develops many other resonances of the continuum.

Further, as already indicated in Figure 7, there is a possibility of resonances in the downstream piping behind the damper. The “optimal” configuration of Figure 22 (flowthrough damper) with respect to pressure P_2 behind the damper reveals (Figure 23) within the speed range the typical resonance of a both side open ended pipe with much larger amplitudes than those at the normal check point P_2 .

As a consequence, the pulsation analysis should regard the entire system and measures have to be taken to reduce secondary resonances, too (e.g., by orifices or other dissipative devices).

ORIFICE PLATES

Orifices in piping systems are representing reflection locations, where pressure waves are partially rejected negative inphase if the mode of orifice action is “acoustically open,” depending on ratio

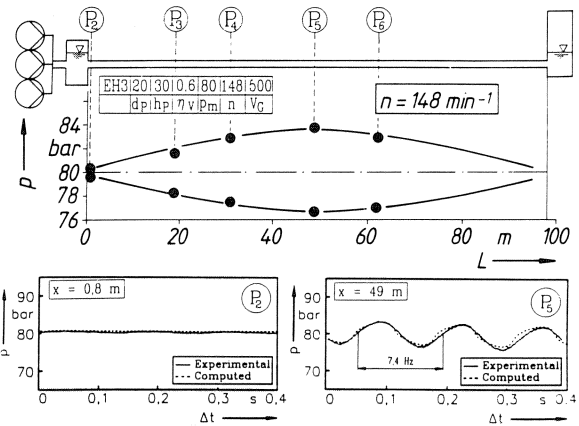


Figure 23. Downstream Pressure Pulsations; Resonance of the Continuum Behind the Damper.

of orifice diameter to pipe diameter, flow, and sound velocity [23, 24].

Acoustically closed acting orifices are not realistic damping devices, because they develop too large pressure drops and are, therefore, not subjects herein.

Orifice action is based on dissipative and interference effects. Experiments show that orifice damping is characterized by their position (Figure 24), which is due to interference effects, as pipe/orifice systems represent cylindrical resonators or acoustic filters.

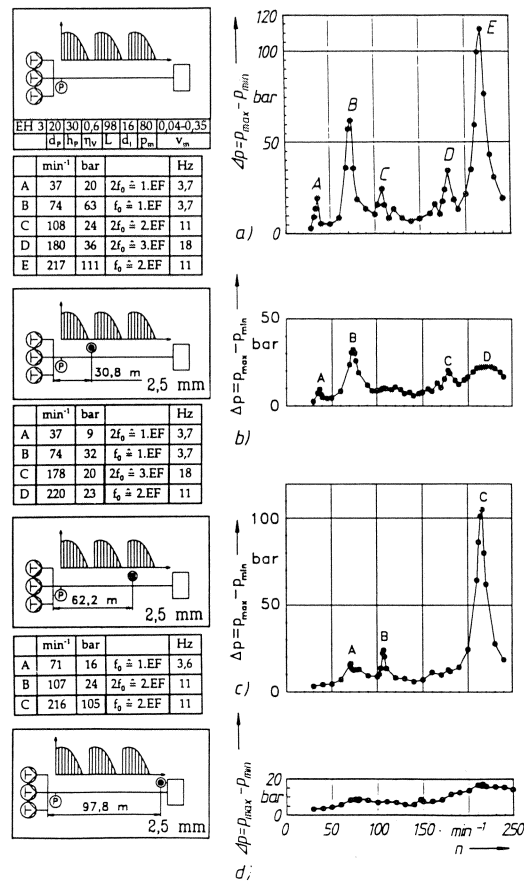


Figure 24. Pressure Pulsations Vs Pump Speed. a) installation without damping device; b) orifice placed at one third of the pipe length; c) orifice placed at two thirds of the pipe length; d) orifice placed at the end of the pipe.

