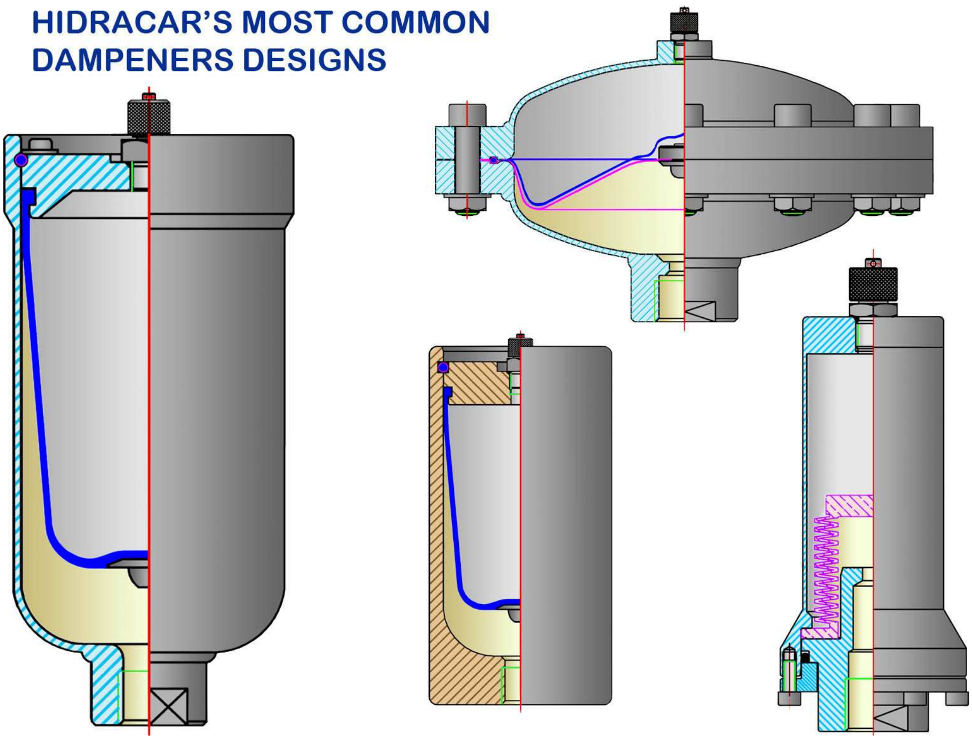
HIDRACAR S.A.

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 "P_0".

In all pulsation dampeners there is an 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 (**Nitrile, EPDM, FPM, 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.



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 characteristic 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).

When there is a pulsation dampener installed in the circuit, the volume supplied by the pump in every impulse or work cycle it generates 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, to start, 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: $$V_2 + \delta V = V_0$$

Every dampener has a constant derived from its size and its filling or charging gas pressure**:**

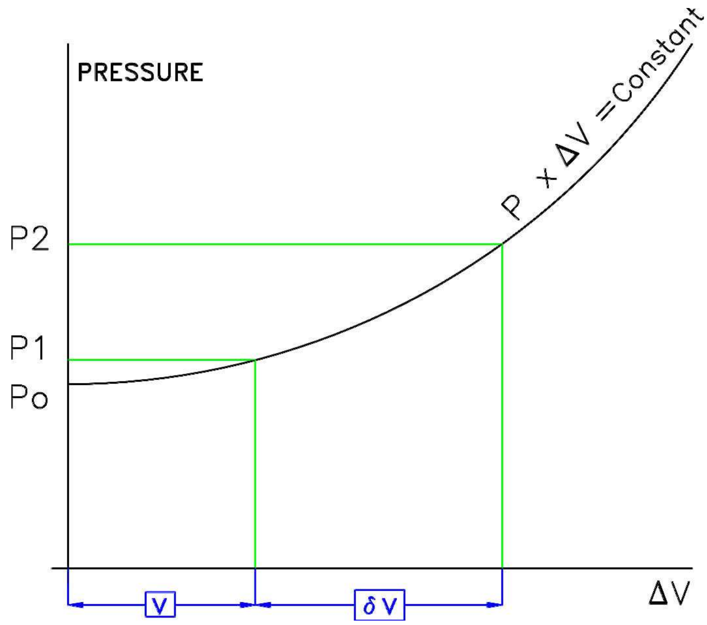
$$P_0 \times V_0 = \text{constant}$$

In working practice, it is not convenient for the dampeners to get totally emptied of the liquid, previously stored, in each cycle, 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:

