The design of pulsation dampeners for gas compressors requires the investigation of some dynamic phenomena related to the internal acoustics of pipelines. These phenomena may originate malfunctioning of the compressor itself, e.g., causing excessive mechanical vibrations (noise) and possible breaks.

The API 618 Standard gives the criteria to test such dynamical phenomena, suggesting precise limits to pressure pulsations at the coupling flange to the machine, and along the flowing line connected on the opposite side of the dampener (suction and discharge).

However, the criteria on the size of these acoustic filters given in this reference are not precise, and are related more to statistical criteria based on power, swept volume and rotation speed of the machine, rather than on the acoustics.

The sizing of the filters is left to the manufacturers, and the tests of acoustic efficiency are left to engineering companies that specialise in internal acoustics, such as S.A.T.E., which has developed specific numerical analysis software (ACUSCOMPTM and ACUSYS®). These tools are very accurate and efficient for ‘verification’ computations, but require iterative ‘trial and error’ procedures, starting from a tentative sizing of the acoustic filters. So far, they have not been conceived as a filter’s direct ‘design’ tools.

To overcome this functional limit, a new procedure has been developed as an automatic optimisation software integrated with the proprietary ACUSYS software. Given the characteristics of the fluid and the oscillating flow signal generated by the machine, it computes the dimensions (in terms of length and diameter of the dampener and choke pipe) that the filter should have in order to satisfy the API standard without being oversized.

The interesting advantage of this procedure, besides saving time and resources spent in the tentative search for a satisfying solution, is that it meets the need to order and build the dampeners before the design of the system is sufficiently completed, in order to make the overall verification of its acoustic behaviour according to the quoted standards. Past experience has shown that wrong sizing can have enormous consequences, in terms of extra costs and delays, on the compression package delivery and plant commissioning.

Design goals

Well designed compressor installations, as stipulated by the API 618 Standard recommendations, must follow two main concurrent goals, with regard to pressure pulsations limitation:

- Limit on compounded peak to peak pulsations at the compressor flange and plenum (Section 7.9.2.5.1).
- Limit on each individual harmonics of the pulsations propagating in the plant piping (Section 7.9.2.5.2.2).

The automatic procedure should ensure:

- To satisfy the API standard for what concerns the suggested pulsation limits, between the machine and dampener and vice versa.
- The dampener is not oversized.
Figure 1 gives a graphical explanation of the aforementioned limits:

- The pressure time signal at the compressor flange should not exceed the peak to peak limit of a few percent of the mean pressure along the line (7% or three times the compression ratio – whichever is lower).

- Concurrently, the amplitudes of the harmonics of the pressure signal on the line flange should not exceed the limit curve, decreasing with the frequency, in relation to the diameter of the line and its mean pressure.

Analogous criteria are suggested by the API 674 Standard for liquid reciprocating pumps.

Optimisation is conceived with the double goal to keep the pressure signals below but close to the limit, in order to dump the system just as necessary. In this way, oversizing is avoided due to the correlation between damping and the volume of the dampener. Moreover, the instrument allows the user to set a preferred safety coefficient as the desired margin between the limit and the pressure signal (in this study, this margin coefficient has been set to zero for clarity).

Before iterating the calculus automatically, it is convenient to identify, by first approximation, suitable starting values of the dampener size. This step is necessary for three reasons:

- It allows an idea of the order of magnitude of the system.
- It makes the subsequent optimising automatic procedure quicker.
- It maximises the success of the procedure, avoiding the automatic optimisation to converge towards ‘local minima’ far from practically feasible values.

The authors have presented the basis of the acoustic fluid dynamics phenomenon and the analytical formulae to be used for the first approximation sizing in part one of this article.4

**Numerical optimisation**

The optimisation routine implemented in MATLAB® uses ACUSYS to compute the pressure pulsations by running it iteratively for subsequent trial geometries, using the same known flow signal at each step. This signal may be easily obtained in a pre-study phase from the time law of the piston, or directly computed on the starting configuration to be optimised (base design) using another computation tool, such as the proprietary software ACUSCOP™. This has been carried out in the following case study.