The distance between pumps (excitation emitter) and the reflection (orifice) position to create optimum damping by interference of reflected (wave, inphase antiwave) waves yields theoretically (f = frequency, λ_w = wave length):

$$L_1 = (2n + 1) \cdot \frac{a}{4f} = (2n + 1) \cdot \frac{\lambda_w}{4} \quad (34)$$

If reflected waves superimpose positive inphase waves, no interference responsive damping occurs

$$L_1 = n \cdot \frac{a}{2} = n \cdot \frac{\lambda_w}{4} \quad (35)$$

When tabulating (Table 5) the pipe sections between pump and orifice of minimum and maximum damping for the installation conditions of Figure 24, it can be well understood that the orifice positions at the piping end and at around 30 m are damping well by interference, whereas the 60 m position turns out to be rather inactive, except for dissipative action.

Table 5. Orifice Positions and their Interference Damping Effects.

natural frequency (Hz)	pipe/orifice min. damping (m)	pipe/orifice max. damping (m)
3,7 (1.EF)	0	97,8 (pipe end)
11 (2.EF)	0 64,5	32,3 97,8
18 (3.EF)	0 39,4 78,9	19,7 59,2 97,8

Those vibration modes and partially very impressive damping effects, especially at selected resonance conditions are demonstrated in Figure 24. The damping action of all configurations can be predicted by computer simulation.

It should be realized that the pressure pulsations along the piping system may vary substantially with the position along the length of the pipe and disappear completely at vibration nodes (Figure 25). This is important when evaluating shaking forces and structure vibrations.

The dissipative effect of orifices that require pressure losses (higher energy costs) can be sensitively controlled (Figure 26) by the orifice dimensions, respectively, the resistance factor, without any influence on the piping system accoustics, as long as the reflection mode is not changing.

The application of orifices is very limited, because due to lacking accumulative and frequency shifting action, the damping results are not always meeting the requirements and poor economy may result from rising energy costs. On the other hand, orifices may be inserted on the spot to improve a system immediately without difficult changes.

Combinations of dampers and orifices at the piping system end turn out to be very efficient for certain configurations.

Damper manufactures claim their dampers to act as resonators, acoustic filters or reflection systems and offer various shapes of pressure vessels. From Equation (20), it becomes evident that the size of a cylindrically shaped resonator would have to be very large for the usually low frequencies ($10 \text{ Hz} \approx 35 \text{ m}$, $40 \text{ Hz} \approx 9 \text{ m}$, $a = 1400 \text{ m/s}$) that reciprocating pumps are exciting. This strengthens the argument that all types of damper volumes are primarily acting due to their elasticity. For the higher frequencies, however, that are excited by the larger harmonics, valve bouncing and compression

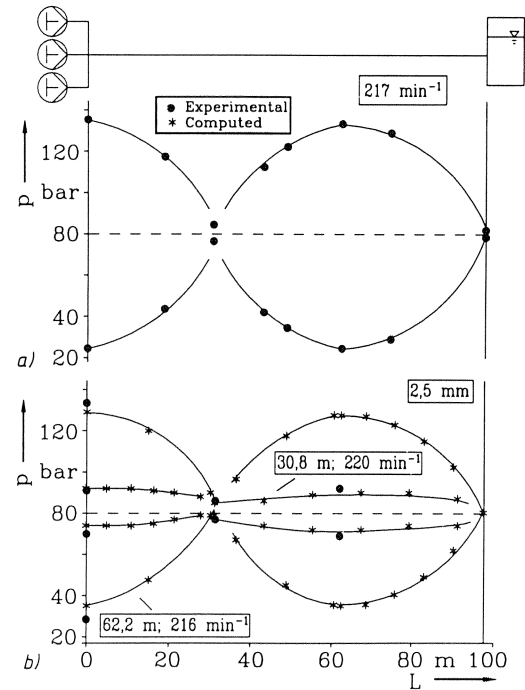


Figure 25. Pressure Amplitudes Along the Pipe at Certain Resonance Conditions. a) installation without damping device; b) orifice placed at different locations along the pipe.