$$V_2 + \delta V + v = V_0$$

where “**v**” is an unused volume of liquid inside the dampener. As a norm this volume is considered to be **10%** of the total dampener volume, as long as the temperature remains constant, and, therefore the former formula can be expressed as:

$$V_2 + \delta V + 0.1\, V_0 = V_0$$ and finally as: $$V_0 = (V_2 + \delta V) / 0.9$$

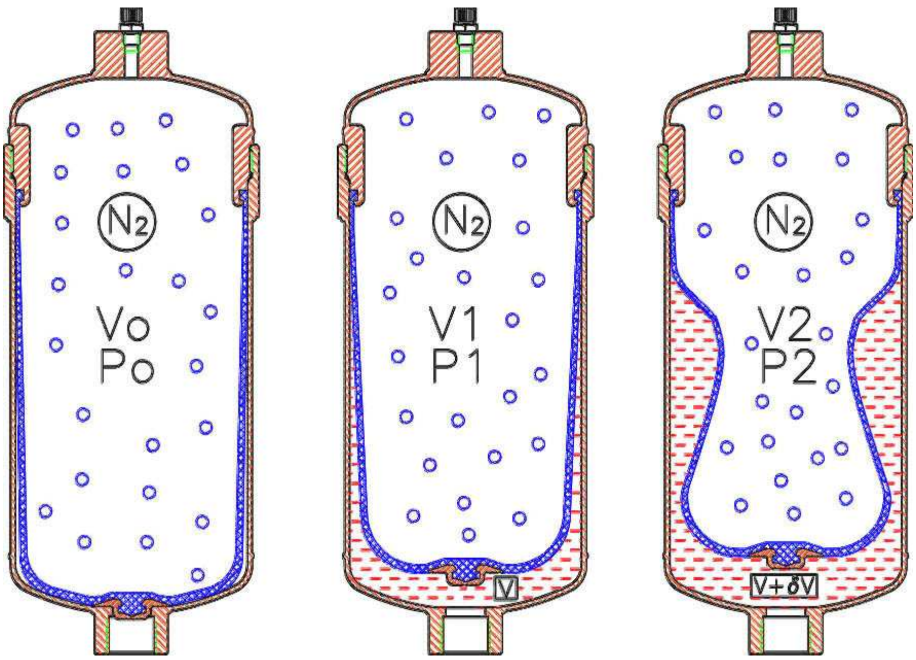


The graph above represents the curve (hyperbola) of gas compression inside the accumulator or pulsation dampener.

At the initial gas charge pressure value “**P_0**” there is no liquid inside the dampener and the gas fills the whole dampener interior. The curve cuts the ordinate axis in the point where corresponds to a zero value in the abscissa axis. This axis is where the amount or volume of liquid introduced into the dampener in each working cycle is represented.

The pressure “**P_1**” is the gas pressure when a volume “**v**” has been introduced into the dampener. The pressure “**P_2**” is the value reached by the gas when the additional volume “**δV**” is introduced into the dampener.

From this curve we can deduce that for a fixed dampener size if the value “**δV**” increases then the pressure value “**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 “**P_2**” will be lower.



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:

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

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

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 $v = 0.1 \times V_0$

we have: $V_1 = 0.9 \times V_0$ (2)

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

Finally, from (1) and (2) we obtain: $P_0 = 0.9 \times P_1$ (4)

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

$$\mathbf{P_0 \times V_0 = P_2 \times V_2 \;; \quad \underline{0.9\, P_1 \times V_0} = P_2 \times (V_1 - \delta V) = \underline{P_2\,(0.9\, V_0 - \delta V)}}$$

