EFRC Guidelines

EUROPEAN FORUM
for RECIPROCATING
COMPRESSORS

GUIDELINES ON HOW TO AVOID LIQUID PROBLEMS IN RECIPROCATING COMPRESSOR SYSTEMS

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Content

Foreword

The EFRC is the European Forum for Reciprocating Compressors has been founded in 1999 by Neuman & Esser, Leobersdorfer Maschinenfabrik, Hoerbiger Ventilwerke, TNO, TU Dresden, Thomassen Compression Systems, Wärtsila Compression Systems and Burckhardt Compression.
The target of the EFRC is to serve as a platform to facilitate exchange of information between vendors, operators and scientists working in the field of reciprocating compressors.
This is achieved by knowledge transfer (conferences, internet, student workshops, training and seminars), standardization work (e.g. API 618, ISO 13707, and ISO 10816) and by joint pre-competitive research projects, aiming at improving the performance and the image of the reciprocating compressor.
In the R&D projects the forces are combined of all interested parties to solve or investigate problems which are beyond the scope of a single player. The basic research and pre-competitive research projects are carried out at research institutes or universities. In this way the R&D group of the EFRC will serve as the scientific arm of the reciprocating compressor community.
The R&D group is open to all EFRC members and the annual budget is funded by participating members. The results are owned by the EFRC and the research results are disclosed to EFRC research group members only.

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1 Scope

Reciprocating compressors are positive displacement machines which are unable to handle substantial amounts of liquids. Significant amounts of liquids entering reciprocating compressors are known to cause reliability problems or even catastrophic failures such as:

- Loss of primary containment due to incompressibility of liquids.
- Damaged compressor parts. A slug as a result of liquid carry-over from a process or from interstage coolers is particularly hard on compressor valves which can damage compressor parts such as valves (especially at the suction side), piston rings piston, piston rod, crosshead, and bearings, due to increased forces caused by increased pressure fluctuations (exceeding design limits). The valves can also be damaged by sticktion and pressure spikes.
- Reduction in reliability of the compressor due to unpredictable wear of cylinder wearing parts, dilution of the lubricating oil film or washing away PTFE transfer films. Insufficient lubricating will result in accelerated wear of the rider and piston rings and an increase in piston rod drop in such a way that piston will touch the cylinder wall or liner.

These EFRC guidelines establish procedures and guidelines for the design and operation of a reciprocating compressor system with respect to liquid handling. The goal of these guidelines is to ensure the reliable operation of reciprocating compressor systems by avoiding liquid related problems. In general, these guidelines are not intended to be the only reference for designing and operation of reciprocating compressor systems. Using the API Standard 618 [1] together with these guidelines is indispensable. In other words, these guidelines can be used as a complementary document to the available guidelines and engineering practices. These guidelines are intended for OEM's, end-users and engineering companies.
The presence of liquid in reciprocating compressors can lead to serious damages and or unscheduled shut-down of compressors. Liquid related problem however mainly originates from other components in systems rather than the compressor cylinder. Thus, in this document, the compressor system is divided into different components. Each component in the system is being investigated to decrease the chance of liquid related problems. Engineering rules are given to increase the awareness on the design and operation of different compressor system's components.

The system components which are covered by these guidelines are:

- Separators and their auxiliaries, such as demisters, level control, etc.
- Pulsation dampers.
- Upstream, downstream and interstage piping.
- Drains.
- Miscellaneous.

In the section of separators, engineering rules on the pre-selection, design check and operation of separators are given. For the design of separators, this document needs to be used in conjunction with the more detailed guidelines which are specifically developed for separators. The rules on how to check the separator vendor's design are valid for pressures up to 105 barg. For pressure above 105 barg, the capacity of separators will be further de-rated and the design procedure should be done in consultation with a separator expert.

It is a common practice that the layout (system (piping, skids, etc.) is worked out prior to detailing the scrubber/ separator sizing, resulting in constraints for separator design to minimize liquid carry-over. For that reason it is important that the design of the separator shall be done in a very early stage of the project. It is also very important in this that the different disciplines for plant layout, mechanical and process work together for designing and fabrication of the most robust system that is also cost efficient.

Even though the scope of the document does not include any vibration related issues, engineering guidelines are given for the separator design to avoid vibration problems. Fatigue failures due to vibration are a significant cause of downtime and safety concerns and shall be avoided. If the design of the separator for "liquid removal" does contradict the design to avoid vibration problems, a decision shall be made by the vendor and purchaser about the preferred design.
In addition to the component description of the system, guidelines on the operation of reciprocating compressor systems at different operational stages are given.

Examples of reciprocating compressor systems covered in these guidelines are:

- Horizontal, vertical, V-, W- and L-type compressor systems.
- Constant and variable speed compressors.
- Compressors driven by electric motors, gas and diesel engines, steam turbines, with or without a gearbox, flexible or rigid coupling.
- Dry running and lubricated reciprocating compressors.
- Compressor systems for all types of gases.
- Diaphragm compressors.
- Labyrinth compressors.

For hyper compressor systems which consist of one or more booster compressors and a hyper compressor, the guidelines are not applicable for the hyper compressor part.

2 Terms and Definitions

List of definitions and nomenclatures used in these guidelines.

2.1 General

2.1.1 Compressor system

Machinery system comprised of compressor (crankcase, crosshead guide, and cylinders), pulsation dampers, separators, coolers, piping and heat exchangers. The boundary at the suction side is the common feed header or a large volume upstream the suction of the first stage separator. The boundary at the final discharge side piping is the piping downstream the discharge pulsation damper. If a cooler and separator are installed at the final discharge, the piping downstream the final discharge separator will be the boundary.

2.1.2 Purchaser

Agency that issues the order and specification to the vendor.

2.1.3 Vendor

Manufacturer (OEM) or manufacturer’s agent that supplies the equipment.

2.1.4 Dry gases

Process gas with dew point lower than -40 ˚C.

2.1.5 Actual (operating) volumetric flow rate

The actual (operating) gas volumetric flow rate is being defined at the given operating pressure and temperature.

2.1.6 Heat tracing

Heat tracing is used to keep the piping temperature high enough and avoid process gas condensation on the inner wall of the piping. In this document the heat tracing is referred to as an electrical heat tracing, not vapour (steam) heat tracing.

2.2 Separator

2.2.1 Separator

Separator vessel including internals and auxiliaries e.g. inlet devices and demister units etc., with the purpose of separating the liquids from the gas stream.

2.2.2 Separator vessel

Separator volume excluding internals and auxiliaries.

2.2.3 Relative humidity

The relative humidity is the ratio of partial pressure of water vapour and the saturated vapour pressure of water at a given temperature is given in Equation 2-1:

$$x = \frac{P_{Partial,H2O}}{P_{sat,H2O}}.100 \quad [\%] \qquad \textbf{2-1}$$

In which:

$P_{partial,H2O}$ Partial pressure of water vapour [*bara*]
$P_{sat,H2O}$ Saturated vapour pressure of water [*bara*]

Note that Equation 2-1 is only valid for water. Other condensates will also influence the relative humidity. For more details about the calculation of relative humidity for hydrocarbons, use process simulations software or chemical engineering handbooks.

2.2.4 Souders-Brown velocity:

It is known as gas phase (vapour) load factor, which is considered as an effective gas velocity for the purpose of expressing the separation capacity limit, corrected by the relative density between phases.

2.2.5 Liquid removal efficiency

The liquid removal efficiency of a Gas/Liquid separator unit is defined as the ratio of the separated liquid (volumetric) flow rate and the feed liquid (volumetric) flow rate (Equation 2-2):

$$\eta_{sep} = \frac{Q_{L,sep}}{Q_{L,feed}} [\%] \qquad \textbf{2-2}$$

In which:

$Q_{L,sep}$ Volumetric flow rate of the separated liquid [m^3/s]
$Q_{L,feed}$ Volumetric flow rate of the liquid in the feed [m^3/s]

2.2.6 Turndown ratio

Turndown ratio is the ratio of the maximum and minimum flow rate into the separator.

2.2.7 Low level trip

An alarm level indicating that the liquid level in the separator vessel is too low for operation and there is a chance that gas being by-passed through the liquid drains.

2.2.8 High level trip

An alarm level indicating that the liquid level in the separator vessel is above the allowed limit and the compressor needs to be switched-off if it cannot be corrected.

2.2.9 Cut-off diameter d_{50} (d_{99})

Droplet size of which 50% (99%) of the droplets present at the inlet of the separator are separated.

2.2.10 Liquid hold-up in pipe systems

Liquid hold-up is defined as the ratio of liquid volume to the total volume in the pipe system (Equation 2-3).

$$\alpha_L = \frac{A_L}{A_{total}} \ [-] \qquad \textbf{2-3}$$

In which:

A_L	Area of the liquid in a pipe cross-section [m^2]
A_{total}	Total area of the pipe cross-section ($A_{total} = A_G + A_L$) [m^2]

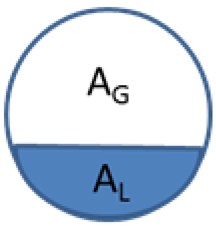


Figure 2-1. Definition of liquid hold-up in pipe systems

2.2.11 No-slip hold-up in pipe systems (λ)

Holdup if there is no slip between the gas and the liquid is called no-slip holdup. In other words, it is the ratio of the amount of liquid to the total volume of the pipe system, when interfacial slip between phases is neglected (Equation 2-4).

$$\lambda_L = \frac{U_{SL}}{U_{SL} + U_{SG}} \ [-] \qquad \textbf{2-4}$$

In which:

λ_L	No-slip liquid hold-up [-]
U_{SL}	Superficial liquid velocity [m/s]
U_{SG}	Superficial gas velocity [m/s]

2.3 Pulsation dampers

2.3.1 Pulsation damper or Pulsation Suppression Device (PSD)

Empty vessel or a vessel including baffle plates, baffle choke tubes, half-pipes, etc., installed for the purpose of the reduction of pressure pulsations and pulsation-induced reaction forces

2.4 Piping

2.4.1 Block valve

A valve which is used to block the flow in one or more direction.

2.4.2 Pocket

Location in the compressor system where condensates or liquids can be accumulated

2.5 Operation

2.5.1 Barring device

This is a device that slowly turns the compressor for maintenance purposes and before start-up after long stoppages of reciprocating compressors.

3 Liquid formation and accumulation

Before giving engineering rules on how to avoid or prevent liquid problems in reciprocating compressor systems, it is important to give some background information on how the liquid will be formed and will flow into the compressor system. The purpose of this section is therefore to give an overview of different mechanisms of liquid formation and accumulation.

3.1 Mechanisms of liquid formation and accumulation

Liquid problems will occur at different situations and lead to different kind of failures and damages. However, the formation and accumulation of liquid in compressor systems can be summarized into three mechanisms: inefficient liquid/gas separation, condensation and excessive compressor cylinder lubrication.

A well-designed separator unit upstream (at suction side) the compressor should capture the liquid content of the gas. In case of an inadequate design or malfunction of a separator, liquid carry over will occur which can cause severe and dangerous situations.

The condensation of a gas with vapour content (e.g. wet gas) is another mechanism for liquid formation. This means that the parameters which can lead to condensation need to be controlled. For instance, if the process gas is saturated and flows into a cold suction line (pipe wall temperature lower than the process gas dew point), then the chance of liquid formation will increase and in this case the temperature of the piping needs to be controlled.

Excessive and uncontrolled lubrication in the compressor cylinder can also lead to liquid failures. This problem can be more crucial during the start-up of the compressor system in which pre-lubrication is being done.

3.2 Process gases with higher chance of liquid formation

Dry gases will not lead to liquid problems. However, in practice gases nearly always have some liquid content and margin to the dew point line is important. Hydrocarbons, saturated and wet gases, natural gases and flare gases have the highest percentage of the reported liquid problems.

3.3 Allowable amount of liquid

Liquid can flow in compressor systems in different forms: droplet, slug, film, etc. More explanation on different liquid flow regimes are given in Annex B. The liquid slug is the most severe and problematic flow regime. If it enters the compressor cylinder, it can result in forces in excess of 5-10 times the normal force for a single-phase gas flow and will lead to system failures.

In case of a liquid film, the impact will be much less in comparison with a liquid slug but it is still sufficient to cause damage. Increased suction and discharge valve losses as a result of the increased density of the liquid-gas mixture will result in higher differential pressure across the piston and compressor valves.

The more the liquid volume inside the compressor cylinder approaches the clearance volume of the compressor cylinder, the more significant the load and potential damage will occur.

Even a small amount of liquid in form of droplets (mist) could initiate cracks on parts, such as valves or piston, which ultimately can lead to fatigue failures.

Most of the compressor valve failures are initiated by liquid presence which cause high loads (high liquid volume fraction) on the valve or will cause sticking and erosion problems (low liquid volume fraction). Valve sticking problems will delay valve opening and consequently will increase the dynamic loads.

According to API 618 paragraph 7.8.2.3 of the 5th edition; 99% of droplets above 10 μm shall be removed by the liquid separation device. In practice these values are not achievable, due to the fact that the cut-off diameter of separators is very dependent on the operating pressure and density difference between the gas and liquid phase. Field experience has shown that with a well-designed separator with a proper inlet device and with a large enough operating envelope, liquid related problems can be mitigated.

More explanation about the cut-off diameters of the separators is given in Annex H.

4 Separators

4.1 Introduction

The main function of a separator is to separate the liquid content from the gas or vapour flow. For that reason, the separator is one of the most critical components in avoiding liquid problems. Accurate design, operation and monitoring of a separator are for that reason of significant importance but can be very complex. An adequate separator should be designed in good cooperation between the separator and compressor vendor, the process and piping engineer and the operator. It should also be clearly recognized that no separator will separate 100% of the liquid content (liquid removal efficiency of 100%) and consequently it needs to be acknowledged that carry-over of liquids is always present.
Experience has shown that separators with the highest efficiency are more reliable w.r.t. to liquid handling and will lead to systems with a higher safety and reliability.
The goal should be therefore to choose a separator with the highest possible efficiency for all process (temperature, pressure, molecular weight, density etc.) and flow conditions (full load, part load, compressor speed range, etc.) as indicated on the compressor data sheets, including transient conditions such as start-up and shut down. However, there might be situations where systems can operate safe and reliable with a separator with a lower efficiency, e.g. systems where only small amount of liquids are expected to occur in systems designed in accordance with the guidelines in order to prevent liquid accumulation. Decisions on the acceptance of lower efficiency (<95% for all specified conditions by the purchaser) shall preferably be based on proven designs and relevant experience in similar applications.
<u>The separator efficiency and final separator design shall be agreed upon between the vendor and purchaser.</u> The type of separator can be determined with the overview of Appendix A together with the distinctive characteristics as summarised in Table 1.

Separators shall be located upstream of the compressor and after every cooler.
It is not the intention of these guidelines to give all detailed design rules for all types of separators. Rather than that, the following main topics will be addressed in this chapter:

- Engineering guidelines for pre-selection of separators and demisters.
- Rules on how to check the vendor’s design of a separator and its auxiliaries.
- Additional engineering rules for the design and operation of separators.

Much different type of separators and demisters are available but in these guidelines only the most suitable and commonly applied types for reciprocating compressor systems will be discussed in detail. Different sections and components of separators are generally defined and given in Annex F. Note that for more detailed design of the separator, internal company procedures, international guidelines, and engineering practices have to be used or a separator expert should be consulted.
It should be noted that the rules on how to check the separator vendor’s design are valid for pressures up to 105 barg. For pressure above 105 barg, the capacity of separators will be further de-rated and the design procedure should be done in consultation with a separator expert.

4.2 Engineering guidelines for pre-selection of separators and demisters

For the determination of the required separation efficiency there are a number of considerations:

- Protection of the compressor (e.g. piston, compressor valves, bearings etc.) and its auxiliaries (e.g. process valves, instrumentation etc.).

- Requirements from the process on intermediate- and final separation (requirements of dry gases, oil-free compression etc.).