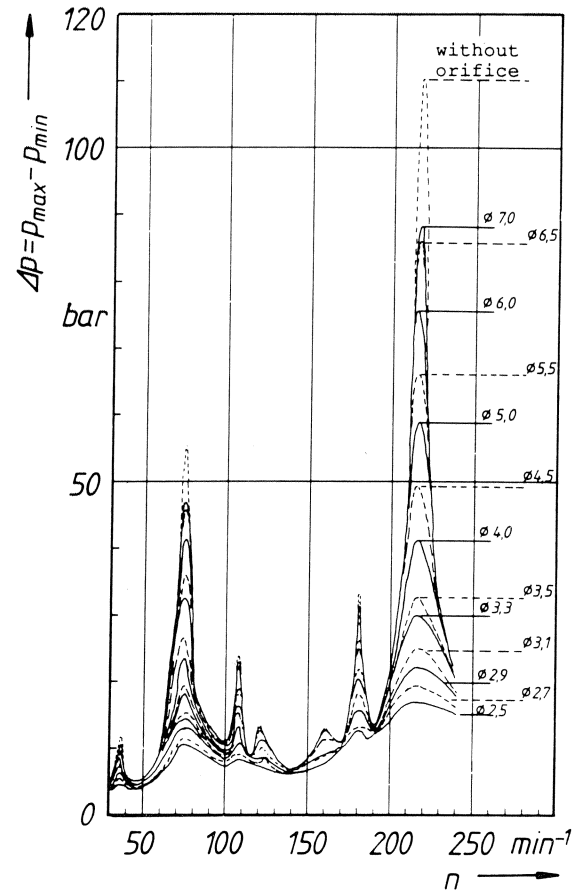


Figure 26. Pressure Pulsations Vs Pump Speed; Different Orifices Placed at the End of the pipe.

shocks in the manifold piping, the interference may contribute to the damping efficiency.

The evidently substantial damping action of high frequencies of the liquid filled damper in Figure 27 (also Figure 14 (c)) should not be traced back to any interference effect, which would, for the existing frequencies (≈ 130 Hz), require a resonator of several meters in length.

In this case, the damping is mainly dissipative in the manifold (reflection at the damper entry) and due to the individual damper pressure losses. These effects may turn out to be relatively important, apart from severe resonances. Whereas for the resonances (Figure 27 (a), (b), and (c)) the superimposed pulsations of high frequency show secondary influence, there is a major damping effect for pulsations apart from resonances (Figure 27 (d)). All those installation configurations may be modelled and the results predicted by computer simulation.

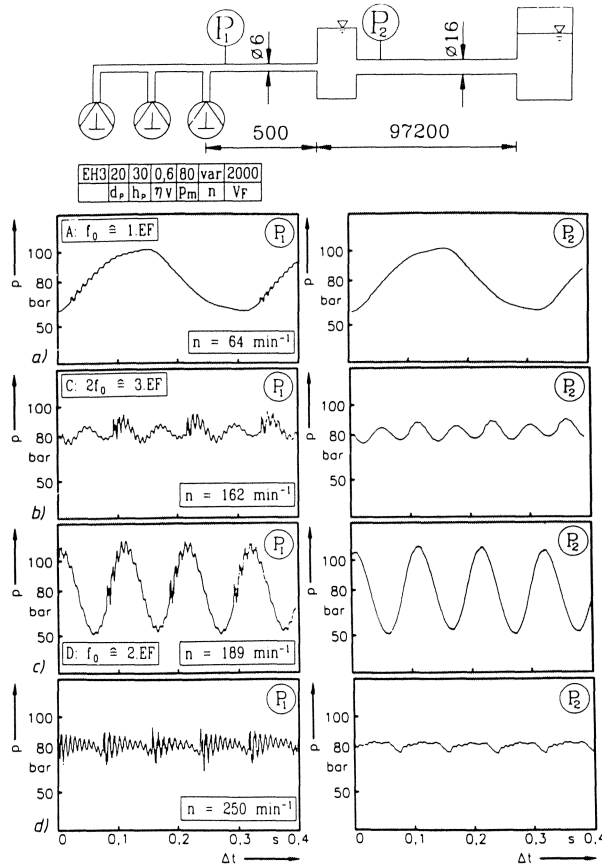


Figure 27. Pressure-Time Plots at Various Pump Speeds Up- and Downstream of the Resonator. a) $n = 64 \text{ min}^{-1}$; b) $n = 162 \text{ min}^{-1}$; c) $n = 189 \text{ min}^{-1}$; d) $n = 250 \text{ min}^{-1}$.