It is worth highlighting that, in gas compressors, the flow signal would be influenced by the system response, in particular by the input impedance, as it influences the phase angle of the flow rate exchange between the cylinder and the plenum. However, in a pre-study phase, one can proceed and ignore these higher order effects for the optimisation because the subsequent, more accurate, simulations would verify all of the effects instead. For liquid pumps, the liquid compressibility is negligible as far as the cylinder exchange process is concerned; therefore, the input impedance variation does not appreciably influence the flow rate signal.

**Case study**

In this section, the results of the application of the routine to two different cases (based on real projects) are shown. In the first case (Case A), referring to a badly designed filter that does not comply with the limits, the routine finds the suitable filter dimensions, obtaining a new choke pipe, as well as a reduced dampener volume. In the second case (Case B), in which the base dampener was too small to satisfy the limits, the routine increases its volume to match them.

The schematisation of the system under consideration, which applies to a reciprocating gas compressor (Figure 1), was realised with ACUSYS and based on the requirements of the API 618 Standard. It considers the filtering dampener applied...
to a pipeline of infinite length, which simplifies the system during the pre-study phase when it is still not defined enough to study the acoustic behaviour in full detail.

The simplified system for the pre-study includes:

- Plenum of the compressor (T1), i.e. the chamber adjacent to the valves through which the fluid goes into and comes from the compressor cylinder (one plenum belongs to the suction and one to the discharge side).
- Choke pipe (T2), a tube that brings the fluid from the plenum to the dampener bottle.
- Dampener (T3), i.e. the bottle up to the outlet flange (T4).
- Infinite pipeline (T5).

The compressor in Case A brings the gas from the pressure of 3 barg (gauge) to 90 barg at the constant speed of 900 rpm. It interfaces with a filter whose characteristics are shown in Table 1 (base design). The bottle of this filter has a capacity of 161.8 dm³.

Figure 2 (base design) shows that, in this configuration, the pressure pulsations exceed the limits on both the flanges, in particular the sixth harmonic on the line side. The scope is then to bring the pressure pulsations below the limits while keeping it close enough to them in order to minimise the final volume of the dampener.

The routine gives the values reported in Table 3 (optimised design). The volume of the dampener has been reduced by 43.1%. This result shows that the dampener was not effective, despite its original oversize, perhaps due to the fact that it may have been designed for a different compressor speed. Concurrently, the choke pipe takes a new configuration, with its dimension having been increased.

Figure 2 (optimised design) shows the pressure pulsation behaviour of the optimised system. The limits are now fulfilled, with the second and sixth harmonics being just below the limit curve. The optimisation procedure met its goal, providing a completely new geometry suited for the velocity of the compressor under consideration, and minimum dampener dimension and cost.

The compressor in Case B brings the gas from the pressure of 3 barg to 35 barg at the constant speed of 1130 rpm. It interfaces with a filter whose dimensions are reported in Table 2 (base design). The bottle of this filter has a volume of 83.6 dm³.

Figure 3 (base design) shows that, in this configuration, the pressure pulsations exceed the limits on both the flanges, in particular the sixth harmonic on the line side. The results of the routine are reported in ‘optimised design’ in Table 4. In contrast to Case A, the volume of the dampener has been now increased by 35.3%, meaning that it was previously undersized. The choke pipe assumes a new configuration as well, having been increased in both length and diameter.

Figure 3 (optimised design) shows the pressure pulsation behaviour of the optimised system. The limits are now fulfilled on both sides of the filter. As can be observed, the optimiser achieved its goal, even in this case, giving a new geometry that dampens pressure pulsations by the same criterion followed for Case A, yet the direction of sizing correction was opposite.