The required separation efficiency is depending on various aspects:
- Type of process (e.g. Ethylene compression, O2, natural gas storage, urea production etc.).
- Expected liquid flow regimes, i.e. droplets or slugs.
- Type of contamination, e.g. fixed particles (e.g. from catalyst).
- Process conditions (e.g. pressure, temperature, density).
- Gas composition (e.g. saturation level, H2O content, presence of heavy HC components etc.)
- Criticality of the compressor in the process (i.e. acceptance of risk of failure and damage, maintenance requirements, etc.).
- Compressor design (e.g. clearance volume, valve type etc.).

An overview of different separators and demisters with their characteristics is given in Annex A. The efficiency of separators can roughly be classified in two main categories:
1. Separator without internals,
    a. e.g. knock-out drums, with a separation efficiency of approximately 90%;
    b. e.g. cyclones, with a separation efficiency of approximately 95%;
2. Separators with internals, e.g. axial cyclone packs, wire mesh and vanes. The separation efficiency depends on the type of internals used and can go up to > 95%, (e.g. vane type) and up to 98% and above (e.g. wire mesh, axial cyclones).

The cut-off droplet diameter is dependent on the operating condition and density ratios (Annex H). Several system parameters are required for choosing an adequate separator. Finally, five parameters were chosen to be the most critical with respect to the overall performance and efficiency of separators in reciprocating compressor systems. These parameters are: (1) Slug Handling, (2) Droplet Handling, (3) Turn-Down Ratio and (4) Pressure Drop and Fouling Tolerance (5). A relative comparison between several types of separators for these four parameters is given in Table 1.

Note that knock-out drums without internals (demisters) are not capable of separating small size droplets and are not recommended. The combination of knock-out drums with a wire mesh coalescer has to be used to increase the efficiency and operation envelope of the separator. Additionally, using inlet devices can increase the slug handling capabilities of separators. The desired pressure drop should be as low as possible. The recommended maximum pressure drop is given in section 4.4.3. Special consideration for selection may be required in case of sticky deposits such as waxes are present.

Table 1. Comparison between distinctive characteristics of Gas/Liquid separators (The order from the top is from the simplest to the most complex configuration)

Type of separators	Slug handling***	Droplet handling	Turndown ratio****	Pressure drop****	Fouling tolerance
Vertical Knockout-drums	+	-	++	- -	++
Horizontal knockout-drums	++	-	++	- -	++
Cyclone with tangential inlet	+	+	0	+	++
Cyclone with straight line and swirler	++	+	0	+	0
Horizontal vane-type*	+	+	0	-	-/--*****
Vertical vane-type (in-line)*	- -	0	0	-	-/--*****
Vertical vane-type (vertical)*	+	+	0	-	-/--*****
Horizontal wire mesh*	++	++	-	-	-/--*****
Vertical wire mesh*	0	++	-	-	-/--*****
Cyclone packs with wire mesh*	++	++	+	0	0/-
Cyclone packs with vane pack*	++	++	0	0	0/-
Coalescers	--	++	++**	++	--

Explanation of used symbols:
--: very low, -: low, 0: moderate, +: high, ++: very high
* These demisters are combined with a knock-out drum
** Limited by droplet size and entrainment onset
*** Slug handling of each device can be improved with a suitable inlet device
**** See Annex A for values of turn down ratio and approximated pressure drop
***** Depends on the type of wire mesh or vane pack

4.3 Rules on how to check the vendor's design of a separator and its auxiliaries

4.3.1 Inner diameter of separator (size)

The minimum inner diameter of the separator vessel should be calculated by Equation 4-1.

$$ID_{\min} = \sqrt{\frac{4Q_{gas,\max,operation}}{\pi K}\sqrt{\frac{\rho_G}{\rho_L - \rho_G}}}\ [m] \qquad \textbf{4-1}$$

In which

ID_{min}	inner diameter [m]
$Q_{gas,max,\ operation}$	Maximum operating gas volumetric flow rate at the inlet nozzle of the separator [m^3/s]*
ρ_G	Gas density [kg/m^3]
ρ_L	Liquid density [kg/m^3]

K Souders-Brown [7] velocity [*m/s*]

*This is the flow without consideration flow pulsations, see also example of Annex I

Souders-Brown velocity is defined as follows:

$$K = \frac{u_G}{\sqrt{\frac{\rho_L - \rho_G}{\rho_G}}} \; [m/s] \qquad \textbf{4-2}$$

In which

u_G	Gas (vapour) actual velocity [*m/s*]
ρ_G	Gas density [kg/m^3]
ρ_L	Liquid density [kg/m^3]
K	Souders-Brown [7] velocity [*m/s*]

For calculating the parameters in the Souders-Brown equation, procedures are explained in the next sections.

4.3.1.1 Determining maximum actual gas volumetric flow rate ($Q_{gas,max}$)

The maximum actual gas volumetric flow rate at the inlet nozzle of the separator should be calculated based on the maximum gas velocity (Figure 4-1) [5] which is as follows (Equation 4-3):

$$U_{G,\max} = \overline{U}_G + U'_G \; [m/s] \qquad \textbf{4-3}$$

In which

$\overline{U}_G$	Average gas velocity based on the average actual volumetric flow rate [*m/s*]
U'_G	Fluctuating gas velocity [*m/s*]

Note that $U_{G,max}$ is based on the inner diameter of the inlet pipe.

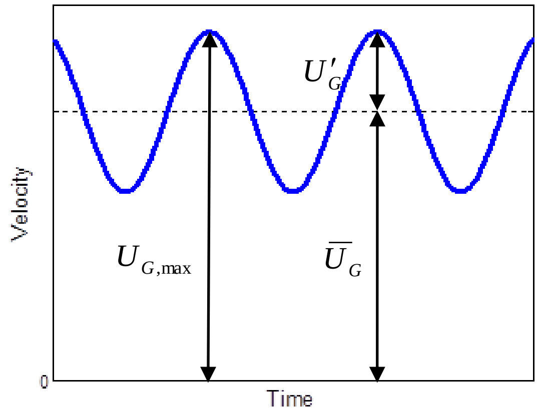


Figure 4-1. Definition of mean, fluctuating and maximum velocity

The gas fluctuating velocity is the velocity which considers the flow pulsations. Typically, this data is unavailable during the initial design phase but an estimated design value for the fluctuating gas velocity U'_G can be calculated based on the recommended maximum pressure pulsations level according to the API Standard 618 [1] which is given in Equation 4-5. If the actual flow pulsations are available from a pulsation analysis this data should be used for the calculation of the fluctuating gas velocity U'_G.

The fluctuating velocity of the gas in the separator inlet nozzle can be related to the recommended maximum pressure pulsation levels with Equation 4-4:

$$U'_G = 0.5.10^5.\frac{P'}{\rho_G c} \; [m/s] \qquad \textbf{4-4}$$

In which

P'	Maximum allowable peak-to-peak level of pressure pulsations [*bara*]
ρ_G	Gas density [kg/m^3]
c	Gas speed of sound [*m/s*]

The maximum allowable pressure pulsations level is calculated with Equation 4-5 (API 618 [1] paragraph *7.9.4.2.5.2.2.2*).

$$P_1 = \sqrt{c/350}\left(\frac{400}{(P_L.D_{pipe}.f)^{0.5}}\right)[\%] \qquad \textbf{4-5}$$

$$P' = P_1.P_L/100$$

In which

P'	Maximum allowable peak-to-peak level of pressure pulsations [*bara*]
P_1	Maximum allowable peak-to-peak level of individual pulsation components corresponding to the fundamental and harmonic frequencies, as a percentage of the average absolute line pressure [%]
P_L	Mean absolute pressure in the line [*bara*]
c	Sound speed of gas [*m/s*]
D_{pipe}	Inlet pipe inner diameter [*mm*]
f	pulsation frequency [*Hz*]

The pulsation frequency is derived from Equation 4-6

$$f = \frac{N.z}{60} \; [Hz] \qquad \textbf{4-6}$$

In which

N	Shaft speed [*r/min*]
z	1,2,3,… corresponding to the fundamental (1) or multiples of [-]

The inlet nozzle sizing should be done by considering the maximum gas flow rate (Equation 4-7).

$$Q_{G,\max} = \overline{Q}_G + Q'_G \ [m^3/s] \qquad \textbf{4-7}$$

In which

$\overline{Q}_G$ Average actual gas volumetric flow rate [m^3/s]

Q'_G Fluctuating actual gas volumetric flow rate [m^3/s]

4.3.1.2 Souders-Brown velocity (K)

The Souders-Brown velocity *K* is design dependent and its value can vary. Typical values for *K* are as follows:

a) For vertical knock-out drums with horizontal mesh pads (Equation 4-8 and Equation 4-9)

K_{vf}= 0.075 [m/s], for a pressure range: $0 < P_{barg} \leq 7$ **4-8**

K_{vf}= 0.7(0.107-0.0004.(P_{barg} -7)) [m/s], for a pressure range: $7 \leq P_{barg} < 105$ **4-9**

In which:

P_{barg}: Maximum operating gauge pressure of all specified conditions [*barg*]

K_{vf}: K vertical flow (Horizontal mist mat)

a1) For pressures above 105 barg the capacity will be further de-rated and the design procedure should be done in consultation with a separator expert.

a2) For most vapours under vacuum conditions: K_{vf}=0.06 [m/s]

b) For vertical vessels without mist eliminators (such as vane or wire-mesh): divide the value of K_{vf} (Equation 4-8 and Equation 4-9) by two.

c) For vertical wire mesh pads (Horizontal flow) $K_{hf} = 0.85*K_{vf}$

d) Typical values for vane-type demisters are:

- Horizontal flow Single-Pocket : K= 0.2 [m/s]
- Horizontal flow Double-Pocket vane: K= 0.25 [m/s]
- Vertical flow, double pocket vane: K= 0.15 [m/s]

Note 1: The K- values for demisters, such as wire mesh or vane are typical values only. The capacity of a wire mesh and vane demisters can substantially change by the operating conditions and fluid properties.

Note 2: The K value for demisters, such as wire mesh or vane, will be decreased by 40% for vacuum or pressures above 7 barg for both horizontal and vertical separator vessels.

4.3.2 Separator vessels' height/length

The height of vertical separators or length of horizontal separators are shown in Figure 4-2 and

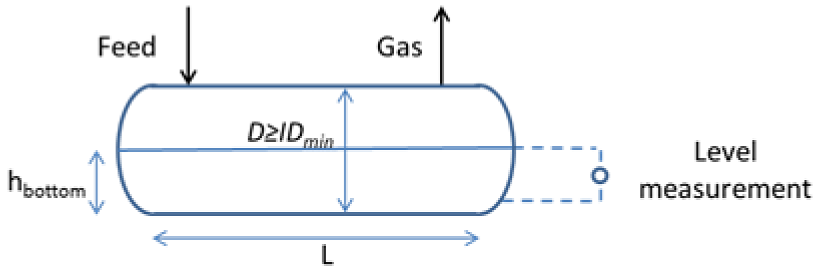


Figure 4-3.

Note: The separator should not be oversized. A larger separator volume will decrease the turn-down ratio of the separator which relies on centrifugal forces (mono cyclones).

4.3.2.1 Vertical separators

The height of vertical separators is calculated by Equation 4-10:

$$H = h_{bottom} + H_{top}\,[m] \qquad \textbf{4-10}$$

In which:

H	The height of separator, between the bottom and top tangent lines [m]
h_{bottom}	The liquid height between the low-level trip (LLT) and high-level trip (HLT) [m]
H_{top}	The height at the top section, between the high level trip and top tangent line [m]

- The height of high level trip is calculated from section 4.4.5.
- The value for the height of top section (H_{top}) can be calculated from Table 2.
- To keep the vibrations as low as possible the height should be kept to a minimum. Appendix J gives engineering rules how to keep the vibrations of the separator acceptable.

Table 2. Recommended value for the height of top section (H_{top}) for vertical separators

Vertical knock-out drums	$H_{top} \geq 1.5D$ (with a minimum of 2 m)
Vertical wire mesh	$H_{top} \geq D$ (with a minimum of 1.5 m)
Vertical vane-type	$H_{top} \geq 1m$ + height of the vane unit
Cyclones with a tangential inlet	$H_{top} \geq 2.5D + 1.3$* diameter of the inlet nozzle
Cyclones with a swirler	Total height should be > 4.2 D.

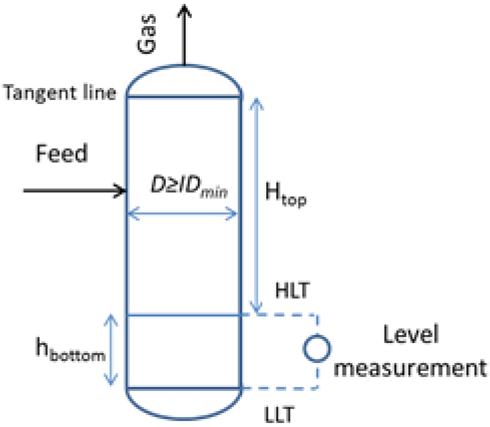


Figure 4-2. Definition of separator height for vertical separator

Note that the values in Table 2 are conservative recommendations. These values can lead to large height to diameter ratios and consequently, especially in combination with heavy top covers, cause vibration problems. An experienced engineer and or vibration expert should be consulted during the engineering design phase.

4.3.2.2 Horizontal separators

- The length of horizontal separators can be determined based on the operating pressure and the ratio of the horizontal separator length to the vessel diameter as given in Table 3.

Table 3. The ratio between the length and diameter of the horizontal separator [16]

Operating pressure [*barg*]	**L/D**
0-20	2.5-3
20-35	4
Above 35	5-6

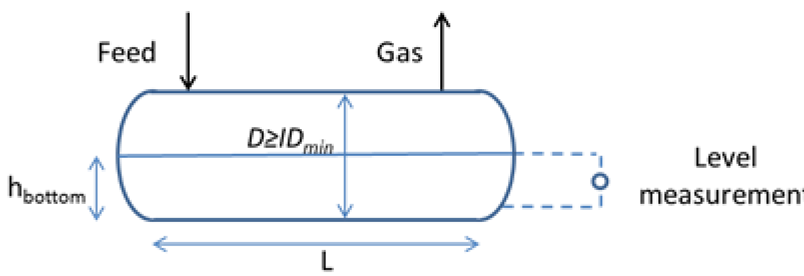


Figure 4-3. Definition of separator length for horizontal separator

- For the preliminary design, the height of bottom section should be half the vessel diameter (h_{bottom}/D=0.5) (more information can be found in [16]).

4.3.3 Droplet size prediction

According to the API Standard 618, the liquid separator device shall remove 99% of all droplets of 10 µm or larger which as indicated previously is not realistically feasible for all applications and under all circumstances. The chosen demister should at least be able to remove the fine droplets in the system. Note that the droplet size is strongly dependent on local flow properties and operating

conditions and the droplets size can be described with a Gaussian distribution. More information about droplet size distribution is given in Annex H.
The approximation of the Sauter-mean droplet size that can be formed in the feed pipe is given by [11] and can be calculated with Equation 4-11:

$$d_{droplet,SM} = 2 * 0.00394 \frac{\sigma}{\rho_G U_{SG}^2} \mathrm{Re}_G^{2/3} \left(\frac{\rho_G}{\rho_L}\right)^{-1/3} \left(\frac{\mu_G}{\mu_L}\right)^{2/3} [m] \qquad \textbf{4-11}$$

In which

$d_{droplet,SM}$	Sauter-mean diameter of the droplet [*m*]
ρ_G	Gas density [kg/m^3]
ρ_L	Liquid density [kg/m^3]
U_{SG}	Average superficial gas velocity [*m/s*]
Re_G	Reynolds number of the gas phase [-]
σ	Surface tension of the droplet in the gas [*N/m*], (Table 4)
μ_L	Dynamic liquid(droplet) viscosity [*Pa.s*]
μ_G	Dynamic gas viscosity [*Pa.s*]

Note that Equation 4-11 gives mean droplet diameter based on the average superficial gas velocity. Based on reference [5], higher gas velocities break up the droplets and lead to smaller droplet size. Thus, the maximum gas velocity (average gas velocity plus fluctuating velocity) can also be used to determine the mean droplets diameter by considering the gas fluctuating velocity.