CASE HISTORY

A typical installation (Figure 28) for a urea plant operates with a speed controlled quintuplex plunger pump. The piping consists of two sections with a heat exchanger in between. Together with the projected liquid filled ball shaped damper (volume 115 l), the system showed during computer simulation resonances, the most serious one at 55 rpm, which exceeded the pulsation limit $\Delta p/2p_m = \pm 2$ percent by far. Due to an expected excessively large damper volume and corresponding high cost, there was little chance to increase the size of the damper in order to avoid the resonances excited by the pump. Use of a gas filled damper (with elastomere

bladder) was not acceptable for operational and reliability reasons. The resonance amplitudes could, however, be successfully minimized by a dissipative throttling device directly downstream (or inside) of the damper (accompanied by a pressure drop of 2.0 bar).

The short description of the case history is only one out of a number of successful pulsation studies which have been performed in practical plants during the stage of planning and for failure analysis improvement, respectively.

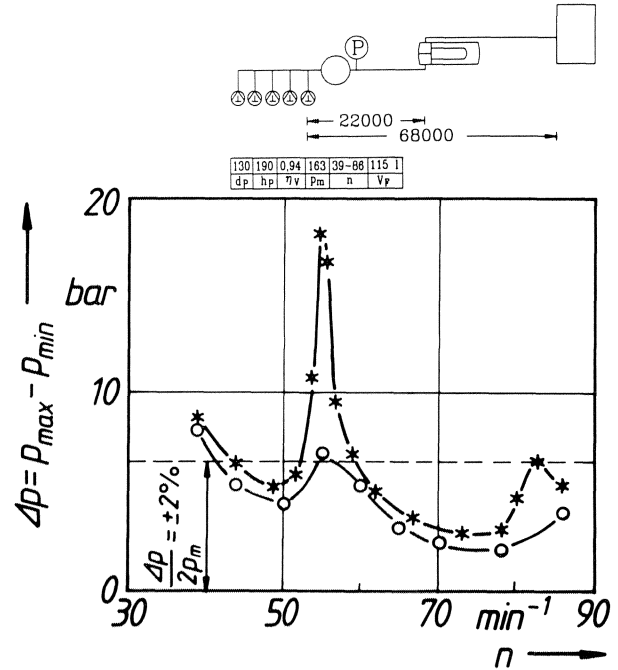


Figure 28. Case History; Calculated Pressure Pulsations in the Piping System Vs Pump Speed. (*) with installed liquid filled damper; (o) with an additional throttling device.

DISCUSSION

The basic strategy to reduce pressure pulsations is to avoid or suppress resonance conditions and reduce excitation energy by primary damping methods at the pump or by secondary damping methods behind the accumulative damper.

In any case, a good understanding of the parameters involved is required to achieve improvements. As a matter of fact, it is difficult to obtain a clear view over all the influences without numerical computation. Obviously, the numerical program used herein is a very good tool and there are other, still more developed ones, available. It seems that modelling of the various pulsation dampers according to their connection resistance data should be improved. It should become standard practice to analyse the entire system and not just an individual damping device.

CONCLUSION

For designing damping methods, it is essential to understand the excitation of pulsation by the pump and the response of the piping system. When performing pulsation analysis, the aims and criteria should be clear, especially at what location which pulsations are admissible.

In the course of such investigations, the use of numerical computer simulation as a tool capable of taking care of compressible fluids and fluid friction, and to be able to model the entire system actually is nowadays mandatory.

Primary damping measures include the piping geometry and the reduction of excitation of pulsations by the pump. There is enough

opportunity for preliminary optimization, if done in time, at an early stage of the project. Secondary damping measures employ damper vessels and/or orifices.

Shifting of the natural frequencies of a system is often a very efficient step, which can be implemented by gas or liquid filled dampers, the latter often becoming rather large and expensive.

The elastic and accumulative action of dampers may be improved by introduction of a necessary minimum of fluid friction by throttling devices (e.g., orifices).

Practical experience reveals that if damper vessels are inserted, it is necessary to take care of pressure pulsations upstream due to reflections. The manifolds should be designed in order to reduce shock excitation.

