SOME TECHNICAL AND PRACTICAL RECOMMENDATIONS
ABOUT PULSATION DAMPENERS IN CIRCUITS
WITH DOSING OR VOLUMETRIC PUMPS

DESCRIPTION OF A PULSATION DAMPENER AND HOW IT WORKS

A pulsation dampener is a vessel with pressurized gas inside, normally **nitrogen**. The initial filling or inflating gas pressure inside the dampener must always be lower than the pressure of the circuit where it is installed. The inflating gas pressure of the dampener will be called "$\mathbf{P_0}$".

In all pulsation dampeners there is a separator element to isolate the gas from the circuit liquid; its main function being to avoid gas leaks. This part that separates both fluids is made basically in two kinds of material: Rubber (**NBR, EPDM, FKM, butyl, silicone,** etc,...) or a thermoplastic material, usually **PTFE**. When rubber is used, the separator element is called bladder and if the **PTFE** is used the dampener can be either membrane or bellows type according to the form of the separator element. The bellows can also be made in stainless steel. The use of one type of separator or another will generally depend on the particular characteristics of the circuit, such as: the working pressure, temperature and the possible corrosive effect of the circuit liquid over the separator element. Below in **Figure 1,** are shown the ***HIDRACAR S.A.*** three different dampeners type mentioned before.

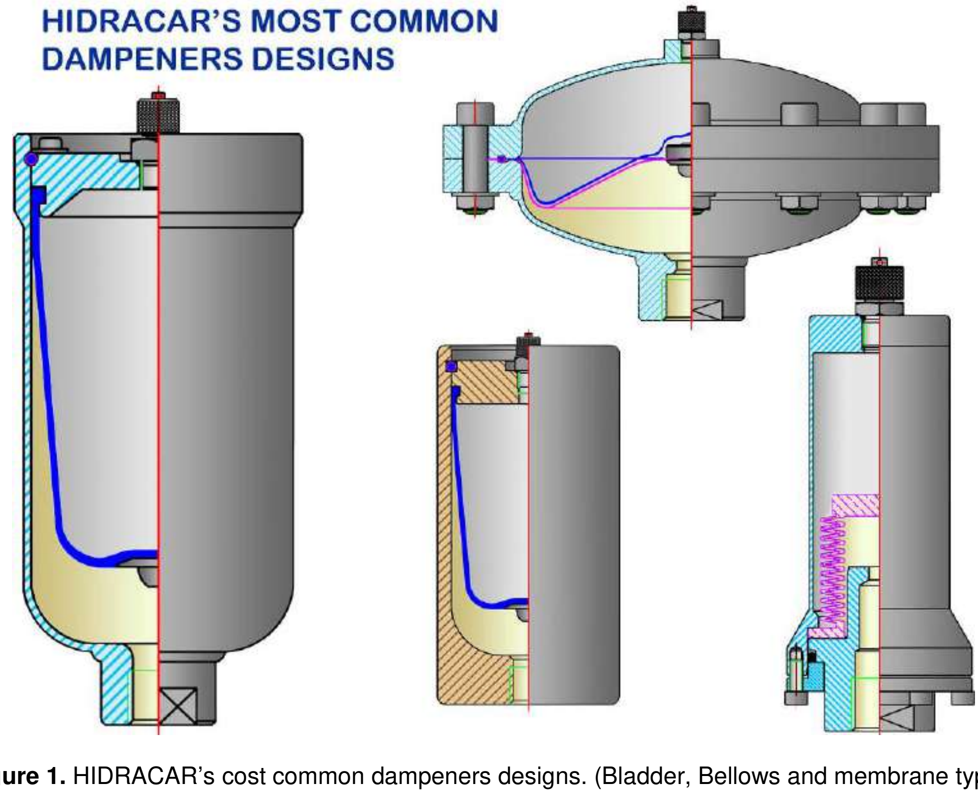

Figure 1. HIDRACAR's cost common dampeners designs. (Bladder, Bellows and membrane types)

THE FUNCTION of a pulsation dampener is to stabilize the variable and oscillating flow generated in a hydraulic circuit in each cycle by volumetric piston or membrane pumps such as dosing or metering pumps. The main function of these pumps being to deliver a constant volume of liquid in every cycle independently of the circuit resistance or pressure (we will later see the characteristics of this kind of pumps), and that's why a pulsation dampener ought to be installed.

When there is a pulsation dampener installed in the circuit, the volume supplied by the pump in every impulse or work cycle is divided in two parts; one goes to the circuit and the other part goes into the pulsation dampener. This volume stored into the dampener is returned right after back into the circuit while the pump is in its suction or chamber filling stage. The amount of liquid going in and out of the dampener in each alternating cycle of the pump will be called "**δV**".

When **δV** gets introduced into the dampener the gas contained inside will be compressed and, therefore, its volume reduced and the pressure increased. The final gas volume (**V_2**) will be the initial gas volume minus the volume of liquid introduced (**δV**).

The initial gas volume is the total volume of the dampener or the size of the dampener. The size of the dampener is an unknown value to be calculated in every case depending on the kind of pump. This volume or size of the dampener will be called "**V_0**"

From all this, we can establish that: $\mathbf{V_2 + \delta V = V_0}$

Every dampener has a constant derived from its size and its filling or charging gas pressure (Boyle-Mariotte law)**:**

$$\mathbf{P_0 \times V_0 = constant\ (*)}$$

(*) This law is only applicable for ideal gases. In practice, this law is not accomplished; later on we will come back to this matter.