Gas Reynolds number is calculated by Equation 4-12:

$$\mathrm{Re}_G = \frac{\rho_G U_{SG} D_{pipe}}{\mu_G} \text{ [-]} \qquad \textbf{4-12}$$

In which:

ρ_G	Gas density [kg/m^3]
U_{SG}	Superficial gas velocity [m/*s*]
D_{pipe}	Pipe inner diameter [*m*]
μ_G	Dynamic gas viscosity [*Pa.s*]

The superficial gas velocity is calculated by Equation 4-13:

$$U_{SG} = \frac{4 Q_G}{\pi D_{pipe}^2} [m/s] \qquad \textbf{4-13}$$

In which:

Q_G	Actual volumetric gas flow rate [m^3/s]
D_{pipe}	Pipe inner diameter [*m*]

The values for surface tension σ shall be taken from handbooks or process simulation programs. For some liquid, the surface tension value is given in Table 4.

Table 4. Values of surface tension (σ) for several types of liquid

Liquid	Surface tension [*N/m*]
Water (25 ˚C)	0.072
n-heptane (20 ˚C)	0.02
Benzene (20 ˚C)	0.029
Ethanol (20 ˚C)	0.023
Glycerol (20 ˚C)	0.063

The selected demister should be able to separate the droplets in the range of d_{10} and $d_{droplet,SM}$. The value of d_{10} can vary for different process conditions. As a first estimate, the ratio of $d_{droplet\text{-}SM}/d_{10}$ is 1.8 to 2 (the higher the liquid or gas rate is, the lower the ratio). More information on the determination of droplet distribution is given in reference [15].

4.3.4 Inlet/Outlet nozzle diameter

4.3.4.1 Inlet nozzle diameter

The feed nozzle diameter should be chosen based on the maximum momentum caused by the feed stream. If no inlet device or diffuser is used at the separator inlet, the diameter of the inlet (feed) nozzle can be calculated with Equation 4-14.

$$\frac{1}{D_{feed}^4} \leq \frac{870}{\rho_{feed}.Q_{feed}^2} \; [1/m^4] \qquad \textbf{4-14}$$

In which:

D_{feed}	Feed pipe inner diameter [*m*]
Q_{feed}	Inlet feed volumetric flow rate [m^3/s]
ρ_{feed}	Feed density [kg/m^3]

If an inlet device is used, for instance a half-open pipe, the inlet nozzle pipe of a separator should be sized based on the internal company procedures, international guidelines, and engineering practices or in consultation with a separator specialist. The procedure for the design of the feed inner diameter is given in Annex I.

- If the calculated feed pipe diameter is smaller than the connecting pipe diameter, the diameter of the feed pipe should be the same as the connecting pipe.
- If the calculated feed pipe diameter is larger than the connecting pipe diameter, consultation with a separator expert is necessary for using proper connectors and sufficient entrance length.

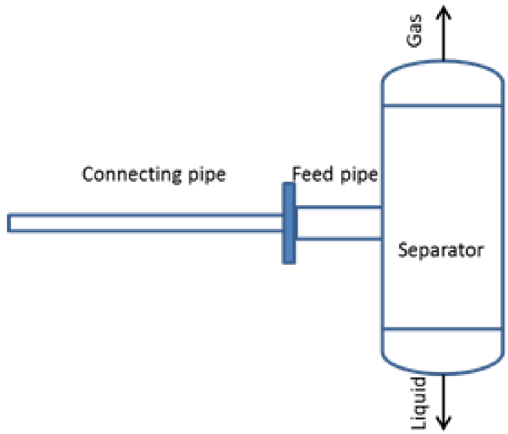


Figure 4-4. The schematic of the connecting and feed pipe of a separator

4.3.4.2 Outlet nozzle diameter

The gas outlet nozzle should be equal to the outlet piping. The liquid outlet nozzle should be sized such that the liquid velocity in the outlet nozzle does not exceed 1m/s. The turbulence of the liquid volume at the bottom of the separator vessel should be broken by using vortex breaker devices.

4.3.5 Inlet/Outlet nozzles spacing

The spacing between the inlet/outlet nozzle and the demister pad of the separator should be checked carefully. A too short distance can lead to the entrainment of the accumulated liquid at the bottom of the separator and will lead the liquid flow to the suction line of compressor. For different inlet/outlet nozzle configurations, the rules according to Figure 4-5 should be followed.

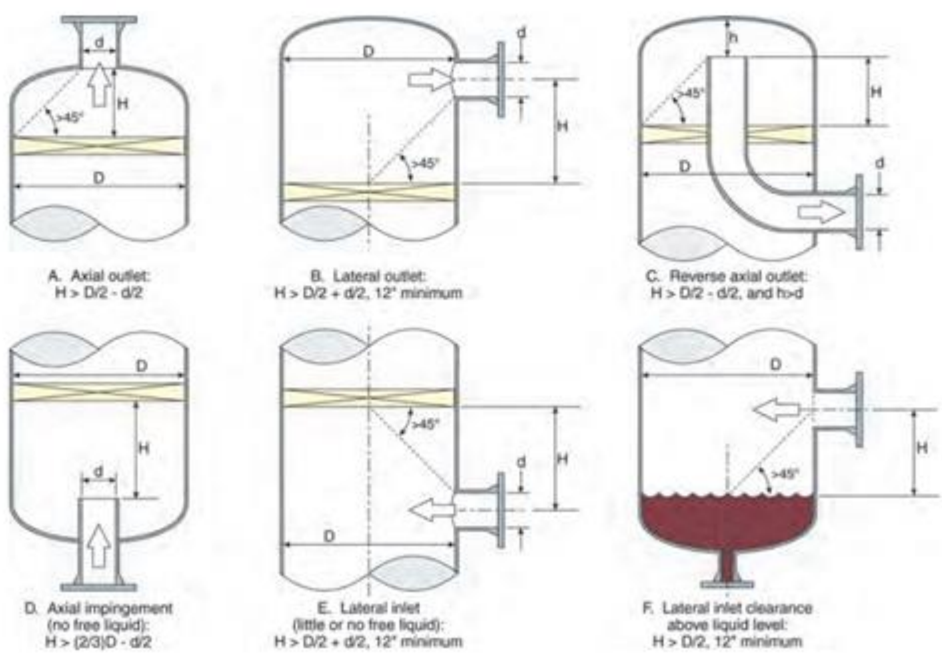


Figure 4-5. Guidelines on spacing between the inlet nozzle and demister in the cylindrical vessels (Courtesy of Amistco [6])

4.4 Additional engineering rules for the design and operation of separators

4.4.1 Number of separators

- In general separators cannot be excluded from reciprocating compressor systems. They can be excluded if all likely future process conditions (including shutdown, start-up and process unit commissioning) are known and if the gas temperature will be at least 10 ˚C above the process gas dew point for all of those conditions and ambient conditions are such that condensation cannot be expected.
  Separators are not required for non-lubricated compressors which compresses dry gases.
- A separator at the final discharge of the compressor system may be specified for a requirement of dry gas downstream of the compressor.
- Each compressor unit must be equipped with an individual separator unit if more than one compressor is mounted in the compressor system unless the separator is designed for the maximum flow of all possible combinations of running compressors.
- For lubricated compressors, a separator must be installed at the suction side (separating lube oil transported via bypass lines), at each interstage and if specified at the final discharge (e.g. if dry gases are required).

4.4.2 Location of separators

- Each separator should be located upstream of the compressor as close as possible to the compressor. For interstages, a separator must be located downstream of each intercooler as close as possible to the compressor.
- As a result of a pulsation and mechanical response analysis according to the API Standard 618 [1], it may be beneficial to locate the separator further away from the compressor to decrease the pulsations and vibrations. In that case more attention should be paid to the design of the connecting pipe between the pulsation damper and separator. A consensus should be reached in good cooperation with the process and piping engineer, pulsation and vibration expert and end-user.

4.4.3 Pressure drop (according to the API Standard 618)

The maximum pressure drop for all specified conditions on the compressor data sheet shall not exceed 0.08% of the absolute mean pressure or the percentage determined by Equation 4-15, whichever is higher. These limits shall be increased by a factor of two when the pressure drop is calculated using the total flow, where the total flow is the sum of the steady state flow plus the dynamic flow components. The dynamic flow components must be calculated with a pulsation analysis or can be estimated on the method in 4.3.1.1

$$\Delta P = 0.5\frac{(R-1)}{R} \quad [\%] \qquad \textbf{4-15}$$

In which:

Δp	Maximum pressure drop over the separator unit based on a steady flow through a separator expressed as a percentage of mean absolute pressure at the inlet of the separator [%]
R	Stage pressure ratio [-]

Note that pressure drops specified in these guidelines may be exceeded by mutual agreement between purchaser and vendor, when this is the consequence of the preferred solution to avoid pulsation and vibration problems.

4.4.4 Dew point calculation

An adequate dew point calculation is required with proven simulation programs. The uncertainty in sampling mixtures and dew point calculation with different simulation property packages can be considerable. Therefore, it is recommended to perform several simulations with several packages. It is recommended to consider at least a 10˚C uncertainty in dew point calculation (T_{dew} ± 10˚C), resulting in a minimum 10 ˚C margin from the dew point. In case of lubricated compressors, lube oil, although normally in very small quantities (ppm's), will be present as a liquid in the gas stream.

4.4.5 Design of lower part of the separator

- According to the API Standard 618, the capacity of the lower part of the separator shall be sufficient to contain the maximum expected liquid flow from any specified operating condition for not less than 15 minutes, without activating any alarm. The lower part of the vessel means up to the high-level alarm (Figure 4-6). This criterion can be used if an incoming slug into the separator is not expected.
- If an incoming slug is expected, the volume of the lower part of separator should be able to handle the liquid hold-up of the expected liquid flow for 5 minutes plus the slug volume. If the incoming slug volume is not known, the volume should be approximated as a minimum of 2-5 seconds of the maximum feed flow rate of the mixture (gas and liquid) based on the density of the liquid (100% liquid pipe hold-up). This means that a liquid column is entering the separator for 2-5 seconds with the maximum feed flow rate, see Figure 4-7.
- For the design of the lower part of the separator, being the volume between the low-level alarm and high-level alarm, the maximum value from the above methods must be taken (15 minutes or 5 minutes + slug volume).

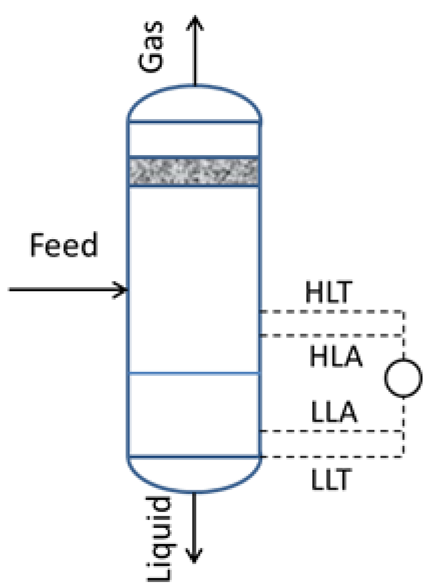


Figure 4-6. Schematic of the different alarm and trip level on the separator (LLT: low level trip, LLA: low level alarm, HLA: high level alarm, HLT: high level trip)

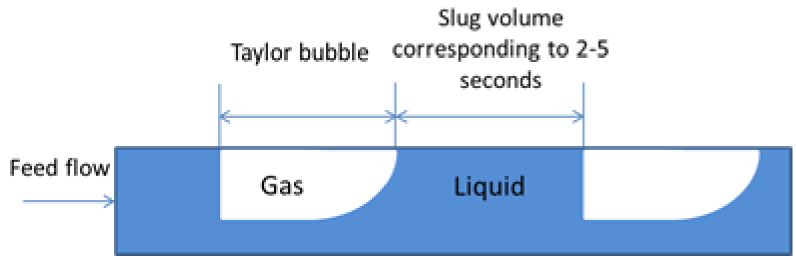


Figure 4-7. Schematic of slug flow and definition of slug volume

4.4.6 Separator versus pulsation damper

In general, the design of a separator should not be combined with the design of a pulsation damper for all type of compressors as the typically high flow pulsations associated with pulsation dampers have a negative influence on the separator efficiency [5] . A pulsation damper can be combined with a separator for those cases where the requirements of chapter 4.3 and 4.4 are fulfilled and the effect of pulsations is accounted for in the design. This should be agreed between vendor and purchaser.

4.4.7 Design of separators for different process and flow conditions

Each separator should be designed for all specified process (temperature, pressure, molecular weight, density etc.) and flow conditions (full load, part load, compressor speed range, etc.) as indicated on the compressor data sheets, including transient conditions such as start-up and shut down. Do not oversize separators as this negatively influences the turndown ratio and thus effectiveness at lower capacity.
All necessary data as required for an adequate separator design which should be given to the separator vendor are summarised in Annex D.

4.4.8 Drain system

- An automatic drain system should be installed at the lowest point of the separator vessel.
- The automatic drain system should be equipped with a manual bypass and a separate level control valve.
- The level control valve should be equipped with a level controller of an agreed type with an alarm and switch-off function to detect separator overloading.
- The capacity of the vessel between the high-level alarm and high-level trip should be equivalent to the maximum expected liquid flow for not less than 5 minutes, see Figure 4-6. The drain system (hardware and software) and its valve must be regularly inspected on its functions and in case of malfunctioning it should be replaced immediately.
- The diameter of the drain pipe should be minimum DN 25 (1 NPS). The drain valve must be mounted as close as possible to the separator wall. Or when located some distance from the separator wall, i.e. bottom drain of vertical separators on full skirts support, they shall be adequately supported. Drain systems should be designed in accordance with the API Standard 618 to minimize overhanging weight and shall be gusseted back to the separator wall in minimum two planes to avoid breakage resulting from vibrations.
- The separator shall be equipped with a level gauge. The gauge connection should be short and stiff and overhanging weight should be minimized and should be supported back to the separator wall to avoid fatigue failures caused by vibrations.

4.4.9 Pressure drop measurement

– The pressure-drop measurement system over the demister pads should not be standard equipment for indication of liquid carryover in the system. The reason is that the pressure drop over the demister is a highly fluctuating value caused by pulsations and it is not a representative indicator for liquid flooding of the demister.

4.4.10 Heat tracing and insulation

– For applications in environments where freezing of condensates can occur heat tracing is recommended at the lower part of the separator vessel.
– There is no restriction or limitation on using insulation and heat tracing on the separator with respect to liquid problems and separation efficiency.
– If heat tracing is installed on the lower part of the separator vessel, insulation should also be installed.

4.4.11 Turn down ratio

– Variation of the compressor load can increase the chance of liquid problems. For compressor systems which are designed for many different flow and unloading conditions (e.g. underground gas storage systems) separators must be selected and designed with a high enough turn down ratio which covers all specified process and flow conditions (including start-up and shut down conditions).

4.4.12 In-line separators

– In-line separators can remove fine droplets but are not capable of handling slugs. If applied they must always be used in combination with knock-out drums.

4.4.13 Multiple stage compressors

– With interstage coolers:
For multistage compressors with an intercooler, a gas/liquid separator must be installed at each interstage downstream of the intercooler and as close as possible to the compressor, unless, the gas temperature will be at least 10 ˚C above the process gas dew point for all of process conditions envelope.
– *Without interstage coolers:*
For multistage compressors without an intercooler, a gas/liquid separator should be installed at each interstage if the process gas temperature could reach below its dew point, or when lube oil accumulation cannot be avoided in the system.
– Separators are not required for non-lubricated compressors which compress dry gases.

4.4.14 Lubricated compressors

– For lubricated machines, a gas/liquid separator should be installed on each suction side of all stages and at the final discharge stage if dry gases are required.
– Omission of separators, even in systems designed in accordance with the guideline in order to prevent liquid accumulation, shall only be considered on the basis of proven designs and relevant experience.

4.4.15 Changing operating and process conditions and upgrading and revamping of the system

– If during the lifetime of the compressor system the operating and processing conditions change, the design of the separator, including the rest of the system, should be checked if it is suitable to avoid liquid problems for the changed situation.

4.4.16 Vibrations

– Internal parts of several types of separators (e.g. coalescer) can be sensitive to vibrations. Proven designs suitable for the pulsation flow of reciprocating compressors shall be used. For alternative designs or new applications additional (vibration) analysis may be required in order to avoid fatigue failures resulting in reduced liquid separation. Separator vendors shall be informed of the application for reciprocating compressor installation and consultation with vibration experts is also required.
  Besides vibrations of internal parts of separators, attached equipment to the separator vessel such as levels gauges, piping etc., can also experience fatigue failure due to large vibrations of the separator vessel.
  Appendix J gives engineering rules to minimize vibrations of the separator vessel.