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

$$\mathbf{V_0 = \frac{P_2 \times \delta V}{0.9\,(P_2 - P_1)}} \qquad (5)$$

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**" **= 0.9 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) **$P_0 \times V_0 = P_1 \times V_1 = ... = P_n \times V_n$** 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:

$$\mathbf{(P_0 \times V_0)^{\gamma} = (P_1 \times V_1)^{\gamma} = ... = (P_n \times V_n)^{\gamma}}$$

where **γ** = specific heat ratio of the gas at constant pressure and volume, respectively. For the majority of gases, **γ = 1.41** This constant is also theoretical. In the practice the value that can be used is **γ = 1.25** But in order not to complicate the formula of dampener size calculation we will use a new constant (**0.8**) that will give the same result.

$$\mathbf{V_0 = \frac{P_2 \times \delta V}{0.8 \times 0.9 \times (P_2 - P_1)}} \qquad (6)$$

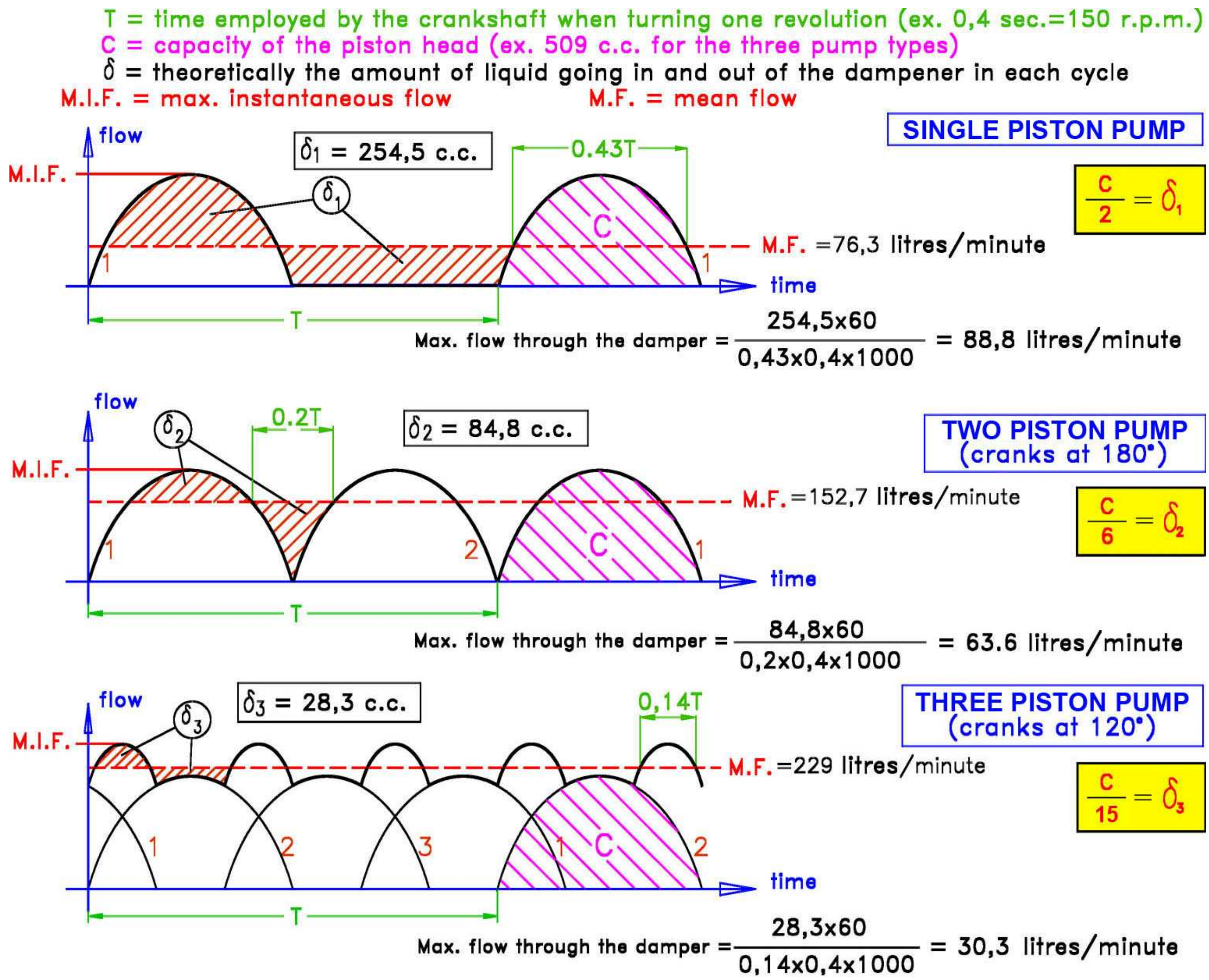
This formula can be used in practice for nearly all industrial applications. It will be very unlikely for the volumes given by this formula to fit any standard dampener volume size from a manufacturer. Except for very exigent applications we can recommend to use the manufacturer's standard closer lower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