In working practice, it is not convenient for the dampeners to get totally emptied of the liquid in each cycle. An extra volume "**v**" is recommended to avoid the anti-extrusion insert of the separator element from repeatedly hammering against the internal bottom surface of the dampener, what could wear prematurely the bladder or membrane out. A new formula results from it:

$$\mathbf{V_2 + \delta V + v = V_0}$$

where "**v**" is a theoretical unused volume of liquid inside the dampener, it is the volume of liquid permanently stored in the dampener. As a norm this volume is considered to be **20%** of the total dampener volume, as long as the temperature remains constant, and, therefore the former formula can be expressed as:

$\mathbf{V_2 + \delta V + 0.2\ V_0 = V_0}$ and finally as: $\mathbf{V_0 = (V_2 + \delta V) / 0.8}$

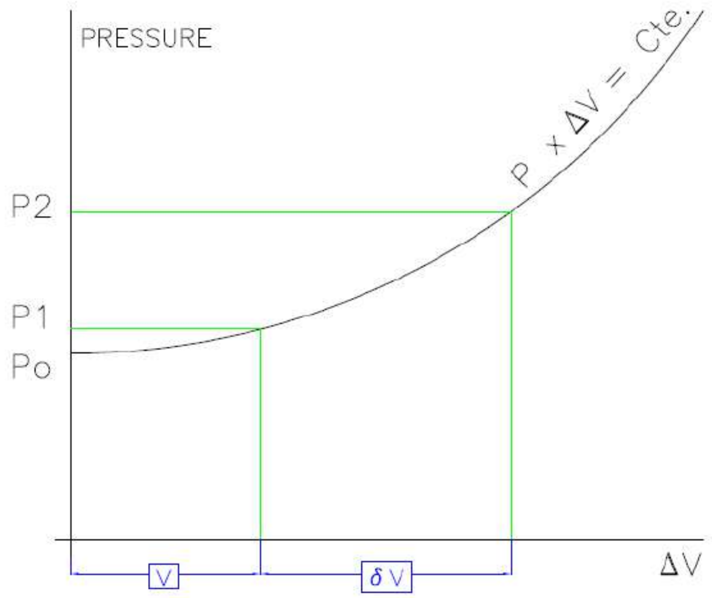

Figure 2. Graph of internal pressure in a dampeners against the volume fluctuations.

The graph in **Figure 2** represents the curve (hyperbola) of gas compression inside the accumulator or pulsation dampener. It is represented the pressure of the gas inside the accumulator against the volume fluctuations.

In the **Figure 3**, we can see the gas volume and pressure evolution at 3 stages (pre-charge, P1 and P2 which are the minimum and maximum pressures in the circuit once the pump is functioning).

At the initial gas charge pressure value "$\mathbf{P_0}$" there is no liquid inside the dampener and the gas fills the whole dampener interior. The curve cuts the ordinate axis in that point where the pressure value is "Po". In the abscissa axis is represented the volume of liquid introduced into the dampener in each working cycle.

The pressure "$\mathbf{P_1}$" is the gas pressure when a volume "**v**" has been introduced into the dampener. The pressure "$\mathbf{P_2}$" is the value reached by the gas when the additional volume "**δV**" is introduced into the dampener.

From the curve in **Figure 2** we can deduce that for a fixed dampener size if the value "**δV**" increases then the pressure value "$\mathbf{P_2}$" will also increase; or the other way around: If we increase the dampener size keeping constant the value "**δV**" the final pressure gas value "$\mathbf{P_2}$" will be lower.

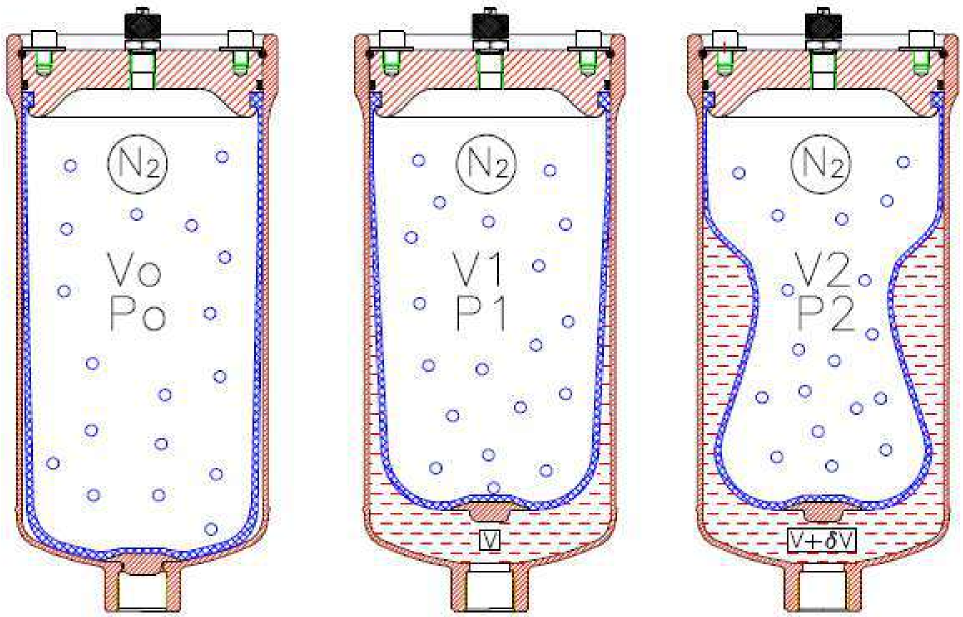

Figure 3. Bladder type dampener in its three stages or internal gas volumes

DAMPENER SIZE CALCULATION

The data needed to calculate the dampener size are:

"**δV**" = Volume of liquid that the dampener must store (in the chapter describing the different types of dosing pumps we will see the relation between "**δV**" and the cubic capacity of each of the three most common types of pumps).