4.4.17 Documentation

– To be able to check the design of the separator for the specified process and flow conditions the minimum documentation as summarised in Annex G should be provided by the vendor to the to the purchaser.

5 Pulsation dampers

5.1 Introduction

The function of pulsation dampers or pulsation suppression device (PSD) is to dampen the pulsations to acceptable levels.
Due to the fact that pulsation dampers are not designed primarily to work as a separator it is not recommended to integrate the design of a pulsation damper with a separator contrary with the API Standard 618. High flow pulsations associated with pulsation dampers have a negative influence on the separator efficiency [5]. A pulsation damper can be combined with a separator for those cases where the requirements of chapter 4.3 and 4.4 are fulfilled and the effect of pulsations is accounted for in the design. This should be agreed between vendor and purchaser.
Assuming that a separator is designed in an adequate way for all defined operating conditions, it is always possible that liquid can enter the pulsation damper during offset conditions or that condensates will be formed or accumulated inside the damper. Several design rules are given in this chapter to minimize the risk of having liquid problems in the system.
Too much cylinder lubrication oil or condensates can be trapped in the cylinder or discharge pulsation dampers in case of applying an incorrect control procedure of the cylinder lubrication system when a compressor is stopped or bad design of the upstream components. For that reason, the system must be able to handle this oil and condensates carry over in an adequate way to avoid liquid problems.

5.2 Engineering rules

In this chapter, engineering rules on the design of pulsation dampers are given. This chapter covers the design rules for both horizontal and vertical dampers. The suction and discharge pulsation dampers are designed mostly with the same principle. There are some minor differences between the suction and discharge pulsation dampers design which is covered in this chapter. However, the design of suction damper is more critical in avoiding liquid problem.

5.2.1 Horizontal damper

- Drains are not necessary for suction dampers without internals such as baffle plates or half pipes.
- Each discharge damper of a lubricated machine should be provided with a manual drain. The drain must be mounted in the first compartment downstream the cylinder nozzle for a damper with a baffle plate.
- Drain valves, holes in a baffle plate or half-pipe are not required if a compartment of a suction manifold damper is in direct connection with a cylinder nozzle (liquid will always flow into the cylinder nozzle).
- If a suction damper has a compartment which is not in direct contact with a cylinder nozzle, a drain valve must be located in that compartment, see example in left hand side picture of Figure 5-1. An alternative is to provide the baffle plate with drain holes, see right hand side picture of Figure 5-1.
- In case a half pipe is applied, either a drain valve should be mounted on both sides of the half-pipe, see Figure 5-2 or the half-pipe should be provided with drain holes on both closed sides of half-pipe (Figure 5-3). Drain hole dimensions are specified below.

- Drain holes should have a minimum diameter of 10 mm and a maximum diameter of 15 mm or 2% of the area of the part in which it is located, whichever is smaller. If the maximum calculated diameter is smaller than 10 mm, a diameter of 10 mm should be applied but it should be recognized that this may influence the acoustic performance of the damper. The aforementioned area is the blue coloured of Figure 5-3 as follows:
    - The inner area of the choke tube if the drain hole is located in a baffle plate, or the area of the hole for a baffle plate without a choke tube
    - The area of the inlet hole of the half pipe if the drain hole is located in a half pipe.
- The diameter of the drain pipe should be minimum DN 25 (1 NPS).
- The drain valve must be mounted as close as possible to the damper wall.
- Drain systems should be designed in accordance with the API Standard 618 [1] to minimize overhanging weight and shall be gusseted back to the damper wall in minimum two planes to avoid breakage resulting from vibration.
- Internals are only allowed if they are necessary to achieve acceptable pulsations levels and pulsation-induced forces.
- Cylinder nozzles must be flush mounted to the damper wall.
- Boot legs are not allowed.
- The purpose of heat tracing of a suction pulsation damper is to keep the process gas temperature constant and above its dew point. The design of the heat tracing and insulation should be done in consultation with the suppliers.
- Insulation should always be installed if heat tracing is installed.
- Insulation is always preferred to install for the suction damper. For the discharge damper insulation or other protection may be required for temperature safety reasons (personnel protection).

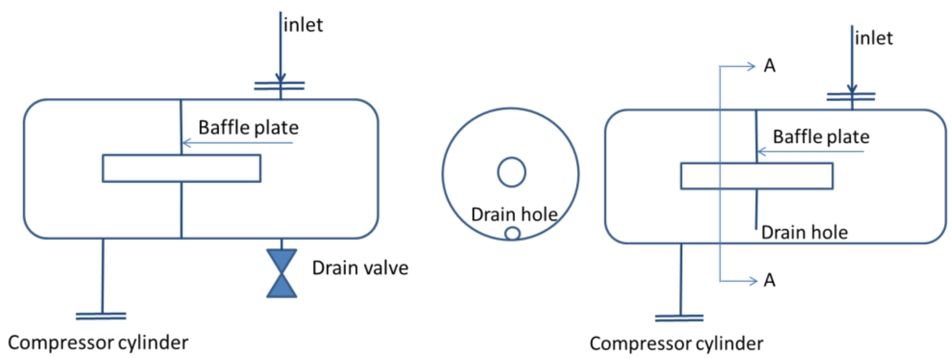


Figure 5-1. Example of a suction pulsation damper with a baffle plate with a drain valve (left) or with drain holes (right)

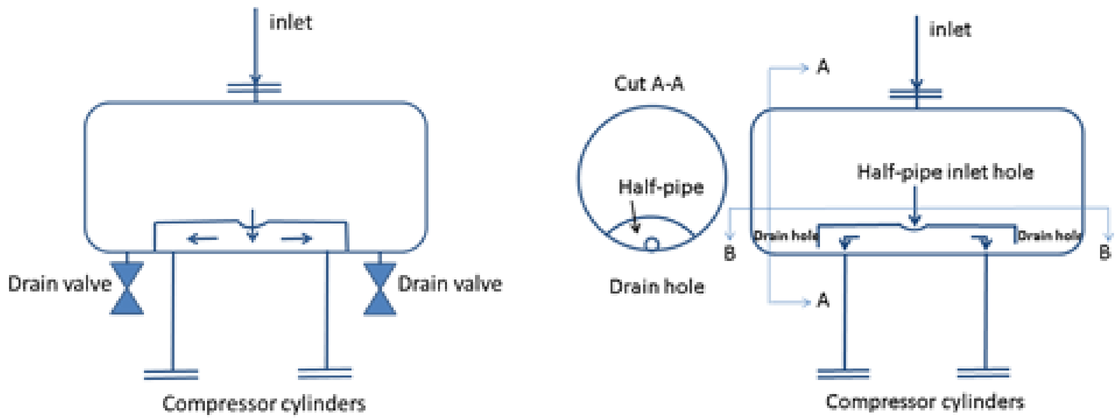


Figure 5-2. Example of a suction pulsation damper with a half-pipe with drain valves (left) or with drain holes (right)

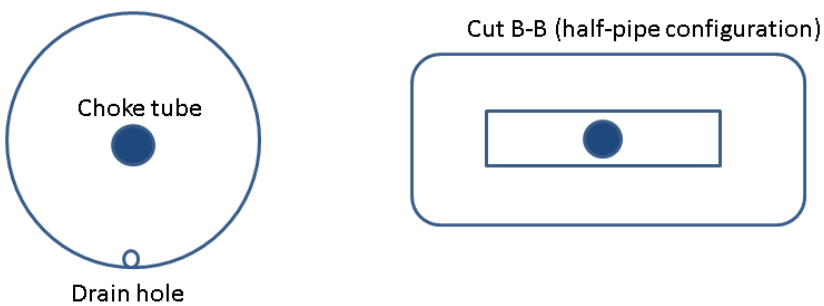


Figure 5-3. The area to be considered for the calculating of the drain hole sizing, coloured area for baffle plate(left) and half pipe(right)

5.3 Vertical pulsation dampers

- Cylinder nozzles should be kept as short as possible.
- The cylinder nozzle configuration of a suction damper as shown in Figure 5-4 is preferred.
- Cylinder nozzles should be designed with a slope towards the damper.
- Pockets in cylinder nozzles are not allowed.
- Suction and discharge dampers should be provided with a manual drain which is located at the lowest point of the damper.
- The diameter of the drain pipe should be minimum DN 25 (1 NPS).
- The drain valve must be mounted as close as possible to the damper wall.
- Drain systems should be designed in accordance with the API Standard 618 [1] to minimize overhanging weight and shall be gusseted back to the damper wall in minimum two planes to avoid breakage resulting from vibration.
- Internals are only allowed if they are necessary to achieve acceptable pulsations levels and pulsation-induced forces.
- In case a baffle plate is applied, the baffle plate must be convex to allow liquid to be removed via the drain hole. The drain hole must be located at the damper wall (Figure 5-5).
- In case internals are applied, the internals should be provided with drain holes. Drain holes should have a minimum diameter of 10 mm and a maximum diameter of 15 mm or 2% of the area of the part in which it is located, whichever is smaller. If the maximum calculated diameter is smaller than 10 mm, a diameter of 10 mm should be applied but it should be recognized that this may influence the acoustic performance of the damper. The aforementioned area is (Figure 5-3):
    - The inner area of the choke tube), if the drain hole is located in a baffle plate.
    - The area of the hole for a baffle plate without a choke tube.
    - The area of the half pipe (based on the inner diameter), if the drain hole is located in half pipes.
- The purpose of heat tracing of a suction pulsation damper is to keep the process gas temperature constant and above its dew point. The design of the heat tracing and insulation should be done in consultation with the suppliers.
- Insulation should always be installed if heat tracing is installed.
- Insulation is always preferred to install for the suction damper. For the discharge damper insulation or other protection may be required for temperature safety reasons (personnel protection).

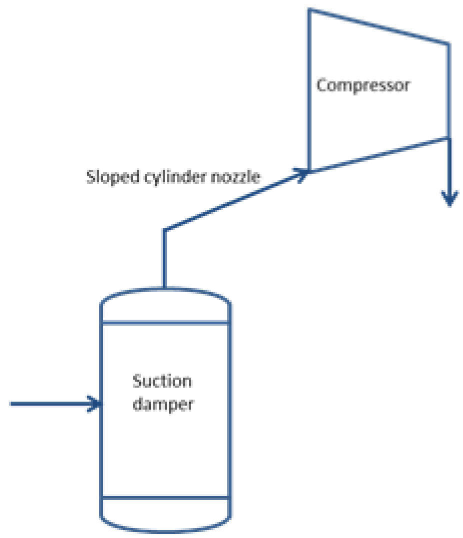


Figure 5-4. Preferred configuration of a cylinder nozzle of a vertical suction damper

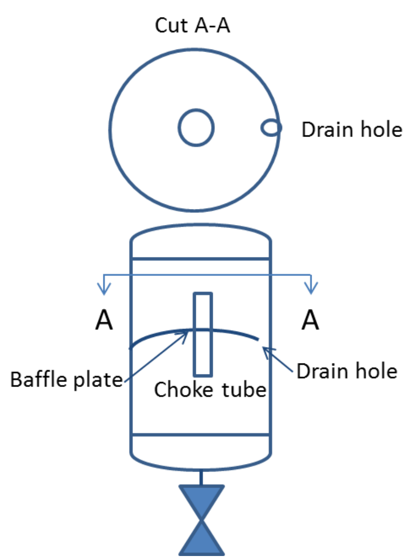


Figure 5-5. Baffle plate configuration for a vertical pulsation damper

6 Piping

6.1 Introduction

The piping design and routing is a crucial factor in avoiding liquid carryover problems in reciprocating compressor systems. In most of the cases an optimum design and routing of the piping is based on the thermal flexibility, available space, installation and maintenance costs and avoiding pulsation and vibration problems. It is always recommended to design and optimise pipe systems of reciprocating compressors systems considering liquid handling capabilities of piping configurations. An optimum design and routing with the aim of avoiding liquid problems should be carried out in good cooperation between the process and piping engineer, the vibration specialist and the end user. The design of the discharge piping with respect to liquid problems is less crucial but it is also necessary to consider. In general, the pipe configuration should be designed in such a way to avoid liquid accumulation in low points and blocked points (dead-end). In this chapter engineering guidelines are given for:

- Piping upstream the compressor inlet separator.
- Piping between the separator and suction damper (most critical part).
- Discharge piping between the pulsation damper and separator for interstage piping.
- Final discharge piping.
- It should be noted that the examples given in the following chapters are mainly included to create awareness about the possible effects of layout and pipe routing and should be treated as such.

6.2 Piping upstream the compressor inlet separator

- The suction piping upstream the inlet separator should always be sloped towards the separator. The minimum slope should always be 1:100.
- The suction piping should be designed in such a way that liquid can be collected upstream the block valve during standby of a compressor. An example of a wrong routing is given in
- Figure 6-1 where no block valve is used or where the block valve is located close to the separator (liquid accumulation in long dead end). The block valve should be mounted as close as possible to the main header at a location above the common suction header, see Figure 6-2. This configuration will help in avoiding liquid accumulation and carryover in the piping of the standby compressors. This is especially the case for the suction side if more than one compressor is mounted on the same header. This solution is in general also a beneficial solution in decreasing pulsations and vibrations.
- If this solution is not possible, provisions must be made to avoid accumulated liquid flowing into the compressor. This can be achieved by the installation of a drain which is located just upstream the blocked valve, see Figure 6-3.
- The uniformity of the flow at the inlet of the separator can enhance the separator efficiency. For this purpose, the first disturbance or device upstream the separator should be installed at the distance of minimum 10D (pipe diameter) upstream the separator inlet although it is recognized that this is not always practically feasible due to space and layout constraints.
- If there are bends with the distance of less than 10D (pipe diameter) upstream the separator, they should be in the vertical plane towards the separator inlet (from top to bottom) to avoid low points upstream the separator.

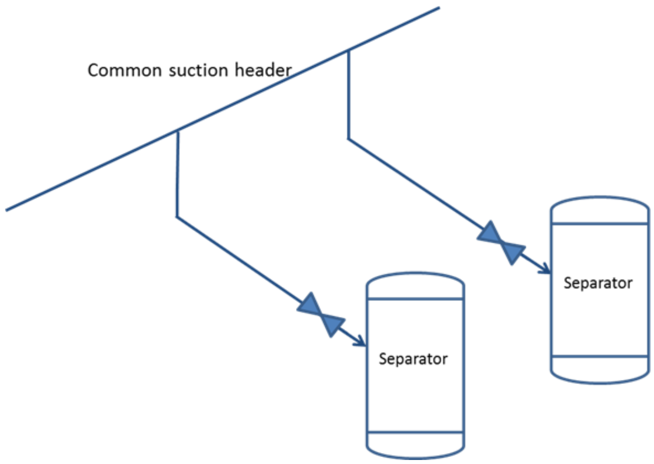


Figure 6-1. Example of a <u>wrong</u> configuration for the pipe routing upstream the separator

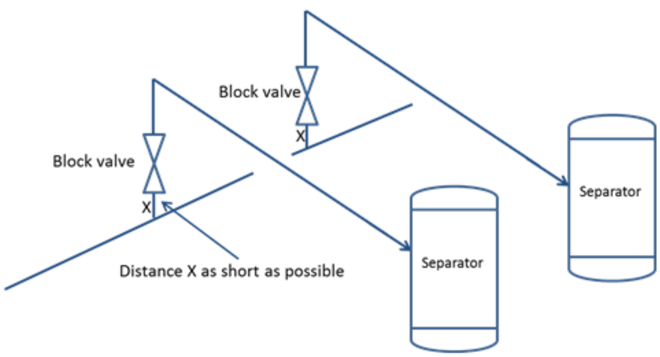


Figure 6-2. Advised location of block valves in suction headers

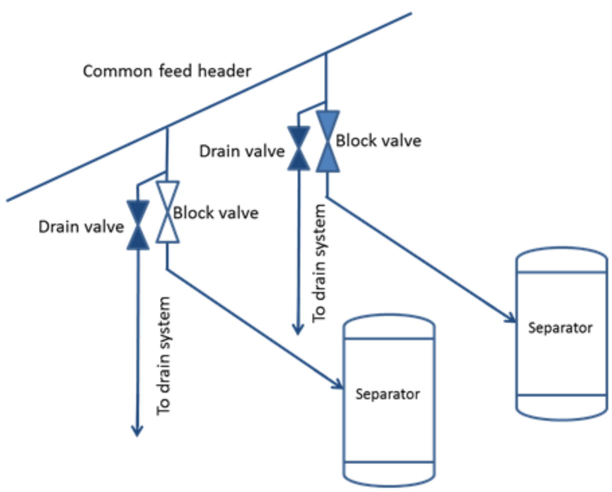


Figure 6-3. Block valves with drain systems

6.3 Piping between the separator and suction pulsation damper

In this chapter different configurations will be given for the piping between the separator and suction pulsation damper.