These curves let us see how a pulsation dampener works: If we pay attention to the first curve, 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, from the point where it is mounted onwards, the maximum flow supplied to the circuit now practically becomes the mean flow of the pump, what 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 medium flow. In some cases this reduction of pipe diameter alone will compensate for the cost of the dampener; on top of the main advantage of stabilizing the circuit's pressure, with all of its obvious associated improvements (pressure in an hydraulic circuit is, basically, a function of the flow, and therefore if it varies, the pressure varies as well).

Carrying on with the first curve we can see that the task of the dampener is to store all the excess volume, over the mean flow line, of the total piston head during the piston impulse stroke; and then to return this volume "$\boldsymbol{\delta_1}$" 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 we can see that, as the number of pistons in a pump increase, the mean flow gets closer and closer to the maximum flow and the liquid volume "$\boldsymbol{\delta_1}$" 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).

To summarise: The more pistons a pump has, the lower the dampener size is required and also the smaller both the pipe section and the port connection between the dampener and the circuit can be (always assuming the pumps provide the same flow independently of the number of cylinders).

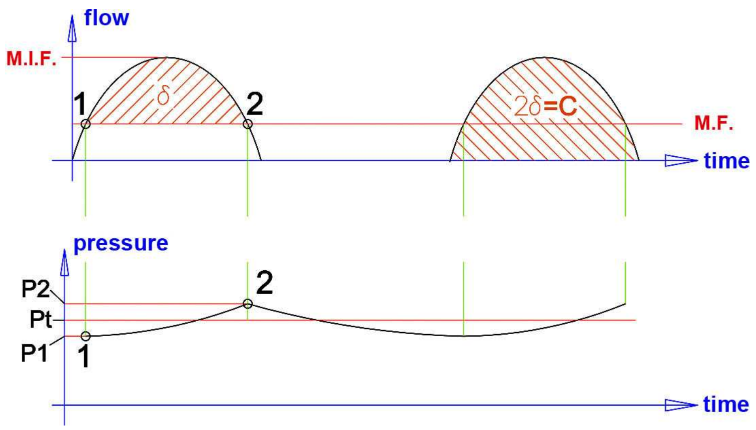
The relation between "**δV**" and the capacity per head "**C**" is

δv = C / 2 For a one piston pump
δv = C / 6 For a two piston pump
δv = C / 15 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 by 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 $\mathbf{P_t}$ will be defined by the technical designer of the circuit or by final customer requirements.

The following graphs will help to better understand the above exposed:



The lower curve from the above graphs 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; and the maximum value of the pressure curve (point **2**) must coincide with the second crossing point between them, the moment when all "**δV**" has been introduced inside the dampener.

Let's remember that the area comprised between the instantaneous flow curve and the abscissa axis (time) 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 $\mathbf{P_1}$, $\mathbf{P_t}$ and $\mathbf{P_2}$ in the pressure / time curve:

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 "$\mathbf{P_t}$".

When a circuit must be designed, one ought to take the mean flow and the opposing resistances to calculate the pressure "$\mathbf{P_t}$".