"$\mathbf{P_1}$" and "$\mathbf{P_2}$" are the minimum and maximum pressure values that are accepted in the circuit.

Note: A pulsation dampener does not eliminate 100% of the pressure oscillation produced in the circuits with volumetric or dosing pumps. Its function is to regulate or control the variations of pressure so it remains within previously set limits. This variation, as a +/- percentage of the theoretical pressure "$\mathbf{P_t}$", *is what determines, together with the value of* "**δV**", the size of the pulsation dampener.

"$\mathbf{P_t}$" is the pressure needed at the pump outlet, in order to overcome all the resistances that will arise, to circulate the liquid all the way to the end of the hydraulic circuit.

Let's see an example: If the theoretical or work pressure in a circuit is "$\mathbf{P_t}$" and the residual pulsation admitted is **+/- 5%** of this pressure, values $\mathbf{P_1}$ and $\mathbf{P_2}$ will be:

$$\mathbf{P_1 = P_t - (5/100) \times P_t} \qquad \text{and} \qquad \mathbf{P_2 = P_t + (5/100) \times P_t}$$

With all this known data: **δV**, **P_1** and **P_2**, we can already calculate the dampener size "**V_0**".

The ideal gas law in isothermal conditions (Boyle's law) (later on we will clarify this equation for this application) gives us the following equality:

$$P_0 \times V_0 = P_1 \times V_1 = P_2 \times V_2 = \text{Constant}. \quad (1)$$

If: $V_1 = V_0 - v$ and $v = 0.2 \times V_0$

we have: $V_1 = 0.8 \times V_0$ (2)

and also: $V_2 = V_1 - \delta V$ (3)

Finally, from (1) and (2) we obtain: $P_0 = 0.8 \times P_1$ (4)

and then from (1), (2), (3) and (4) we will get:

$$P_0 \times V_0 = P_2 \times V_2 ; \quad 0.8\, P_1 \times V_0 = P_2 \times (V_1 - \delta V) = P_2 (0.8\, V_0 - \delta V)$$

From the underlined ends of the equalities we obtain the final formula:

$$V_0 = \frac{P_2 \times \delta V}{0.8\, (P_2 - P_1)} \quad (5)$$

This is the simplified theoretical formula to calculate the pulsation dampener volume as a function of **δV**, **P_1** and **P_2**.

As we have already said, it is accepted as a norm that the charging gas pressure, "**P_0**" **= 0.8 P_1**. This difference between **P_0** and **P_1** prevents the complete emptying of liquid from the dampener in each work cycle. Having this extra quantity of liquid **"v"** (stored in the dampener in between **P_0** and **P_1**) can also be used to compensate, in some instances, the potential changes in the gas pressure produced by variations in the exterior temperature that would modify the calculated theoretical "**δV**" and in that case it could not be completely introduced into or discharged out of the dampener.

The former equality (1) **$P_0 \times V_0 = P_1 \times V_1 = \ldots = P_n \times V_n$** does not comply in practice because, when a volume of gas is compressed (in a short time), the temperature rises, what increases the pressure, and when a gas expands its pressure drops an extra value because the temperature is reduced (refrigerator effect). This effect happens with the majority of gases, included Nitrogen and air, which are the more commonly used for charging the dampeners (atmospheric air can be used for pressures below **10 bar**, providing there is no risk of chemical reaction between the Oxygen in the air and the pumped liquid).

The formula (1) gets, thus, transformed into:

$$P_0 \times V_0^{\gamma} = P_1 \times V_1^{\gamma} = \ldots = P_n \times V_n^{\gamma} \quad (6)$$

where **γ** = specific heat ratio of the gas at constant pressure and volume, respectively. For ideal diatomic gases (N_2), **$\gamma = 1.4$** This constant is also theoretical.

We can obtain from both formulas (5) and (6), the **Vo** as a function of the residual pulsation.

If we consider **$\Theta = \pm$ residual pulsation (%) / 100**

From (5). Isotherm curve

$$V_0 = \frac{1 + \Theta}{1{,}6\Theta}\, \delta V \quad (5.1)$$

From (6). Adiabatic curve

$$Vo = \frac{1}{\left(\frac{0.8}{1-\theta}\right)^{1/\gamma} - \left(\frac{0.8}{1+\theta}\right)^{1/\gamma}}\, \partial V \quad (6.1)$$

If we divide the above formulas (5.1 for Isotherm curve) divided by (6.1 for Adiabatic curve), we obtain a relation **K** which is function of the residual pulsation **Θ**. For low values of admissible residual pulsations (below +-5%), the value obtained is practically constant (**K=0,8).** So, we will incorporate the factor **K** in the formula (5), to take in consideration the adiabatic expansion and compression of the gas inside the dampener:

$$V_0 = \frac{P_2 \times \delta V}{0.8 \times 0.8 \times (P_2 - P_1)} \qquad (7)$$

This formula can be used in practice for nearly all industrial applications. It will be very unlikely for the volumes given by this formula to fit any standard dampener volume size from a manufacturer. Except for very exigent applications we can recommend to use the manufacturer's standard closer lower size, favouring cost efficiency.

Note: We have not considered a possible temperature variation of the fluid or environment. This would change the charging gas pressure value at 20º (take note that for each 10ºC variation in temperature the gas pressure will change approximately by 3%).