6.3.1 Configuration A (Figure 6-4)

The most optimum design of a suction piping with respect to avoiding liquid problem is given in Figure 6-4. In this design the outlet of the separator is at the top and the inlet of the suction pulsation damper is at the bottom. The inlet of the suction damper must be located at a point which is higher than the outlet of the separator.
The piping should be designed with a continuous slope towards the separator. The slope should always be minimum 1:100. The advantage of this design is that the liquid or condensate will always flow back to the separator during stand-by.
The disadvantage of this design is that the compressor must be mounted at a higher elevation than the separator. Another disadvantage may be that the length of the compressor cylinder nozzles will increase due to the required space for the inlet piping, which means in general that the pulsations and vibrations will increase.
It should be noted that sloped lines will not prevent liquid in the form of droplets, film or slug to enter the compressor during operation as this will be transported with the gas stream.

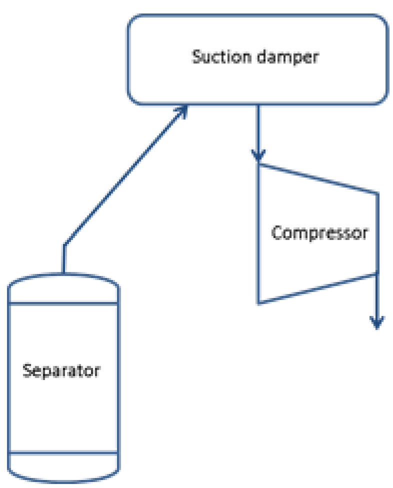


Figure 6-4. Example of Configuration A pipe routing

6.3.2 Configuration B (Figure 6-5)

If the piping of configuration A is not feasible configuration B is preferred.
In this design the outlet of the separator and the inlet of the suction pulsation damper are both on the top. The inlet of the pulsation damper may also be located at the elevation of the centreline, mounted either to the cap or wall which is the most commonly encountered execution. The piping should be designed with a continuous slope towards the separator. The minimum slope should always be 1:100. The slope should be calculated such that the liquid or condensate will be able to flow back to the separator during stand-by of the compressor. It should be noted that sloped lines will not prevent

liquid in the form of droplets, film or slug to enter the compressor during operation as this will be transported with the gas stream.

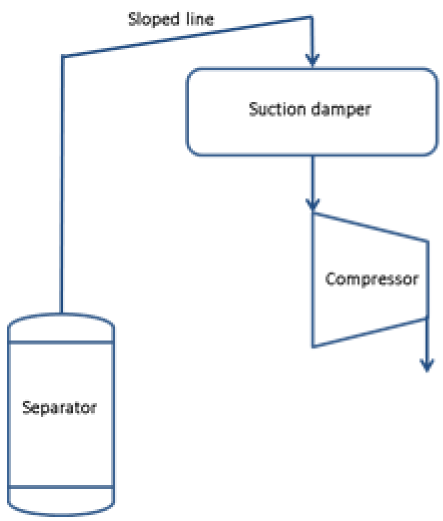


Figure 6-5. Example of configuration B pipe routing

6.3.3 Configuration C (Figure 6-6)

Configuration A and B are not possible in many cases due to space constraints, pulsation and vibration issues (see section 6.6), maintenance, etc.

A typical configuration which is applied in many cases is shown in Figure 6-6. One of the advantages of this routing is that the lowest part is located in general close to the foundation which means that it is easy to install pipe supports in this part with adequate stiffness to keep the vibrations within acceptable levels.

One of the disadvantages is that liquids or condensate accumulate in the lower parts during stand still of the compressor which can be carried over to the compressor cylinder. If this configuration is used, the piping should be equipped with heat tracing and insulation (sections 6.3.5.4) and it should be possible to remove the accumulated liquid from these pockets with a drain system as shown in Figure 6-7. Even when no liquid is expected to be formed and accumulated during operation or standstill it is recommended to install at least a manual drain at the lowest point. The valve shall be opened prior to start-up after a long standstill period.

Sloping of each horizontal pipe part should always be towards the separator.

Note: Pockets should be avoided for pressure safety lines to avoid too high dynamic liquid loads in case of opening of the pressure safety valve.

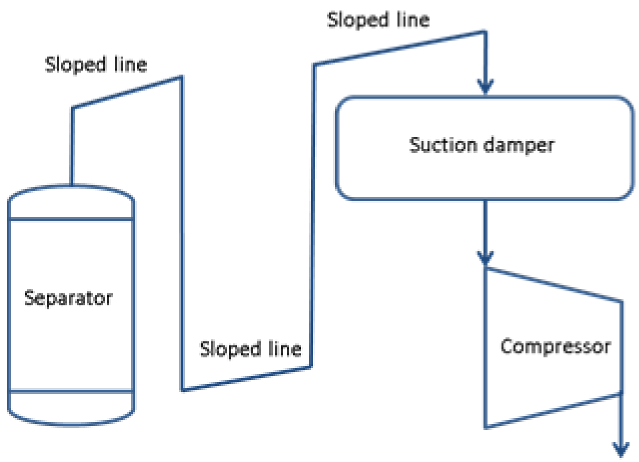


Figure 6-6. Example of configuration C pipe routing

6.3.4 Summary of advantages and disadvantages of different pipe configurations

An overview of advantages and disadvantages with respect to liquid handling capabilities and vibration handling capabilities of the piping between the separator and suction damper is given in Table 5.

Table 5. Overview of different pipe configurations

Configuration	Liquid handling capability	Vibration restrain capability*
A	++	+
B	+	-
C	-***	0**

--: very low, -: low, 0: moderate, +: high, ++: very high
Red: highly not desired, orange: not desired, light green: desired, dark green: highly desired
* Strongly depending on piping upstream capabilities.
** This configuration can lead to more pulsation-induced forces, due to the presence of more bends and can lead to higher vibration. On the other hand, the pipe routing is such that it is easier to install additional pipe supports at lower points.
*** Liquid handling capability can only be guaranteed with an automatic drain system installed as shown in Figure 6-7.

6.3.5 General engineering and design rules for the piping between the separator and suction pulsation damper

6.3.5.1 Dew point calculation

- An adequate dew point calculation is required with proven simulation programs (Annex C).
- The dew point calculation of the gas should be carried out also at interstage conditions to make certain that no fraction has gone into the two-phase region.

6.3.5.2 Sloped lines

- Each horizontal line should be sloped towards the separator (with respect to horizontal axis) with a minimum slope of 1:100 to make it possible for the liquid to flow back to the separator during the stand-still of the compressor.

– Flare lines should also be sloped with a minimum slope of 1:100.

6.3.5.3 Pockets

– Pockets should be avoided.
– If pockets cannot be avoided, a drain valve must be installed at the pocket as indicated in Figure 6-7.
– As an alternative, a bootleg can be installed at the pockets to prevent the incoming slug from the separator to the compressor. The bootleg has to be equipped with a drain valve and level control system (Figure 6-7).

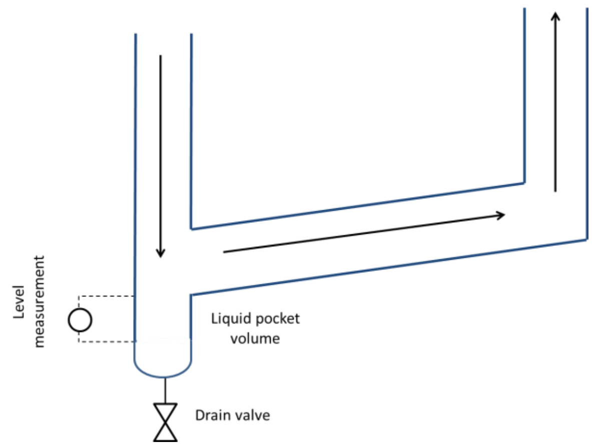


Figure 6-7. Example of a pocket with bootleg, level measurement device and a drain valve

6.3.5.4 Heat tracing and insulation

– The purpose of heat tracing and insulation of the piping is to keep the process gas temperature constant and above its dew point. The design of the heat tracing and insulation should be done in consultation with the suppliers.
– If specified, piping, pulsation suppression devices and knockout vessels at the initial and interstage suction points shall have provisions for heat tracing and insulation.
– Insulation should always be installed if heat tracing is installed.

6.3.6 Suction piping without separator

– If separators can be excluded from the system, according to section 4.4.1, the design of the suction piping of configuration A through C can be applied with the same priority as that for a system including a separator. The piping should always be sloped with a minimum value of 1:100 downward from the compressor.

6.3.7 Common suction line for multiple compressors

– It is recommended to use an individual separator for each compressor unit.
– However, if one separator is used for multiple compressors, and at least one of the compressors is stand-by, each compressor suction line should be equipped with a block valve including a drain system. As an alternative to the drain system upstream the block valve, a drain bypass line to the discharge side can be used if the discharge header is at a lower point than the suction header, see Figure 6-8.

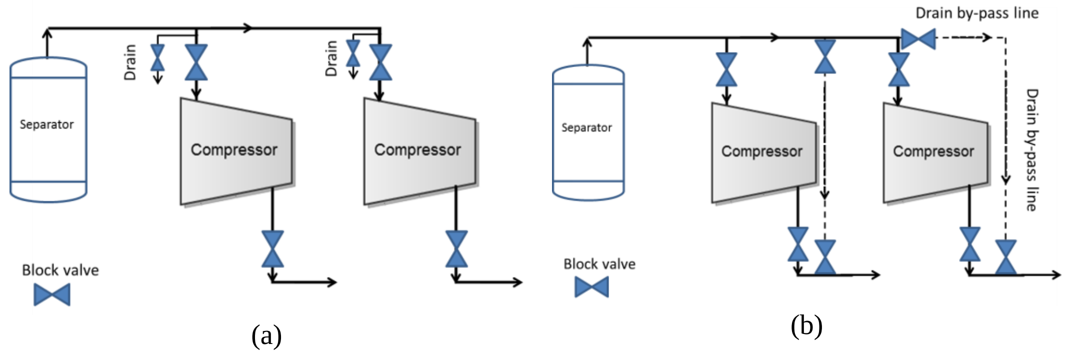


Figure 6-8. Examples of the drain by-pass line and block valve configuration for a multiple compressor system with one common separator.

Note that option (b) in Figure 6-8 is only possible when the common suction header is located at a higher elevation than the discharge side piping.

6.4 Piping downstream the discharge pulsation damper and separator for interstages

- The piping between the discharge pulsation damper and intercooler is preferred to be sloped towards the cooler.
- The piping between the cooler and separator should be sloped towards the inlet of the separator.
- If separators can be excluded from the system, according to section 4.4.1, an adequate drain system must be installed at each pocket in each interstage.
- If separators can be excluded from the system, according to section 4.4.1, heat tracing and insulation of the piping is necessary.

6.5 Final discharge piping

In case a final after cooler is installed and dry process gases are required, it is recommended to install a final discharge separator downstream the after cooler. The piping upstream and downstream the final discharge separator should always be sloped towards the separator. The slope should be minimum 1:100.

6.6 Pulsation and vibrations

A pulsation and mechanical response analysis of the pipe system shall be carried out according to the latest version of the API Standard 618 [1].
One of the solutions to decrease pulsation and vibration levels is to change the pipe routing. Due to the fact that a change in pipe routing may have an effect with respect to liquid carry over, the proposed change in pipe routing by the pulsation and vibration expert should be agreed upon by the vendor and purchaser.
The most critical part with respect to liquid problems is the piping between the separator and suction pulsation damper.

7 Miscellaneous

7.1 Cylinder Lubrication and cylinder cooling water jacket

- To prevent excessive lubricant in the compressor cylinder prior to start-up, the pre-lubrication time must be limited and should be programmed in the start-up control system.
- To prevent excessive lubricant in the compressor cylinder after a manual or automatic stop of the compressor, the cylinder lubricator pump must be programmed such that it is switched off directly after a stop.
- Some gases, although dry at the start of compression, tend to condense on cylinder walls as pressure increases. This is because the temperature at which a gas can be liquefied by heat removal (i.e., lowering its condensing temperature) increases with pressure, and in the case of certain gases, condensing temperatures that are above cylinder wall temperatures are reached as compression proceeds. Under such conditions, part of the gas in contact with the cylinder wall is cooled to the condensing temperature and liquefies. The condensate formed will do the following:
  - Tend to wash the lubricant film from the cylinder walls.
  - Dissolve in the lubricating oil, reducing its viscosity.

  Viscosity reduction in service due to these effects should be compensated for by the use of a heavier-bodied oil than would be otherwise selected or the use of a lubricating oil with a synthetic instead of mineral oil base. In addition, higher than normal jacket water temperatures are advisable to prevent or minimize condensation, which requires heavier-bodied oil.

7.2 Compressor valves

- For compressing saturated gases or in case that the chance of liquid carryover is high, the choice of compressor valves should be decided and agreed upon between the valve OEM, compressor specialist and end user.

7.3 Process conditions

- If the system will be operated for off-design conditions, such as changing the process gas composition and the process conditions (pressure, temperature etc.), or if the compressor system (piping, pulsation dampers, heat exchangers, separators, compressor etc.) will be modified, the system should be checked for the changed conditions and layout. These modifications should be done in consultation with the process and pipe engineer, OEM's, vibration specialist and end user to ensure a safe operation for the new situation with respect to liquid problems. If necessary the system should be modified accordingly.

7.4 Valve unloaders

- In many applications where valve unloaders are used and the compressor is lubricated, lubricating oil may accumulate in the cylinder during the time that suction valves are unloaded. Due to the suction valve unloading the stage or cylinder side is not working and the discharge valves remain closed. With all cylinder valves unloaded, lubricating oil cannot drain or flow away with the process gas to the discharge damper. For partly loaded cylinders oil at the unloaded cylinder side is partly transported past the piston and back through the suction valves into the loaded side due to the very turbulent flow regime but some oil can accumulate on the discharge valve of the unloaded side.
- In order to prevent accumulation of the oil it may be required to load the suction valves again after 10-15 minutes to allow the draining of lubricating oil. If this is required and the time between load-unload may be specific for a machine make, type, lubrication rates, cylinder size, etc.

7.5 Compressor Recycle operation

During compressor recycle operation, the discharge gas should be routed back to the compressor suction upstream of the separator. For most real gases the recycle gas will experience a drop-in temperature due to Joule-Thomson effect over the recycle valve, see Figure 7-1.
During significant turndown ratio's, it should be recognized that the suction temperature will be reduced and the overall process gas may be below the dew point line (or 10°C margin) resulting in a higher liquid load on the separator. In case the separator is not designed for this case, the liquids will enter the compressor.
Any turndown case should be reviewed and the Joule-Thomson effect over the recycle valve during design of the system shall be considered.

Note: Some gases such as Hydrogen, Helium and Neon will warm up under Joule-Thompson expansion.