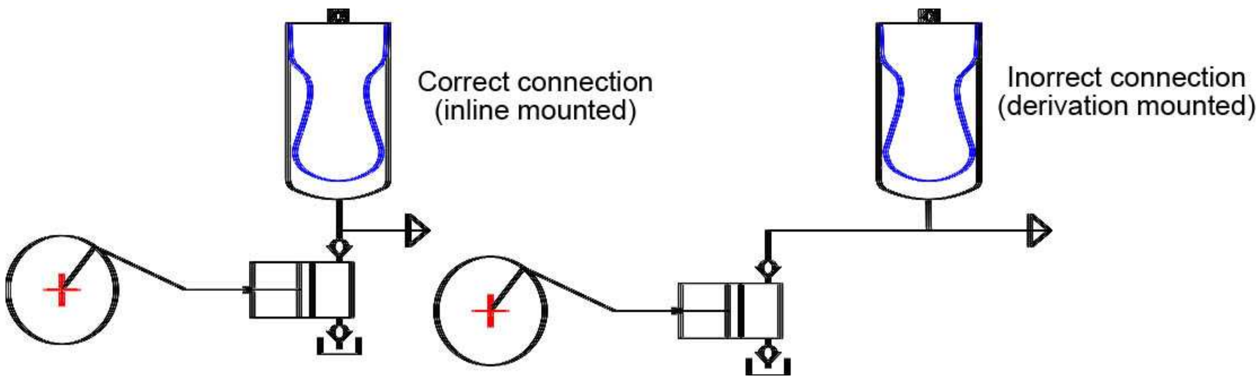
We see that on one side the dampener stabilizes the flow and for that also the pressure, but in fact the pressure goes from "$\mathbf{P_1}$" to "$\mathbf{P_2}$". The reason behind this is that the dampener has to stabilize the flow and for that it needs to compress and expand a volume of gas, and these pressure variations in +/- percentage of $\mathbf{P_t}$ are those that regulate the values accepted in the circuit.

We have already seen that this pressure fluctuation can be reduced down to very small values by increasing the volume of the dampener. "P_1" and "P_2" are the percentage values of "P_t", already commented which are pressure variations that the final customer must determine; though we do not 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, the single piston pump is the pump with the higher "**maximum instantaneous flow / mean flow**" ratio and also the one for which the liquid getting into and out of the dampener in each cycle, "**δV**", has a higher value; always considering both same piston diameter and pump stroke or displacement for all three pumps. Therefore in the next example we will refer to this type of pump.

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



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

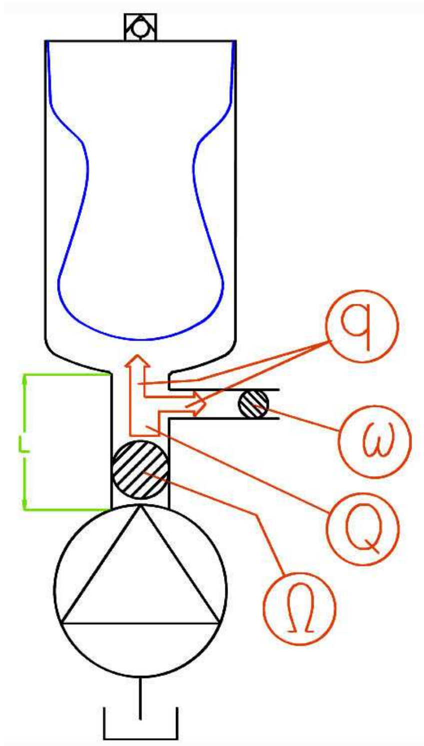
2.- The distance between the pump outlet port and the dampener port connection must be as short as possible.

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 next drawing we will see more clearly all the concepts we have exposed so far.

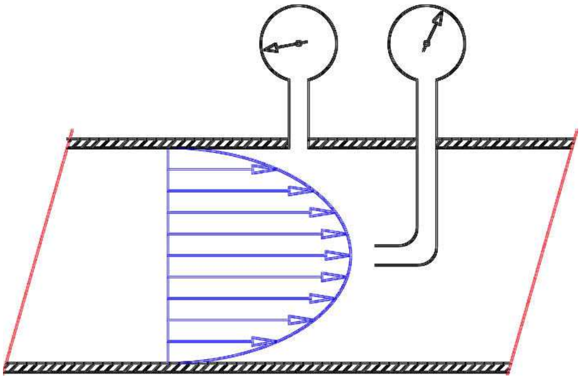
- **ω :** Pipe section for the mean flow.
- **Ω :** Small length of pipe section for the maximum instantaneous flow.
- **Q :** Maximum instantaneous flow.
- **q :** Mean flow.
- **L :** Distance between pump and dampener, as short as possible.