DIFFERENT TYPES OF DOSING PUMPS TO WHICH A DAMPENER CAN BE MOUNTED

We will consider pumps with one, two or three pistons and crankshaft movement being these the most extended and used and also those in bigger need for a dampener (for air operated, peristaltic, etc... pumps please consult ***HIDRACAR S.A.*** technical department).

The graphics below in **Figure 4** corresponds to these three types of piston pumps and represent the instantaneous flow during a complete crankshaft revolution. We have taken the same piston dimensions (diameter x stroke) for all three types of pumps.

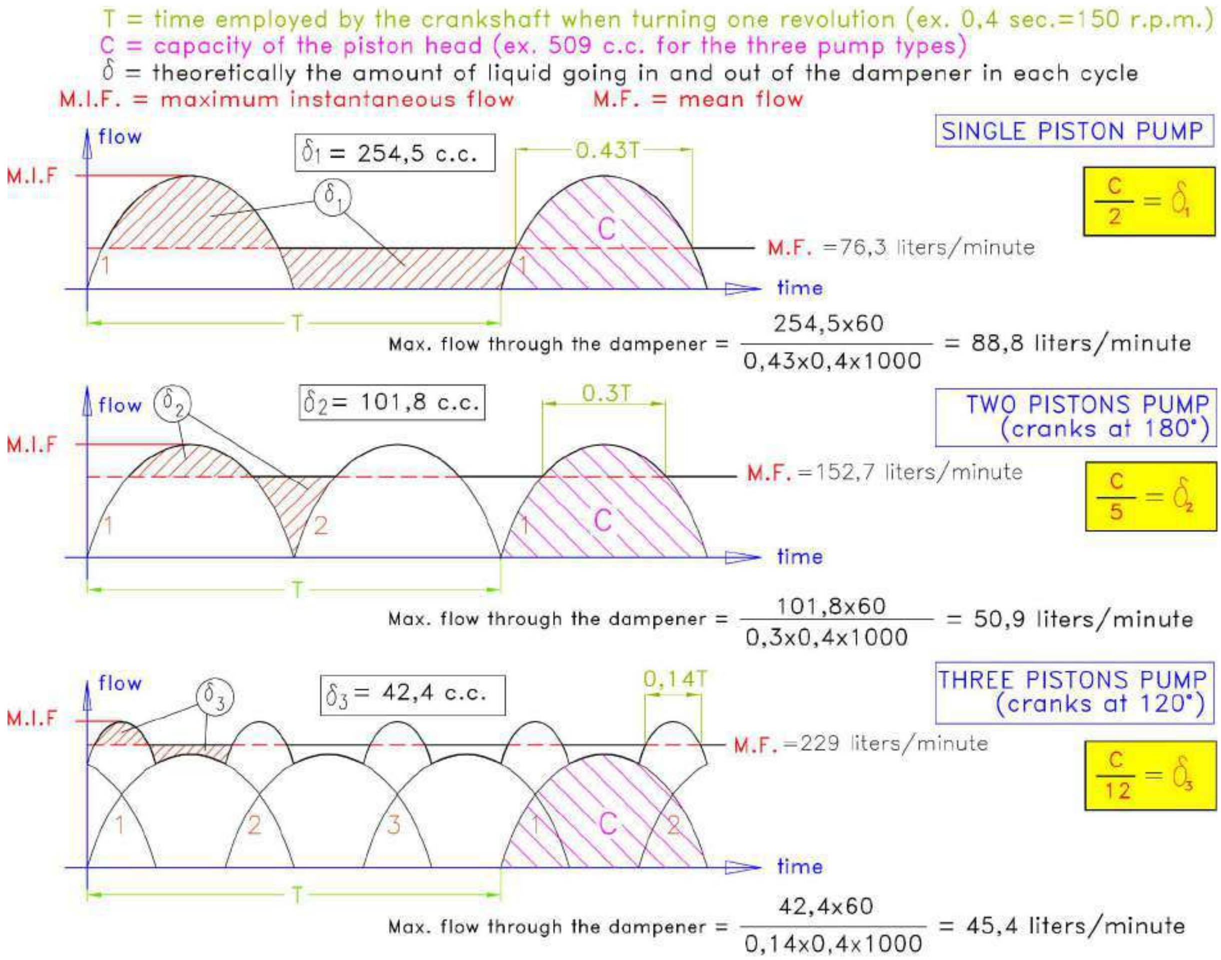

Figure 4. Graph for the instantaneous flow evolution in different pump types. From up to down: 1 piston pump, 2 piston pumps, 3 piston pumps (all of them single acting)

The curves in **Figure 4** let us see how a pulsation dampener works: If we pay attention to the first curve (on the top), representing a single piston pump, we can observe that for this type of pump the use of a dampener is almost essential, as otherwise during half revolution of the pump crankshaft no liquid flow is delivered. Also if the pump does not include a dampener, the diameter of the pipe must be calculated for the maximum instantaneous flow, which takes place when the piston speed is also at its maximum, in the middle of piston stroke (the flow curve is a sinusoid).

With a dampener installed in the pump outlet, the circuit flow will became practically constant. Hence, the pipe diameter downstream the dampener can be designed considering the mean flow. It makes possible to reduce the pipe diameter **by approximately 40%!!** And this because the maximum instantaneous flow of the pump is **2.8** times superior to its mean flow. In some cases this reduction of pipe diameter will compensate the cost of the dampener; furthermore the dampener will stabilize the circuit's pressure, with all of its obvious associated improvements (pressure in a hydraulic circuit is, basically, a function of the flow and losses of head).