The damper connection geometry is influencing the damping efficiency, with a flowthrough design, sometimes combined with certain throttling provisions, frequently representing the optimal solution.

Modern computer simulation is able to predict the pulsations of nearly all practical installation configurations very well.

When using dampers, it is necessary to analyse the downstream piping with respect to secondary resonances.

Orifice plates may be an easy tool to improve systems without major changes and investment, if the associated pressure losses and higher energy costs are acceptable. Orifices are effective via dissipative and interference effects but require careful design predominantly by means of computers.

While usual dampers are sometimes called resonators, acoustic filters or reflection dampers, they develop their main effects due to elasticity but not due to interference. For the low excitation frequencies of reciprocating pumps, dampers with large dimensions would be required. The very high frequencies of the pulsations only may be dampened by interference effects for common damper configurations.

The design methods and computer tools explained have been successfully employed in numerous practical pulsation studies and contributed substantially to the design and construction of reliable installations.

NOMENCLATURE

A	m ²	cross sectional area
a	m/s	speed of sound
b	m ² /s	acceleration of fluid
c	N/m	elasticity constant
c _B	N/m	elasticity constant of the bladder
c _K	N/m	elasticity constant of the continuum
d _i	mm	inside diameter of the pipe
d _O	mm	inside diameter of the orifice plate
d _P	mm	diameter of the plunger
f	Hz	frequency
f ₀	Hz	basic frequency of the pump excitation
h _P	mm	length of the stroke
h _P /h _{P,100}		relative stroke setting
i		number of cylinders
K	N/m ²	bulk modulus of the fluid
K _D	N/m ²	bulk modulus of the damper
L	m	length of the pipe
L ₁	m	distance between the pump and the location of reflection
L ₂	m	distance between the location of reflection and the pipe end
L ₃	mm	distance between the pipe and the liquid gas surface of the appendage damper

m	kg	mass
n	min ⁻¹	pump speed
p	Pa	pressure
p _{ad}	Pa, bar	permissible pressure
p _m	Pa, bar	medium pressure
p _{max}	Pa, bar	maximum pressure
p _{min}	Pa, bar	minimum pressure
p _v	Pa, bar	vapor pressure
Δp _J	Pa, bar	pressure amplitude due to the Joukowsky shock
p _{SF} , p _{DF}	Pa, bar	pressure at the suction / discharge flange of the pump
p _w	Pa, bar	working chamber pressure
Δp	Pa, bar	pressure pulsation
R	N/kg	friction force
S	J/K	entropie
s	m	displacement
t	m	thickness of the pipe wall
t	s	time
V	m ³	volume
V _D	m ³	volume of the damper (general)
V _F	ml	volume of the resonator
V _G	ml	volume of the gas filled damper
V _H	m ³	volume of the pump displacement per revolution
\dot{V}	m ³ /s	volumetric flow rate
\hat{V}	m ³ /s	amplitude of the volumetric flow rate
\dot{V}_{eff}	m ³ /s	effective volumetric flow rate
\dot{V}_{theo}	m ³ /s	theoretical volumetric flow rate
v	m/s	velocity of the fluid
v _P	m/s	velocity of the plunger
v _m	m/s	mean velocity of the fluid
Δv	m/s	velocity step
x		coordinate in axial direction
z	Ns/m ³	impedance (regarded to Δv)
α		parameter of integration
γ	rad	inclination angle of the pipe
δ		parameter for the characterization of the crank system
ϑ		coefficient of throttling
ε		relative dead space
ζ		friction coefficient
η	Pas	viscosity
η _V		volumetric efficiency
η _E		elasticity factor
η _S		slip factor
κ	m ² /N	compressibility
κ _G		polytropic exponent
λ	m ² /N	elasticity coefficient of the working chamber
λ _w	m	wave length
λ _r		rod ratio
ρ	kg/m ³	density
φ		plant coefficient
φ	rad	crank angle
φ _E		eigenvalue
φ ₂	rad	phase cut angle

τ		elasticity coefficient (c/c_k)
ω	s^{-1}	angular frequency
EF		natural frequency
EH		pump with eccentric crank drive unit
FR		pump with cam and spring return drive unit

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