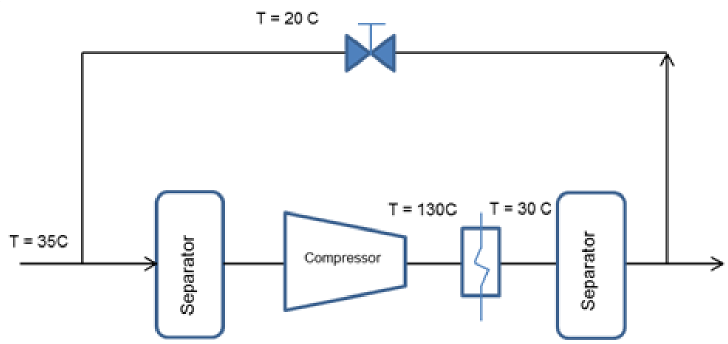


Figure 7-1. Schematic of the recycle operation

7.6 Coolers

Coolers have to be self-draining with a small slope to avoid the accumulation of liquid inside coolers. For that reason, the outlet of the coolers should be at the bottom.

8 Operation

8.1 Introduction

Liquid problems can occur during start-up of the compressor system which is caused by collected liquids and condensates in separators, pipe system and pulsation dampers. For lubricated compressors the lubricants collected in the cylinder or other parts of the systems may lead to failures if not drained in a correct way.
Because every system and compressor is different these guidelines will not attempt to provide instructions on compressor start-up etc. as OEM's may have different requirements and the large variety of process systems require different approaches.
These guidelines are intended to create awareness of items related to the prevention of liquid related problems and for the rest the end users should have their own procedures for start-up, shutdown and stand-by.
Standard procedures prepared by the end-user in cooperation with the system designer shall be used by the operators for start-up, shutdown and during stand-by conditions.
Procedures shall be based on the process and process gas system requirements and information and requirements provided by the compressor OEM.

8.2 Specific items during start-up related to liquids are as follows:

- Cooling water temperature at the cylinder inlet shall be at least 5°C above the gas inlet temperature and adequate time must be allowed prior to start-up for the heater and cooling water flow to warm up the compressor cylinders. The time required for this heating depends on the cooling water system size, heater power, cylinder sizes and ambient conditions.
- Heat tracing of piping and equipment shall be switched on and adequate time must be allowed prior to start-up to reach a thermal equilibrium. Heat tracing and cooling water systems shall be active when compressors are on stand-by.
- Pre-lubrication of cylinders and packings should be kept to a minimum and shall be advised by the OEM or shall be based on experience with a specific compressor or lubrications systems, such as point-to-point lubricators or progressive feeders (divider blocks).
  Electrical driven lubricators can easily be included in the automated compressor control system.
- After shutdown, the cylinder and packing lubrication systems shall be switched off automatically so that over-lubrication and filling of the cylinders with lube oil is avoided.
- Before start-up all process equipment such as coolers, pulsation dampers, piping and separators etc. shall be drained. Discharge pulsation dampers should not be excluded as liquid condensates and lube oil can accumulate and may cause problems in downstream compressor stages and equipment or enter the stage suction stream via by-pass lines.
  Draining shall not be limited to equipment only, but also include any low/cold spots and pockets in piping systems or at closed branches where liquid can accumulate. Depending on the separator type the liquid level shall not go below the minimum level.
  Closed drain systems make it much easier to drain under all circumstances especially when the compressor system is kept under pressure in the stand-by condition (e.g. for underground storage systems or for offshore applications).

- Before start-up and on a regular basis during periods of stand-by the compressor shall be barred over to ensure the compressor rotates freely and no excessive amounts of liquid are present inside the compressor cylinders.
- If the process allows for it, capacity should be increased in small increments by means of valve unloading, clearance pockets, by-pass control and or compressor speed control or a combination as to prevent high initial flow variations.
- Draining of all points should be performed regularly and a common practice is doing this once every shift. This interval can be altered depending on the amount of liquid found in the process gas streams which is drained from the separators and at other drain points.

ANNEX

Annex A Type of separators and demisters

The liquid capturing mechanism is different for different devices and operating conditions. In general, the separation mechanisms can be divided into gravity, inertia and droplets coalescing based, see Annex E for a detailed description.
Note that the efficiency values as given below are estimated values only for a typical design of a separator. The intention of this chapter is to make a relative comparison between different type of separators.
Alternative solutions may be considered when it has been agreed between purchaser and vendor once all normal and transient conditions in combination with compressor capabilities have been reviewed in consultation with a separator specialist.

A.1. Knock-out drum

- Working principal: gravity based.
- No internals – shell only.
- There are two types of knock-out drums: horizontal and vertical (Figure A-1).
- Very high turndown ratio (turndown ratio of knock-out drums is not limited).
- Liquid removal efficiency: moderate of approximately 90%.
- They are not working efficiently in separation of fine mist. Demister or filter pads should be added to increase the liquid removal efficiency (specifically droplet separation). The selection of demisters will be explained in next parts.
- Not sensitive to fouling.
- Knock-out drums add a pressure drop of less than approximately 5 mbar to the system. However, the pressure drop in the system is very dependent on operating conditions.

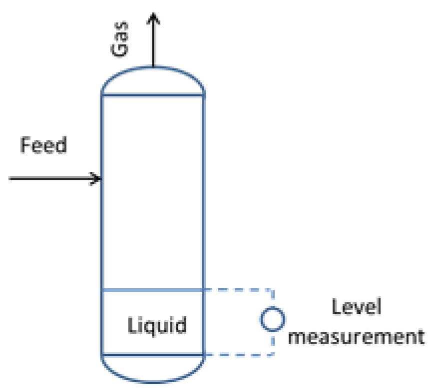
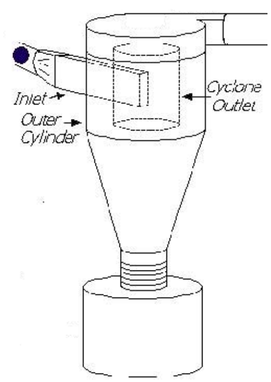


Figure A-1. Schematic of a vertical knock-out drum

A.2. Cyclone

- Working principal: inertia based (centrifugal force).
- There are two types of cyclones: with tangential inlet (left hand sided of Figure A-2) and with swirler (right hand side of Figure A-2)
- Moderate turn down ratio for a cyclone in combination with a swirler (between 3 to 5).
- Liquid removal efficiency above approximately 95%.
- Cyclones with a swirler are more efficient in removing droplets.
- Cyclones with tangential inlet have better fouling handling capabilities than cyclones with swirler.
- A cyclone adds a pressure drop of approximately 15-20 mbar to the system. However, the pressure drop in the system is very dependent on operating conditions.



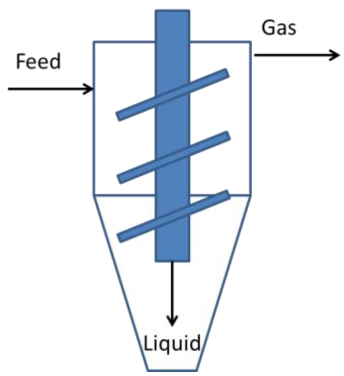


Figure A-2. Schematic of a cyclone with a tangential inlet (left) and with a swirler (right)

A.3. Vane-type demister

- Working principal: inertia based.
- There are two types of vane-type demisters: horizontal and vertical.
- Moderate turn down ratio (between 3 to 4).
- Liquid removal efficiency above approximately 95%.
- It is capable of removing fine droplets above 10 μm.
- Vane-type demisters can also be installed in an in-line configuration (Figure A-3). This configuration cannot handle large liquid volume or slugs.
- Sensitive to fouling and highly viscous fluids.
- Not to be used at operating pressures > 70 bar.
- Vane-type separators add an additional pressure drop of approximately 10 mbar to the system. However, the pressure drop in the system is very dependent on operating conditions.

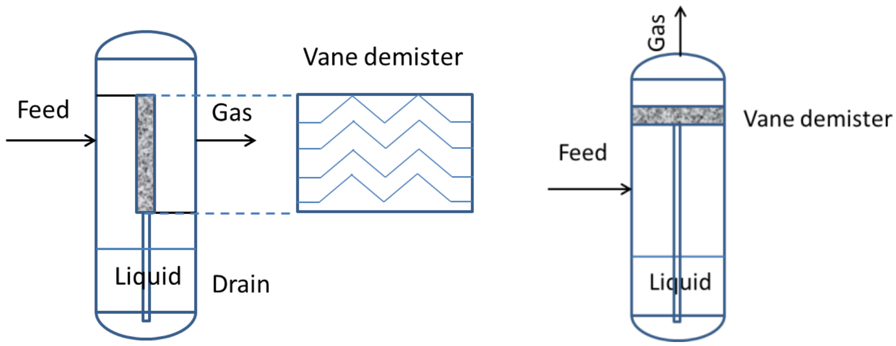


Figure A-3. Schematic of a vertical (left) and vertical (right) vane-type demister

A.4. Wire mesh demister

- Working principal: inertia based (interception capture).
- Wire mesh demister can be used in both horizontal and vertical knock-out drum.
- Low turn down ratio (between 2 to 3).
- Liquid removal efficiency around approximately 99%.
- It is capable of removing fine droplets and slugs.
- Sensitive to fouling and highly viscous fluids.
- Wire mesh separators add an additional pressure drop of approximately 10 mbar to the system. However, the pressure drop in the system is very dependent on operating conditions and type of wire mesh.

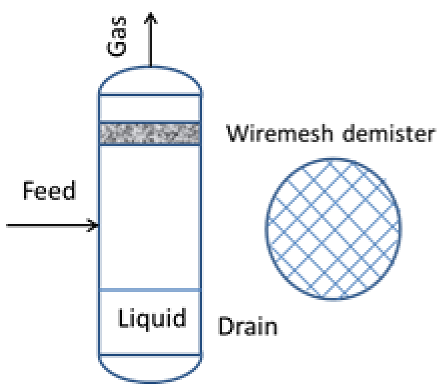


Figure A-4. Schematic of vertical wire mesh demister

A.5. Cyclone packs

- Working principal: inertia based (centrifugal force). The cyclone pack consists of several swirl tubes, which are axial cyclones. It can be used in combination with a wire mesh or vane pack, based on the application and limitations.
- Turn down ratio is dependent on the combination used in the separator vessel (4 for cyclone packs with a vane pack and 10 for cyclone packs with a wire mesh).
- Liquid removal efficiency around approximately 99%. These separators are suited for high pressure applications.
- Moderate tolerance for fouling and highly viscous liquid.
- The pressure drop caused in the system is larger than vane types and smaller than coalescers.

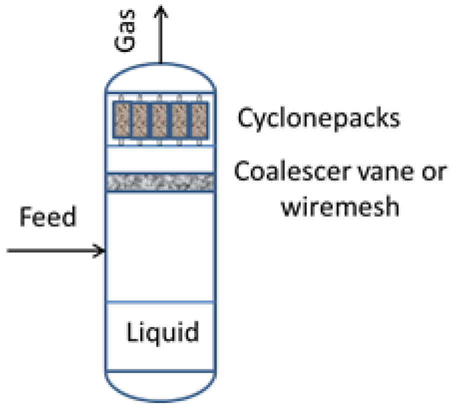


Figure A-5. Schematic of vertical separator with cyclone packs and coalescer vane/wire mesh

A.6. Coalescers

- Working principal: inertia and turbulent deposition based.
- High turn down ratio (unlimited).
- Liquid removal efficiency around approximately 99%.
- It is capable of removing fine droplets in the order of 0.1 μm.
- It is not capable of handling slugs.
- Very sensitive to fouling.
- Very high pressure drop. This depends on the type of coalescers but a typical additional pressure drop is in the range of 15 to 40 mbar [8, 10]. However, the pressure drop in the system is very dependent on operating conditions and droplets size.

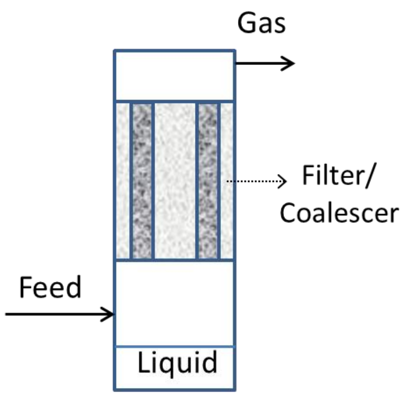


Figure A-6. Schematic of vertical separator with coalescer filter

Annex B Multiphase flow

B.1. Fundamentals of multiphase flows

The multiphase pipe flow can be categorized into different pattern, such as mist, stratified flow, plug flow, slug flow, churn flow, etc. The flow regimes depend on the pipe flow parameters (horizontal or vertical pipe flow, gas and liquid velocity, etc.). The superficial velocities for different phases are calculated by Equation B-1 and Equation B-2.

$$U_{SG} = \frac{Q_G}{A_P} \text{ [m/s]} \quad \textbf{B-1}$$

$$U_{SL} = \frac{Q_L}{A_P} \text{ [m/s]} \quad \textbf{B-2}$$

In which:

U_{SG}	Superficial gas velocity [*m/s*]
U_{SL}	Superficial liquid velocity [*m/s*]
A_P	The cross-sectional area of the pipe [m^2]
Q_G	Actual volumetric flow rate of gas [m^3/s]
Q_L	Actual volumetric flow rate of liquid [m^3/s]

The flow regimes are shown schematically for the horizontal and the vertical pipe flow in Figure B-1 and Figure B-2.

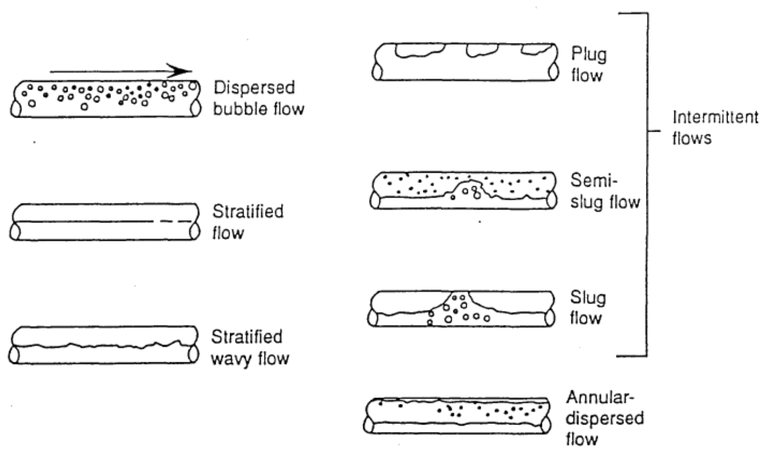


Figure B-1. Flow patterns in a horizontal pipe flow

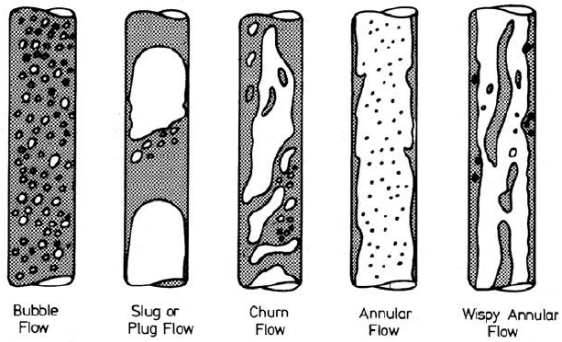


Figure B-2. Flow patterns in a vertical pipe flow

Flow maps are used to show the boundaries between the flow regimes based on the superficial velocity of the liquid and gas phases. An example of the horizontal and vertical flow maps is given in the Figure B-3 and Figure B-4. It is important to note that these flow maps are not universal and they will differ for different process gases and liquids at different operating conditions.

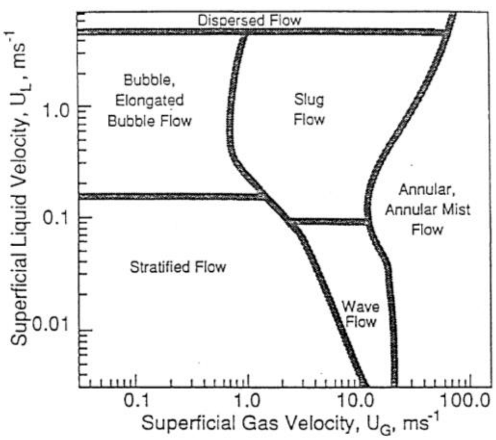


Figure B-3. Flow pattern map for a horizontal air-water flow [12]

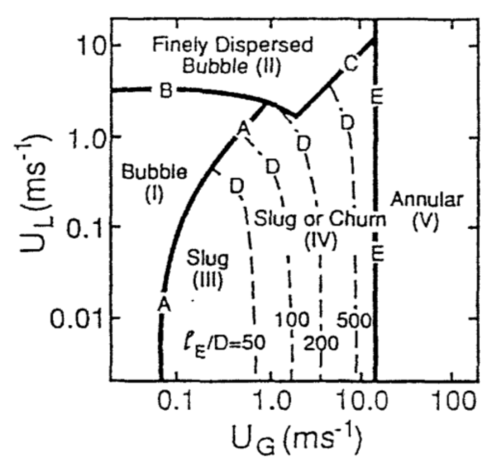


Figure B-4. Flow pattern map for a vertical air-water flow [13]

Annex C Dew point calculation

Dew point is the temperature in which the mixture components will start to condense out of the gaseous phase and forms liquid at a given pressure. This temperature depends on the pressure and the composition of the mixture. For understanding the dew point dependence on the pressure and compositions, the binary boiling point diagram is introduced.