To show the difference between in-line and derivation mounting to a circuit, and the higher efficiency of the former, we will remember some fluid mechanics principles:

The flow of a liquid inside a pipe follow different speed lines: In the centre of the pipe the velocity is higher, while it becomes nearly 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 drawing below reflects this: in-line mounting corresponds to the dynamic pressure reading; the derivation mounting to the static pressure. *(Note: We assume the fluid circulates in a laminar regime)*



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.

WARNING¡: It is of utmost importance that the pulsation dampener admission duct is not narrower than its connection port. Any reduction in the diameter, when attached to 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

All pulsation dampeners, whatever its type - though certainly some more than others - have, because of its own design, hard to reach internal corners which are difficult to clean or to totally eliminate residues of the pumped product from.

The most reliable and efficient solution to this problem, in accordance with our longer than 35 years experience, is to use a quick dismantling system 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 out of the question, we recommend the pressure of the cleaning liquid to be higher than the pumping pressure of the process product; that way the internal corners between the bladder or membrane and the inner surface of the dampener will expand, allowing a better access to the cleaning fluid.

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 a maximum residual pulsation accepted at **200 bar** of **+/- 5%** (*****). The pump is simple effect single piston type and its capacity per stroke is: **30 c.c.** To simplify 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: $P_2 \times V_2 = P_0 \times V_0$ $P_0 = 0.9 \times 20 = 18$ **bar** $P_2 = 200 + 5\% = 210$ **bar**

$$P_2 / P_0 = V_0 / V_2 = 210 / 18 = 11.66 \quad (6)$$

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

$$V_{o\,(a\ 200\ BAR)} = (210 \times 15) / [0.8 \times 0.9 \times (210 - 190)] = 218.75 \text{ c.c.}$$ (from formula (6) in page 4)

This volume is equivalent to "V_2" from the equality (7), and consequently:

$$(210 / 18) = (V_0 / 218.75) = 11.66$$

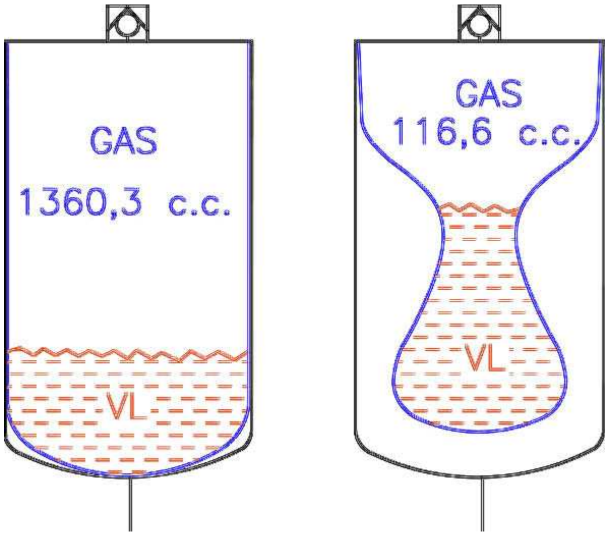
and $$V_0 = 218.75 \times 11.66 = 2{,}550.62 \text{ c.c.}$$

This is in theory the total dampener volume necessary for this application; nevertheless, the ratio, V_0 / V_2 ratio 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 V_0 / V_2 of **2,550.62 / 218.75 = 11.65**, nearly **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 figure), is calculated:

$$(2{,}550.62 + V_L) / 218.75 + V_L) \leq 4 \quad \text{and operating:} \quad V_L = 558.54 \text{ c.c.}$$

The total dampener volume needed will be: **2,550.62 + 558.54 = 3,109.16 c.c.**



WHEN TO INSTALL A PULSATION DAMPENER AT THE SUCTION INLET OF PISTON TYPE OR SIMILAR VOLUMETRIC OR 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 allow the pressure at the inlet port to exceed the resistance of the valve spring beyond 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 that, on being compressed during the pump discharge cycle, causes, because of the pump impulsion pressure, the condensation of the vapour and, in consequence, produces a reduction in volume. 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.

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 in order to prevent it 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 resort 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 must be at least identical to 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.

The gas charging or inflating pressure must be below atmospheric.

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.

7th rev., November 2013

Manuel Carcaré Gimeno
Technical Director & ***HIDRACAR S.A.*** founder