### Table 1. Main data of the reference design dampener of Case A (third stage discharge of a high pressure compressor)

<table>
<thead>
<tr>
<th>Description</th>
<th>Symbol</th>
<th>Base design: Case A</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plenum volume</td>
<td>V_p</td>
<td>11.0</td>
<td>dm³</td>
</tr>
<tr>
<td>Number of delivering cylinders</td>
<td>z</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>Line diameter (beyond the dampener)</td>
<td>D_i</td>
<td>66.6</td>
<td>mm</td>
</tr>
<tr>
<td>Average line pressure</td>
<td>p_i</td>
<td>90</td>
<td>bar</td>
</tr>
<tr>
<td>Speed of sound at the nominal conditions</td>
<td>c</td>
<td>320.4</td>
<td>m/s</td>
</tr>
<tr>
<td>Dampener bottle diameter</td>
<td>D_b</td>
<td>611</td>
<td>mm</td>
</tr>
<tr>
<td>Dampener bottle length (equivalent)</td>
<td>L_b</td>
<td>552</td>
<td>mm</td>
</tr>
<tr>
<td>Dampener bottle volume</td>
<td>V_b</td>
<td>161.8</td>
<td>dm³</td>
</tr>
<tr>
<td>Choke pipe diameter</td>
<td>D_c</td>
<td>101.6</td>
<td>mm</td>
</tr>
<tr>
<td>Choke pipe length</td>
<td>L_c</td>
<td>236</td>
<td>mm</td>
</tr>
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</table>

### Table 2. Main data of the reference design dampener of Case B (third stage discharge of a high pressure compressor)

<table>
<thead>
<tr>
<th>Description</th>
<th>Symbol</th>
<th>Base design: Case B</th>
<th>Unit</th>
</tr>
</thead>
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<tr>
<td>Plenum volume</td>
<td>V_p</td>
<td>11.0</td>
<td>dm³</td>
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<tr>
<td>Number of delivering cylinders</td>
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<tr>
<td>Line diameter (beyond the dampener)</td>
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<tr>
<td>Average line pressure</td>
<td>p_i</td>
<td>35</td>
<td>bar</td>
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<tr>
<td>Speed of sound at the nominal conditions</td>
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<td>320.4</td>
<td>m/s</td>
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<tr>
<td>Dampener bottle diameter</td>
<td>D_b</td>
<td>611</td>
<td>mm</td>
</tr>
<tr>
<td>Dampener bottle length (equivalent)</td>
<td>L_b</td>
<td>285</td>
<td>mm</td>
</tr>
<tr>
<td>Dampener bottle volume</td>
<td>V_b</td>
<td>83.6</td>
<td>dm³</td>
</tr>
<tr>
<td>Choke pipe diameter</td>
<td>D_c</td>
<td>101.6</td>
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<tr>
<td>Choke pipe length</td>
<td>L_c</td>
<td>236</td>
<td>mm</td>
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</table>

### Conclusion

From the case study discussed above, it is evident that the design of a filter is sensitive to the choice of the dimensions of the bottle and of the choke pipe, and how much the damping characteristic of a filter depends on the compressor spinning velocity – the signal harmonics generated being directly proportional to the latter. This aspect makes the design of a filtering system even more complicated for variable speed compressors.

It is worth stressing that the optimisation procedure shown herein for the so called ‘pre-study’ phase, with infinite pipeline, may also be used in more advanced
stages of analysis, allowing the inclusion of the whole actual piping system connected on the line side of the dampener. This would not modify the procedure, yet computing time would increase as a function of pipe network complexity.

The study and the tools developed in this project provide an effective and efficient automated tool for dampener design and the analysis of remedies in acoustic pulsation studies, for either a gas compressor or liquid pumps systems, in which the pressure pulsations do not match the limiting criteria suggested by the API 618 and 674 Standards.

### References


<table>
<thead>
<tr>
<th>Table 3</th>
<th>Case A, from 3 - 90 barg, 900 rpm: (left) the filter dimensions for the base design; (right) the filter dimensions for the optimised design; a 43.1% volume reduction of the dampener can be observed</th>
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<tbody>
<tr>
<td>Case A</td>
<td>Base design</td>
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<td>Choke pipe diameter (mm)</td>
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<td>Choke pipe length (mm)</td>
<td>236</td>
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<tr>
<td>Bottle diameter (mm)</td>
<td>611</td>
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<tr>
<td>Bottle length (mm)</td>
<td>552</td>
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<tr>
<td>Bottle volume (dm³)</td>
<td>161.8</td>
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<th>Table 4</th>
<th>Case B, from 3 - 35 barg, 1130 rpm: (left) the filter dimensions for the base design; (right) the filter dimensions for the optimised design; a 35.3% volume reduction of the dampener can be observed</th>
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<td>Case B</td>
<td>Base design</td>
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