C.1. Water vapour

The dew point is a function of temperature, pressure and the compositions. There are several ways to calculate the dew point of a mixture. The formula for calculating dew point is given in Equation C-1:

$$T_d = \frac{b.\alpha(T,x)}{a-\alpha(T,x)} \quad [^\circ\mathrm{C}] \qquad \textbf{C-1}$$

$$\alpha(T,x) = \frac{a.T}{b+T} + \ln\left(\frac{x}{100}\right) \quad [-] \qquad \textbf{C-2}$$

In which:

x	Relative humidity [%]
T	Temperature [˚C]
a	value of constant = 17.27 [-]
b	value of constant = 237.7 [˚C]

The relative humidity is defined as the ratio of water vapour partial pressure in the mixture and saturated vapour pressure of water. This equation will be valid for temperature range of $0^\circ C \leq T \leq 60^\circ C$. The uncertainty incorporated in the dew point calculation is about ±0.4˚C.

C.2. Hydrocarbons

Dew point calculation for hydrocarbons can be done through two methods: direct measurement and indirect measurement (using compositional analysis and equation of state). In direct measurement, experimental investigation and measurements are used for determination of the temperature when the first drop of condensate observed.
The indirect method uses compositional analysis, for instance with gas chromatography (GC), together with equation of state (EOS). One of the most mentioned EOS for calculating the dew point is the Peng-Robinson equation of state (Equation). The dew point can be calculated for hydrocarbons using proven in-house codes or commercial software.

Annex D Required separator design data

The following data should be provided for all possible process conditions:

1. Design pressures [bara]
2. Design temperatures [ºC]
3. Minimum surrounding temperatures for all weather conditions including freezing and snow conditions [ºC]
4. Molecular weight [kg/kmol]
5. Flow [kg/hr]
6. Gas density [kg/m^3]
7. Gas compositions
8. Compressor unloading scheme
9. Cylinder cooling temperature [ºC]
10. Type of compressor: horizontal, vertical, L, V or W
11. Compressor speed range [rpm]
12. Heat tracing of connection pipe between separator and pulsation damper [Y/N]
13. Insulation of connection pipe between separator and pulsation damper [Y/N]
14. Manual or automatic drain system [manual/automatic]
15. Type of level gauge control system
16. Maximum gas fluctuation velocity at the inlet of the separator U'_G [m/s]
17. Indication of low and high-level trip locations

Annex E Fundamentals of separator mechanism

This annex is a description of the physical phenomena in separating liquid from gas in different Gas/Liquid separators.

E.1 Inertial capture in vanes

According to Figure E-1, as the gas changes its direction, the liquid is not able to change its direction as fast as that of gas due to higher inertia. Thus, the liquid drops will be kept on the surfaces and will be drained downwards. The configuration of the vane arrays for a horizontal and vertical flow are different and they are shown in Figure E-1.

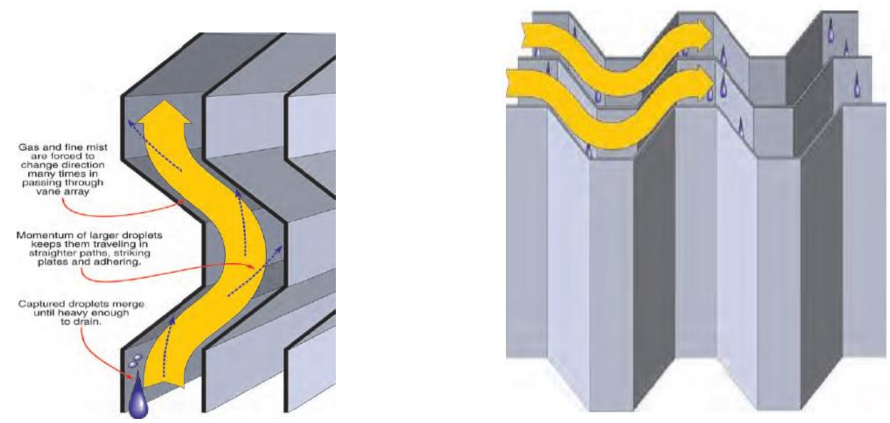


Figure E-1. Vane array with vertical flow (left) and horizontal flow (right) [14]

E.2 Inertial capture in mesh

Each strand act as a bluff body and the gas must travel around it. The liquid droplets are unable to follow the gas flow line which means that the droplets will form a film on the wall, adhere and coalesce to form larger films and will finally fall down (Figure E-2).

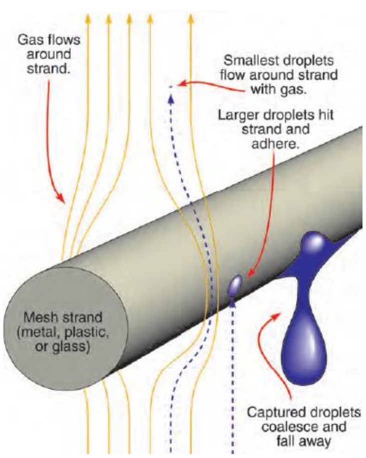


Figure E-2. Droplet capture in a mesh-type mist eliminator [14]

The inertial capture efficiency will be affected by several factors. One of these factors is the droplets diameter. Larger droplets are less sensitive to the changes in the flow path which means that they can be captured easier. The other factor is the strand diameter and corrugation spacing. The smaller the diameter of the strand or the spacing, the more difficult it is for the droplets to follow the gas flow path.
The other parameter is the gas velocity which will increase the separation efficiency at higher velocities. However, there is a limiting value for the velocity, due to the fact that very high velocities can enhance the droplet entrainment and flooding will occur. Another factor which will influence the capturing efficiency is the pad density and thickness. The thicker the pad will lead to a higher efficiency but will also increase the pressure drop. Thus, there shall be a balance between the pressure drop through the pad and the capturing efficiency.

E.3 Interception capture
This capture can occur in both mesh and vane type separators. The droplets that cannot be efficiently removed by the inertia will stay close to the centreline of the strand and adhere. It is important to note that the distinction between interception and inertia capturing is not easy to determine.

E.4 Brownian capture
This mechanism is efficient in removing submicron droplets with fibre mist eliminators. It is caused by the random motion of small droplets which can make contact between the droplets and the wall.

Annex F Separator components

According to API 12J [3], the separator components and sections in general can be defined as follows:

Primary Separation Section
This section aims to remove the bulk of liquid at the inlet stream to minimize the gas turbulence and liquid entrainment at the inlet and make the second separation more efficient. This function is performed by absorbing the flow momentum by means of inlet baffling.

Secondary Separation Section
This is the major unit of the separator in which the liquid is separated from the gas stream. There are several types of separators which have different working principles. One type is the knock-out vessel which separates the liquid by means of gravity. The efficiency of this component depends on the particle or droplet size, turbulent intensity, etc.

Liquid Accumulator Section
This section accumulates liquid after separation. The liquid in this section should have very low level of turbulence in order to avoid the entrainment of the liquid droplets to the gas stream. In some cases, vortex breaker is utilized to prevent the entrainment. Sufficient capacity of the accumulator is an important consideration.

Mist Extraction Section
The mist extractor of the coalescing section can be one of several designs (a series of vanes, woven wire mesh pad or a centrifugal device). The mist extractor removes from the gas stream the small droplets of liquid before the gas leaves the vessel.

Relief Devices
This is a protective device which is necessary for all the separators. One of the common devices is a pressure relief valve in conjunction with a rupture disk.

Discharge Lines
Discharge lines from pressure relief devices should receive consideration on an individual basis.

Other Controls and Accessories
When specified by the purchaser, separators may be equipped with other controls and accessories such as the following:

- inlet shut-in valve;
- pressure sensor or control;
- level sensor or control;
- temperature sensor or control.

Annex G Documentation

The following minimum documentation should be provided together with the separator:

1. Dimension of the vessel (diameter, height).
2. Orientation of the drawing.
3. Dimension of the piping (inlet, gas outlet, liquid outlet).
4. Designed pressure and temperature.
5. Inlet device (if exist).
6. Capacity.
7. Turn down ratio and designed liquid removal efficiency.
8. Specification on demister (type, size, range of droplet handling, droplet removal efficiency, etc.)
9. Drain sizing.
10. Specification on level gauge, level switch and level alarm.
11. Specification of heat tracing or insulation (if exist).

Annex H Droplet size distribution and cut-off diameter

In gas/liquid flows, liquid droplets can be formed by the instability at the interface of two phases and can be entrained. Based on the local forces, process conditions and materials properties, droplets are formed and travelled with different sizes. For this purpose, the droplet sizes are usually described with a distribution which is representative for the total volume of the droplets. An example of the droplet size distribution for horizontal annular gas/liquid flow is shown in Figure H-1.

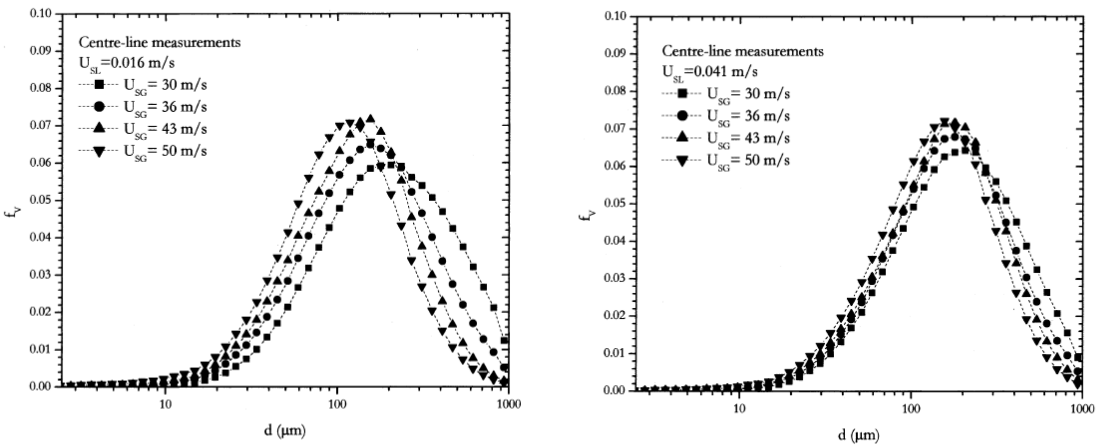


Figure H-1. Sample droplet diameter distribution for U_{sl} = 0.016 m/s (left) and U_{sl} = 0.041 m/s (right) [15]

The chance of presence of droplets in a certain range, can be calculated by the area under the distribution (Figure H-2). The quantile (α in %) of a property, in this case droplet size (d), is defined as the chance of having droplets sizes under a certain value is α %. For instance, d_{10} is a specific droplet size in which the chance of having droplets smaller than that value (d_{10}) is 10%.

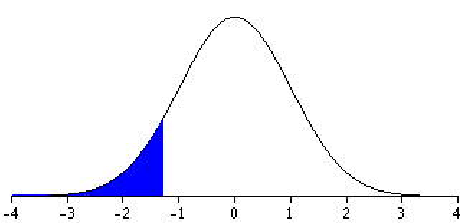


Figure H-2. The probability of an area (indicated in blue) in the distribution function

The cut off diameter of separators mainly depends on the operating pressure and density difference between gas and liquid. For axial cyclones (Figure H-3), which are the most efficient separators at high pressure conditions the cut off diameter varies from a few microns at atmospheric conditions up to some 30-40 microns at the higher gas densities.
Note that d_{99} will also follow a similar trend. However, it will be obvious that the cut off diameter will be larger in that case.

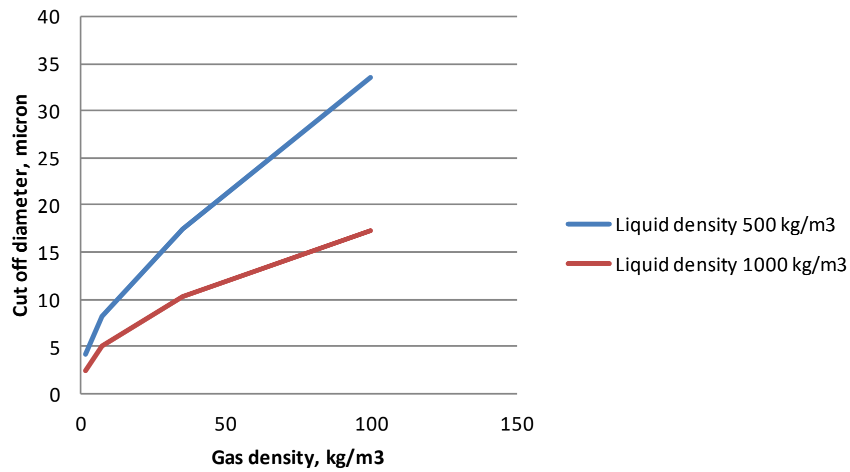


Figure H-3. Typical cut-off diameter (d_{50}) for axial cyclones versus the gas density

The same general trend will be observed for other cyclonic devices and vane packs. Note that the efficiency for droplets smaller than d_{50}/d_{99} is lower and the separation of larger droplets is more efficient.

Annex I Separator feed pipe sizing example

In this Annex, the procedure for sizing the separator feed pipe is given. Air-water mixture is used as an example. The system characteristic and operating conditions are given below.

The design conditions are summarised in the following table:

Shaft speed (N)	375 [r/min]
Feed volumetric flow rate (Q_{feed})	0.2 [m^3/s]
Connecting pipe diameter ($D_{connecting}$)	0.16 [m]
Mean absolute line pressure (P_L)	10 [bara]
Feed line temperature (T_L)	15 [˚C]
Feed liquid holdup (no-slip holdup) (λ)	0.1 [-]
Gas density (ρ_g)	11.6 [kg/m^3]
Liquid density (ρ_l)	999.2 [kg/m^3]
Sound speed of gas (c)	340 [m/s]

The feed density is:

$$\rho_{feed} = \lambda\rho_l + (1-\lambda)\rho_g = 110.36 \qquad kg/m^3$$

The first approximation for the separator feed diameter is calculated from Equation 4-14:

$$\frac{1}{D_{feed}^4} \le \frac{870}{\rho_{feed}.Q_{feed}^2} = \frac{870}{110.36 \times 0.2^2} \;\rightarrow\; D_{feed} \ge 0.266 \qquad m$$

Use $D_{feed} = 0.27\ m$. The allowable pulsation level for the current machine and feed pipe is calculated from Equations 4-5, 4-6. The first harmonic (z=1) should be used, because it corresponds to the highest allowable level for pressure pulsations:

$$f = \frac{N.z}{60} = \frac{375 \times 1}{60} = 6.25 \qquad Hz$$

$$P_1 = \sqrt{c/350}\left(\frac{400}{(P_L.D_{pipe}.f)^{0.5}}\right) = \sqrt{340/350}\left(\frac{400}{(10 \times 270 \times 6.25)^{0.5}}\right) = 3.04 \qquad \%$$

$$P' = 3.04 \times 10/100 = 0.304 \qquad bara$$

The fluctuating velocity is calculated from Equation 4-4:

$$U_G' = 0.5 \times 10^5.\frac{P'}{\rho_G c} = 0.5 \times 10^5 \times \frac{0.304}{11.6 \times 340} = 3.85 \qquad m/s$$

The maximum gas velocity and volumetric flow rate are calculated:

$$U_{G,\max} = \overline{U}_G + U_G' = \frac{Q_G}{0.25 \times \pi \times D_{feed}^2} + U' = \frac{0.2}{0.25 \times \pi \times 0.27^2} + 3.85 = 7.34 \qquad m/s$$

$$Q_{G,\max} = \overline{Q}_G \times \frac{U_{G,\max}}{\overline{U}_G} = 0.2 \times \frac{7.34}{\frac{0.2}{0.25 \times \pi \times 0.27^2}} = 0.2 \times \frac{7.34}{3.5} = 0.42 \qquad m^3/s$$

The separator feed diameter should be corrected for the maximum gas volumetric flow rate:

$$\frac{1}{D_{feed}^4} \leq \frac{870}{\rho_{feed}.Q_{feed}^2} = \frac{870}{110.36 \times 0.42^2} \rightarrow D_{feed} \geq 0.38 \qquad m$$

Notes:

1. No further iterations are necessary after the feed pipe diameter has been adjusted.
2. In case the compressor has a variable speed, the pulsation P_l as calculated with Equations 4-5 and 4-6 should be based on the minimum compressor speed.
3. The separator should be designed for all specified operation conditions on the compressor data sheet.
4. If a pulsation study according to the API standard 618 has been performed, the feed pipe diameter should be calculated from equation 0-3 based on the calculated maximum flow rate ($Q_{G,max}$).
5. This approach is not applicable in case of an acoustic resonance (in feed pipe/separator).
6. If the calculated feed diameter is smaller than the connecting pipe diameter, the diameter of the feed pipe should be changed into the diameter of the connecting pipe.
7. For variable speed machines, the lowest shaft speed should be taken in the calculation.
8. By using an inlet device, a smaller feed pipe diameter is possible. However, this should be decided in consultation with a separator expert.
9. This approach is based on the worst-case scenario, due to the fact that the lowest frequency (being one time the minimum of compressor speed) is chosen for determining the maximum allowable pulsation level.