Carrying on with the first curve in **Figure 4**, we can see that the task of the dampener is to store all the excess volume over the mean flow line. It occurs during the piston head impulse stroke; and then this volume "δ_1" is returned back into the circuit during the piston suction stroke. So then, in this type of pump the volume stored by the dampener is half of the pump head or capacity per revolution.

As we analyse all three curves in **Figure 4**, we can see that, as the number of pistons in a pump increase, the mean flow gets closer and closer to the maximum instantaneous flow and the liquid volume "δ_1" stored by the dampener gets correspondingly reduced, and therefore the required size of the dampener also gets reduced (this is totally valid in a case like this, where all the pistons in the three pumps have the same diameter, stroke and number of revolutions per minute).

The relation between "**δV**" and the capacity per head "**C**" is

$\delta v = C / 2$ For a one piston pump

$\delta v = C / 5$ For a two piston pump

$\delta v = C / 12$ For a three piston pump

(Practical values for the calculation of the dampener size).

We know that when a gas is compressed its pressure increases, and decreases if it expands its volume. When a dampener is installed in the outlet of a piston pump, the pressure of the liquid in the circuit will fluctuate according to the values of the volume of gas inside the dampener. This pressure variability (a +/- percentage of pressure P_t) will be defined by the technical designer of the circuit or by final customer requirements.

The following graphs in **Figure 5** will help to better understand the above exposed:

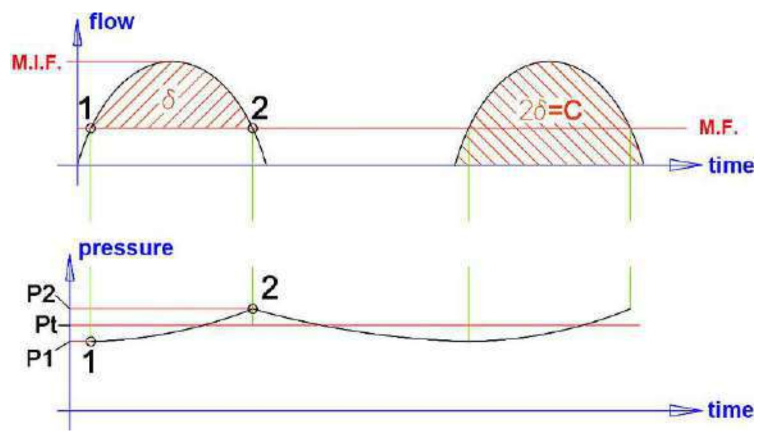

Figure 5. Pressure evolution in 1 piston pump complete cycle with the installation of a pulsation dampener.

Before all, let's consider that for the mean flow (M.F. in the graph) corresponds the working pressure "P_t". When the pump is in its impulsion cycle and the instantaneous flow increases and achieves the point 1 in the graph of the **Figure 5**, the dampener starts to store liquid (see in the top graph which represents the instantaneous flow delivered by the pump and the lower graph where the pressure variability with the use of a dampener is represented). The dampener ought to be charged at the adequate inflating pressure (80% of the working pressure). In the point **1** the damper starts to store liquid, in the point **2** the damper is full of liquid (all "**δV**" has been introduced in the damper). In the pump suction stroke, the damper discharges the volume "**δV**" previously stored.

The lower curve of the **Figure 5** shows the pressure fluctuation of a circuit with a dampener installed. This curve relates to the pump flow variation curve. As we have seen before, a dampener stores the volume of liquid above the pump mean flow. For this reason, the minimum value of the pressure curve (point **1**) must coincide with the first crossing point of the instantaneous flow curve with the line of the mean flow (the time where begins the liquid storage inside the dampener); and the maximum value of the pressure curve (point **2**) must coincide with the second crossing point between them (the time where finishes the liquid storage inside the dampener), in between these 2 points all the stored volume "**δV**" has been introduced inside the dampener.

Let's remember that the area comprised between the instantaneous flow curve and the abscissa axis (time) in Figure 5 top graph represents a volume which in the case of a single piston pump is equal to the pump capacity per stroke or revolution. (flow x time = volume) .

Let's see now the meaning of P_1, P_t and P_2 in the pressure / time curve of **Figure 5**:

In all hydraulic circuits the pressure at the pump outlet port is a function of the flow, pipe length and diameter, viscosity of the pumped liquid, internal pipe surface roughness, geometric height, etc... If the flow keeps constant over time, the pressure needed to pump the liquid will also be constant as long as there is no change either in flow resistance (for instance, due to sedimentation on filters, etc...) We call this constant working pressure or "P_t".

When designing a circuit, the mean flow and the opposing resistances shall be considered to calculate the pressure "P_t".

We have seen that the dampener stabilizes the flow and in fact also the pressure, the pressure in the circuit with a pulsation dampener installed varies from "P_1" to "P_2". The reason behind this is that the dampener has to stabilize the flow and for that it needs to compress and expand a volume of gas, and these pressure variations in +/- percentage of P_t are those that regulate the values accepted in the circuit.

We have already seen that this pressure fluctuation can be reduced to very small values by increasing the volume of the dampener. "P_1" and "P_2" are the minimum and maximum pressures in the circuit and can be expressed as a percentage value of "P_t". The end user or the circuit designer shall determine the admissible values of "P_1" and "P_2" or in fact the admissible residual pulsation in the circuit**.** We don't recommend them to be less than +/- 2%, as the environmental temperature conditions will very probably modify the theoretical calculation.

MOUNTING SUGGESTIONS FOR MAXIMUM DAMPENER EFFICIENCY

As we have seen so far, taking into consideration the flow curves for the three types of pump (flow graph curves for the three different pump types), the single piston pump is the pump with the higher "**maximum instantaneous flow / mean flow**" ratio and also the one with highest liquid fluctuation inside the dampener in each cycle, "**δV**" if we consider the same piston diameter and stroke length for all three pumps. Therefore in the next example we will refer to the one piston single acting pump.

We can assume that for **99%** of industrial applications, if the recommendations that we detail below are followed, the dampener's efficiency will be guaranteed.

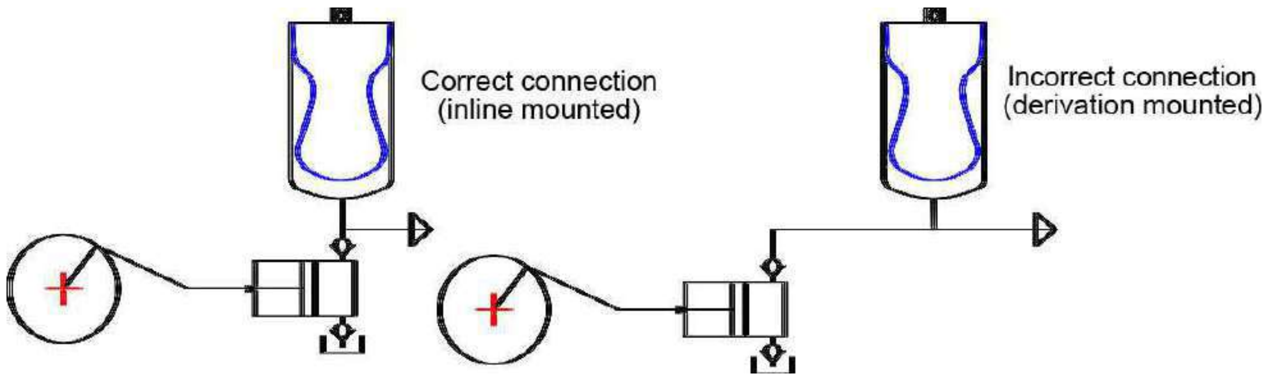

Figure 6. Scheme for the proper installation of a discharge pulsation damper in the pump outlet.

1.- The dampener must be mounted with its axis aligned with the axis of the pump outlet.

2.- The distance between the pump outlet port and the dampener port connection must be as short as possible.

3.- The pipe section between the pump and the dampener connection must be calculated for the pump maximum instantaneous flow.

4.- The remaining pipe section of the circuit must be calculated for the mean flow.

In the scheme of **Figure 7** we will see more clearly all the concepts we have exposed so far.

ω : Pipe section for the mean flow.
Ω : Small length of pipe section for the maximum instantaneous flow.
Q : Maximum instantaneous flow.
q : Mean flow.
L : Distance between pump and dampener, as short as possible.

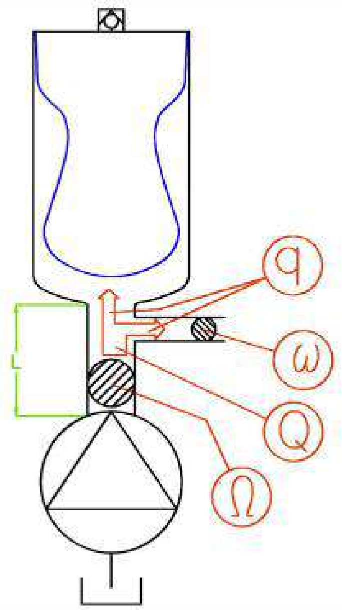

Figure 7. Scheme of the main parameters involved in the in-line assembly of a pulsation dampener.

To show the difference between in-line and derivation mounting of the pulsation dampener (see both installations in the scheme of **Figure 6**) into a circuit, and the higher efficiency of the in-line installation, we will remember some fluid mechanics principles:

The flow of a liquid inside a pipe follows different speed lines: In the centre of the pipe the velocity is maximum, while it becomes nearly to zero close to the pipe inner wall (see next drawing). If the mean liquid velocity increases, the difference between the dynamic pressure (the pressure measured in the liquid movement direction) and the static pressure (the pressure measured perpendicular to the liquid movement direction) also increases.

The scheme of **Figure 8** reflects this phenomenon: in-line mounting corresponds to the dynamic pressure reading; the derivation mounting corresponds to the static pressure reading. In fact, the alignment of the flow with the dampener port connection in the in-line mounting facilitates the entrance of the liquid in the damper due to its higher dynamic pressure. *(Note: We assume the fluid circulates in a laminar regime)*

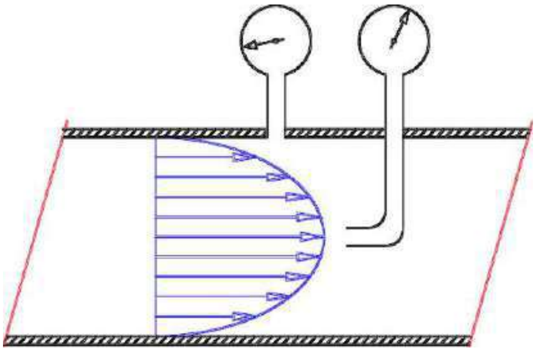

Figure 8. Scheme of the dynamic pressure in a pipe cross section

If the dampener is not only mounted in derivation but also far from the pump outlet, the efficiency of the dampener will be reduced a great deal. And if on top of this it is installed in a pipe section with a smaller diameter than the main circuit pipe, then the effect of the dampener will be negligible.

Regarding the used expression of in-line assembly, we want to empathize that ***HIDRACAR S.A.*** has developed a NEW in-line dampener (see on scheme of **Figure 9**) with a flexible rubber hose. In these new dampeners all the circuit flow pass through a flexible rubber tube which is expanded and compressed due to the effect of the variable flow.

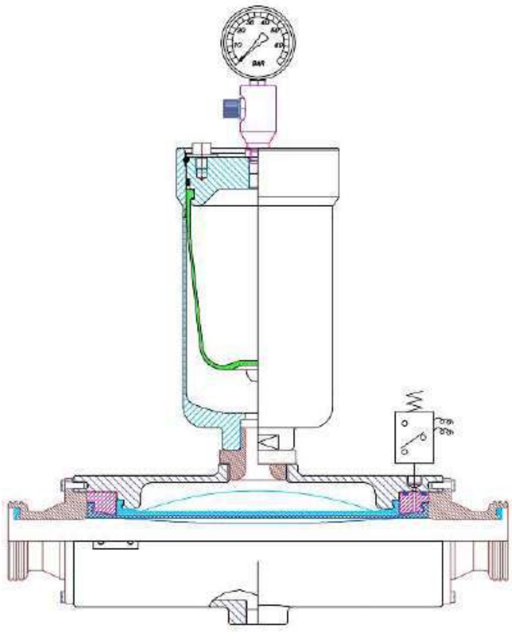

Figure 9. Scheme of the ***HIDRACAR S.A.*** new in-line dampener

WARNING¡: It is of utmost importance that the pulsation dampener hole passage must be as similar as possible than its connection port and the pipe section. Any reduction in the diameter of the hole passage, in dampeners installed in low pressure circuits, will greatly reduce the performance and efficiency of the dampener.

SOLUTION TO PROBLEMS OF PARTICULAR PULSATION DAMPENER APPLICATIONS

I) CIRCUITS THAT HAVE TO BE CLEANED PERIODICALLY AT THE END OF EACH PROCESS

Our NEW IN-LINE tube pulsation dampers (see on scheme of **Figure 9**), thanks to its special design without corners, can be cleaned in place using CIP processes (a cleaning agent is pumped in the circuit at certain pressure and temperature to clean all pipes and wetted elements in the circuit).

All the rest of pulsation dampers, though certainly some more than others, have internal corners which are hard to reach and difficult to clean or totally eliminate the residues of the pumped product with a CIP process.

The most reliable, low cost and efficient solution to this problem, in accordance with our longer than 45 years experience, is to use our quick dismantling system for bladder dampeners, to extract the bladder out of the dampener, and then clean separately both the bladder and the interior of the dampener body. In the case of applications where the charging gas pressure is lower than **10 bar** and compressed air can be used to fill the dampener, it is the most effective solution. ***HIDRACAR S.A.*** has designed a quick bladder dismantling system that makes unnecessary any additional tool.

If for whatever the reason, dismantling the bladder is not possible, we recommend the pressure of the cleaning liquid to be higher than the pumping pressure of the process product. That way the bladder or membrane will be more compressed, allowing a better access of the cleaning fluid in the internal corners in between the bladder/membrane and the dampener inner wall.

II) CIRCUITS WITH A VARIABLE WORKING PRESSURE

The problem arisen by the application of dampeners to this type of circuits has different solutions. But also in this case the experience has shown us that the best solution is, as always, the simplest one, or at least the solution requiring a lower implementation and maintenance cost and no extra energy.

Let's consider the following example: A circuit that must work at an initial pressure of **20 bar** and a final pressure of **200 bar**, with a **δV = 15 c.c.** and an admissible residual pulsation at **200 bar** of **+/- 5%** (*). The pump type is 1 piston single acting and its capacity per stroke is: **30 c.c.** To simplify the calculations we will consider that the gas volume variation takes place at a constant temperature (isothermal curve complying with **P x V = Constant**).

((*): At 20 bar the residual pulsation will be much lower because, as shown below, the dampener size is calculated for the maximum circuit pressure and therefore when the circuit is working at the minimum pressure - here, 20 bar - the gas inside the dampener will expand and consequently the residual pulsation will decrease from the +/- 5% initially admitted).

Since: $\mathbf{P_2 \times V_2 = P_0 \times V_0}$ $\quad$ $\mathbf{P_0 = 0.8 \times 20 = 16\ bar}$ $\quad$ $\mathbf{P_2 = 200 + 5\% = 210\ bar}$

$$\mathbf{P_2 / P_0 = V_0 / V_2 = 210 / 16 = 13.13} \quad (8)$$

We will calculate the volume of a hypothetical dampener for the maximum pressure of **210 bar**.

$$\mathbf{V_{0\,(\text{at 200 bar})} = (210 \times 15) / [0.8 \times 0.8 \times (210 - 190)] = 246.09\ c.c.} \quad \text{(from formula (7) in page 4)}$$

This volume is equivalent to "$\mathbf{V_2}$" from the equality (8), and consequently:

$$\mathbf{(210 / 16) = (V_0 / 246.09) = 13.125}$$

and

$$\mathbf{V_0 = 246.09 \times 13.125 = 3{,}229.98 c.c.}$$

This is in theory the total dampener volume necessary for this application; nevertheless, the ratio, $\mathbf{V_0}$ / $\mathbf{V_2}$ cannot be higher than **4** (In bladder type dampeners. The value will be different in other design types of dampener. Please, consult ***HIDRACAR S.A.*** technical department for further details on the particular) in order not to wrinkle the bladder excessively, what could tear it prematurely. In our example, we have a ratio $\mathbf{V_0}$ / $\mathbf{V_2}$ of **3,229.98 / 246.09 =13.125**, more than **3** times higher than the value of **4** that we have just recommended.

To avoid exceeding this ratio of **4:1**, a certain amount of liquid must be introduced inside the bladder together with the gas (usually the same liquid of the circuit or any other unable to react with either the bladder material or with the circuit liquid). Again, in our example this volume of liquid which has to be introduced into the bladder, **"V_L"** (see the scheme in **Figure 10**), is calculated:

$$(3{,}229.98 + V_L) / 246.09 + V_L) \leq 4 \quad \text{and operating:} \quad V_L = 748.54 \text{ c.c.}$$

The total dampener volume needed will be: **3,229.98 + 748.54 = 3,978.52c.c.**

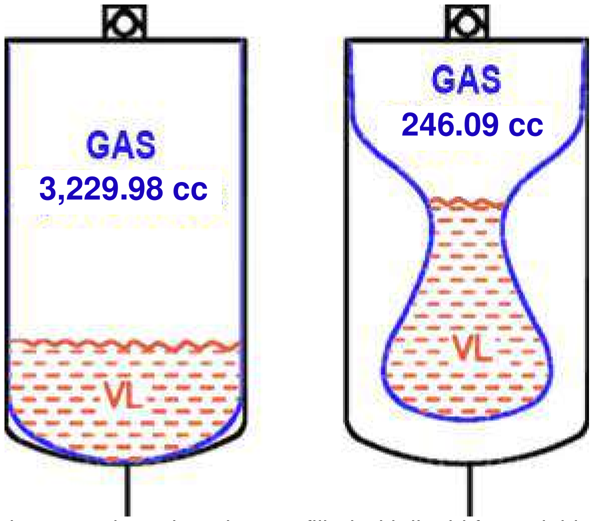

Figure 10. Scheme of the gas volume in a damper filled with liquid for variable pressure applications.

WHEN TO INSTALL A PULSATION DAMPENER AT THE SUCTION INLET OF A VOLUMETRIC PISTON PUMP TYPE OR SIMILAR DOSING PUMPS

As already said, volumetric pumps are used to dose with precision a constant volume of liquid. And therefore, the pump must get completely filled in every suction stroke piston displacement cycle.

When in the liquid inlet port of the pump the pressure can easily overcome the resistance of the suction valve spring that all pumps have (we can assume that it happens when the pressure at the inlet port to exceeds the resistance of the valve spring more than 3 bar) and the section of the suction pipe is about twice the discharge section of the pump, it won't be necessary to install a pulsation dampener at the suction inlet.

If the static pressure of the liquid at the pump inlet is low (below the already mentioned 3 bar) the suction pipe is long enough (longer than 3 to 5 metres from the suction liquid supply tank to the pump inlet) and also the liquid has a low vapour tension at the working temperature then a phenomenon called "**cavitation**" could take place.

When this anomaly takes place, the pump could suction a mix of liquid and its vapour. When this mixture (liquid and vapor) is compressed during the pump discharge cycle causes the condensation of the vapour because of the pump impulsion pressure. Consequently there is a reduction in the volume delivered in the outlet of the pump and witch performance loses efficiency. This effect, which can be detected by a soft explosion-like sound, reduces considerably the life of the pump, which also stops providing the required dosing.

In order to eliminate this problem it is necessary to prevent the pressure at the pump inlet port to be lower or close to the vapour tension of the liquid. And a condition for this, other than having enough pressure, is to avoid the suction pipe liquid column to be subjected to accelerations and decelerations caused by the operation of the pump.

It is precisely, **and exclusively to avoid these fluctuations in the liquid column** (accelerations and decelerations) at the suction pipe, **that a pulsation dampener is needed at the suction** of volumetric or dosing pumps in the above mentioned situations.

The pulsation dampener installed at the suction of the pump fulfills the same task as the one installed at the discharge: To keep the velocity of the liquid as constant as possible; and therefore, its pressure. If the low pressure of the liquid at the suction does not experiment any substantial drop, the possibility of reaching the vapour tension of the liquid will decrease and the main cause for the appearance of "**cavitation**" will be eliminated.

The pulsation dampener will not be able to avoid the "**cavitation**" phenomenon if all its determinants are present; and therefore it is convenient, when a risk exists, to install an auxiliary centrifugal or similar pump, or else, to raise the liquid supply tank or pressurize it and this way increase the pressure at the inlet port of the dosing pump.

If all these recommendations can't be applied, there is the possibility of installing the pulsation dampener to try to avoid the appearance of the "**cavitation**" effect. For this it is specially recommended that:

The size or volume of the dampener installed at the suction must be approximately twice as much as for the one installed at the discharge.

The size of the connection port of the dampener shall be as similar as possible as the diameter of the suction pipe.

The dampener must be installed as close as possible to the pump liquid inlet port, with the least possible pipe length in between.

If the suction pressure in the pump inlet is lower than atmospheric pressure (< 1 bar), then the gas volume inside the suction damper shall be reduced. When the dampener is delivered, the bladder must be compressed with hands with internal pressure of 1 bar.

*Currently **HIDRACAR S.A.** has designed a very effective in-line bladder damper (see in our BDOS catalogue ref. BLADDER IN LINE S.S.LOW PRESSURE PULSATION DAMPENERS), that can be considered the unique suction dampener with efficiencies nearly to 100%.*

For further detail explaining about the above exposed, please, contact ***HIDRACAR S.A.*** technical department.

We have written this paper in the modest hope of helping any people interested in these devices to understand the applications of the hydro-pneumatic accumulators used as pulsation dampeners. If it results useful to anyone, we will feel satisfied and rewarded for the time and effort spent in the making.

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