Annex J Vibration assurance of vertical separators for reciprocating compressors

J.1. Introduction

A significant risk factor in a reciprocating compressor installation design is high vibration and cyclic stress level which may lead to fatigue failures. Separation devices, named separators for convenience in this section, are frequently found to be a concern for fatigue failure.

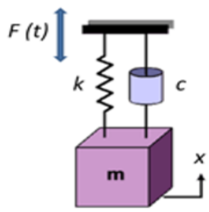
Several engineering rules are given in section 4.3.1 and 4.3.2 as to how to determine the optimal diameter and height of the separator to remove liquids in an efficient way. In general, the wire-mesh demisters, vane type demisters, cyclone packs and coalesce filters have a relatively large height/diameter ratio compared to other type of separators to account for the internal parts and to store liquids. The optimal design to achieve the highest separation efficiency (large height/diameter ratio) is often contrary with the optimal design (small height/diameter ratio) to minimize vibrations. The optimal design to achieve the highest separation efficiency can result in a slender construction which can lead to high vibration levels. This is the case if the frequency of the forces as summarised in section J.3 is close to or coincides with a mechanical resonance frequency, called Mechanical Natural Frequency (MNF).
In that case there is a risk of fatigue failure of the separator body, internal parts and attached equipment e.g. levels gauges, drain lines etc.

This annex gives some fundamentals of mechanical vibrations, a summary of the excitation forces, do’s, don’ts and engineering rules how to avoid unacceptable vibrations and fatigue failure. Although the focus in this Annex is placed on the vibrations of the vessel shell, it is noted that internals parts may be subjected to severe hydrodynamic loads, e.g. in slugging conditions, pressure surges etc. This can lead to mechanical overload and/or repetitive forces potentially leading to fatigue failure of internal parts. In general, this is an exception and requires a detailed flow and mechanical analysis which is not a part of the scope. If the separator shall be designed for these exceptional conditions, it is recommended to consult a specialist on this topic.

However, if the vibrations of the shell of the separator will be decreased, the vibrations of internals parts will also be decreased in many cases. The best way to ensure the reliability of the internal parts is to avoid at all times excitation of mechanical natural frequencies (MNF’s).

J.2. Fundamentals of mechanical vibrations

The basic principles of a complex frequency form of solution for a single degree of freedom system with viscous damping as shown in picture J-1. The equation of motion for a forced harmonic excitation is given in equation (J-1).



$$m\ddot{x} + c\dot{x} + kx = Fe^{j\omega t} \quad \text{(J-1)}$$

Figure J-1 Single Degree of Freedom System

, where x is the vibration displacement; $\dot{x}$ the vibration velocity, $\ddot{x}$ the vibration acceleration, F the excitation force, ω the angular velocity (ω is 2π times the frequency f); m the mass; c the viscous damping and k the spring stiffness.

The mechanical natural frequency (MNF), or resonance frequency, is the frequency at which an object vibrates when it is not disturbed by an excitation force. Each degree of freedom of an object has its own resonance frequency, expressed as ω_o (rad/s). This frequency ω_0 =2πf, where f is the undamped (ζ=0) resonance frequency in Hertz (Hz).

A natural frequency can be either undamped or damped. For an undamped system the resonance frequency ω_o is given by equation (J-3).

It can be derived from equation (J-1) that the solution for the vibration displacement (x) can be written in the form as given in equation (J-2). The quantity in the square brackets is the amplification factor |H(w)|. This factor is a dimensionless ratio between the amplitude of the vibration displacement (x) and the static displacement given by F/k.

$$x = \left[\frac{1}{\sqrt{\left\{1-(\omega/\omega_0)^2\right\}^2 + (2\zeta\,\omega/\omega_0)^2}} \right] \frac{F}{k} e^{j(\omega t-\theta)} \quad \text{(J-2)} \qquad \omega_0 = \sqrt{k/m} \quad \text{(J-3)}$$

The vibration displacement (x) is a function of the excitation force (F), the static stiffness (k), and the damping ratio ζ. Figure J-2 shows the amplification factor |H((ω)| as a function of the dimensionless frequency ratio of the excitation frequency ω and resonance frequency ω_0 for various values of the damping ratio ζ. If the excitation frequency equals the mechanical natural frequency (ω=ω_0), the system is called to be in resonance. At resonance the vibration and cyclic stress levels can be too high, leading to fatigue failure. For that reason, the resonance shall be avoided or shall be controlled in an adequate way.

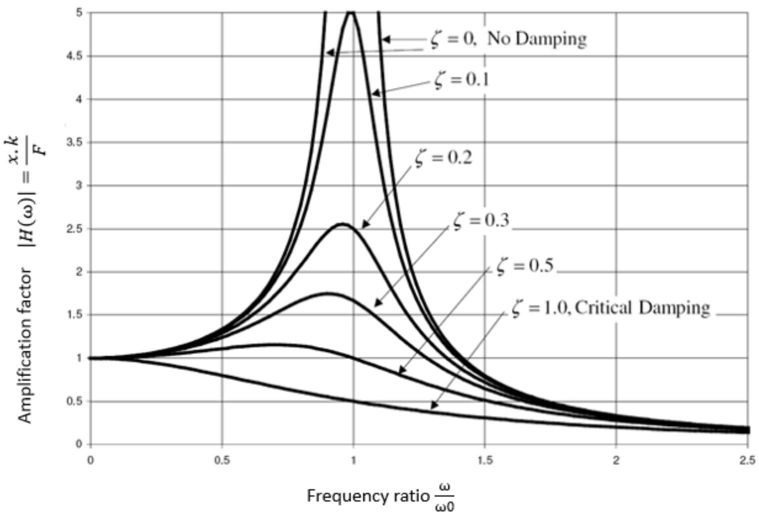


Figure J-2 Amplification factor as a function of the dimensionless frequency ratio ω/ω_0

It can further be concluded from Figure J-2 that increasing the damping ratio ζ, tends to diminish the amplitudes and shifts the peaks to the left of the vertical through $\omega/\omega_0 = 1$.
For lightly damped systems ($\zeta<0.05$), which is typically the case for (parts of) compressor systems, the curves are nearly symmetric around the vertical through $\omega/\omega_0 = 1$. It can be derived that the amplification factor $|H(\omega)|$ for lightly damped systems is according equation (J-4). This means that the vibration amplitude at resonance is mainly determined by the damping ratio ζ.

$$|H(\omega)| \approx \frac{1}{2\zeta} \quad \text{(J-4)}$$

If the excitation frequency ω is lower than the resonance frequency ω_0, the system is called a detuned system and the vibration amplitude is mainly determined by the static stiffness. If the excitation frequency is higher than the resonance frequency, the system is called an over-tuned system and the vibration amplitude is mainly determined by the mass. More detailed explanation on this is given in section J-4.

J.3. Summary of dynamic forces

The dynamic forces, sometimes referred to as loads, generated by a typical engine drive reciprocating compressor are shown graphically in Figure J-3. All these loads must be considered in the design of a separator to minimize vibrations and the risk on fatigue failure. The most important dynamic forces in terms of separator vibration are typically the compressor and driver unbalanced loads, crosshead guide forces, cylinder gas forces and pulsation shaking forces. The engine rolling torque and lateral torsional forces are normally not an important factor in separator vibration.

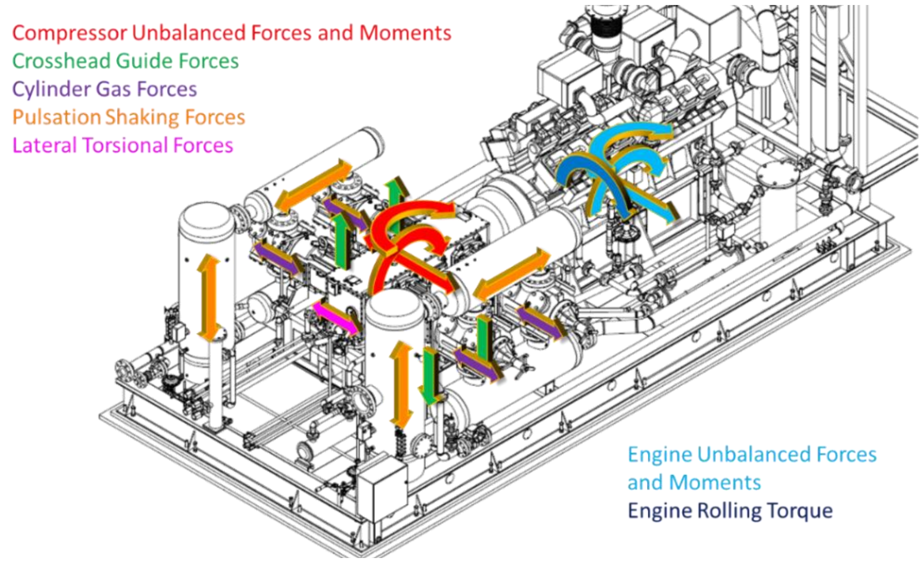


Figure J-3 Dynamic Forces in an engine driven reciprocating compressor

J.3.1. Unbalanced loads of compressor and driver

Dynamic forces and moments from unbalanced reciprocating and rotating inertias are present in reciprocating compressors and engines. These unbalanced forces and moments are often the highest dynamic loads. These loads are the results of the machinery design and the tolerances in the weight of the reciprocating and rotating components. These loads are present even when the compressor is fully unloaded. In general, there is very little that can be done to minimize these loads for a compressor design.

In general, these forces can excite MNF's of the separator and attached equipment such as level gauges, vent and drain lines if the vibrations of the compressor are transferred to the separator via a flexible structure. The transfer of these vibrations is named structure borne vibrations and can occur if the separator and compressor are mounted on the same flexible structure e.g. a skid or on the deck of an off-shore platform. An example of the unbalanced loads is shown in Figure J-4. These loads are normally provided by the compressor manufacturer.

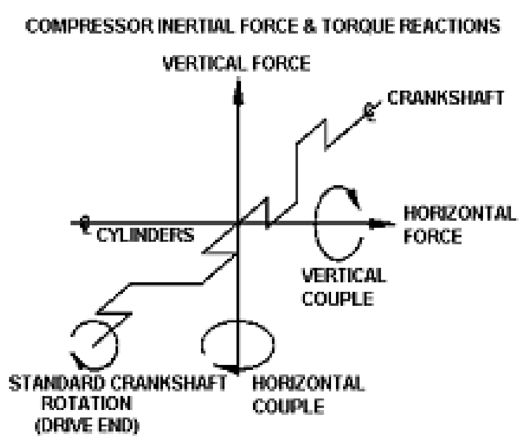


Figure J-4 Example of unbalanced loads of a reciprocating machine

J.3.1. Crosshead guide forces and cylinder stretch forces

Crosshead guide forces and cylinder gas (stretch) forces are primarily the result of compressing the gas and moving masses of the piston and piston rod. An example of the crosshead forces and the cylinder gas forces are shown in respectively Figure J-5 and J-6. Especially the vertical crosshead forces can generate structure borne vibrations.

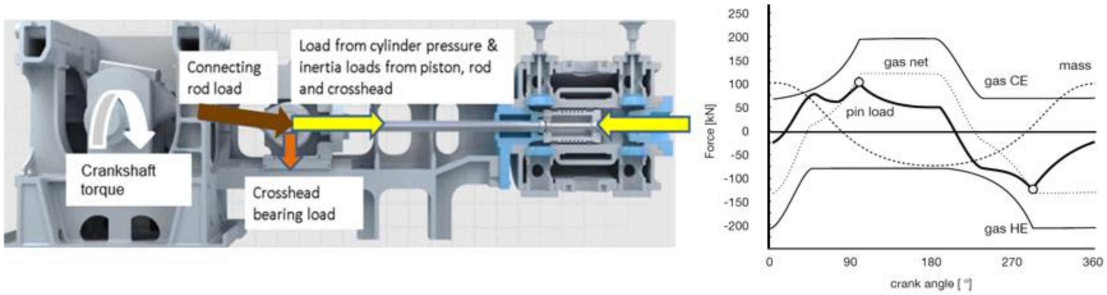


Figure J-5 Example of crosshead guide forces

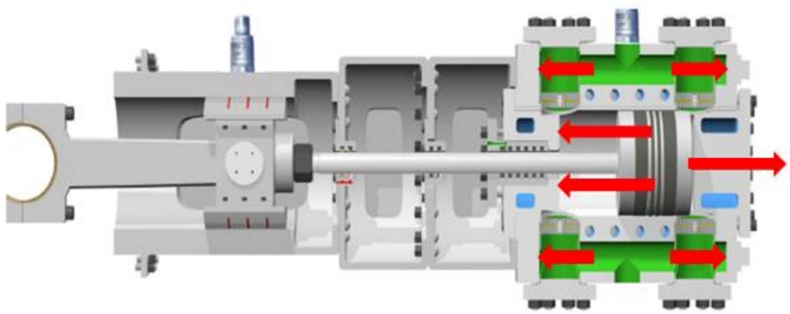


Figure J-6 Example of gas forces acting on cylinder

Operating factors such as the compression ratio, cylinder loading, and gas composition impact these forces for a given compressor geometry. In general, they vary greatly in amplitude and frequency with changes in the operating factors. Typically, it is not possible to lower the crosshead and cylinder stretch forces by changing the operating factors as the pressure and flow requirements for the compressor application must be met. The frequency and phase of the crosshead and cylinder stretch forces will change with operating factors such as cylinder loading (clearances) so it may be possible to avoid some situations where there is high load by avoiding certain cylinder load steps.

Identical to the unbalanced loads, the crosshead guide forces can excite separator MNF's and attached equipment. This is the case if the vibrations caused by the crosshead forces are transferred to the separator via a flexible structure on which the compressor is mounted, e.g. the compressor skid. The cylinder vibrations are most commonly transferred to the separator via the suction pulsation damper and the pipe between the suction pulsation damper and the separator.

J. 3.2. Pulsation-induced shaking forces

Pulsation-induced shaking forces are the result of pressure pulsations generated by the reciprocating compressor. A design study according e.g. to the API Standard 618 [1], to design the pulsation dampers, piping and other components can reduce these pulsation-induced shaking forces. However, reducing the pulsation-induced shaking forces to zero is not practical. There will be some residual pulsation-induced shaking forces that must be considered in the final design.

J.4 Design goals to minimize vibrations and the risk on fatigue failures

J.4.1. Introduction
Minimizing vibrations requires minimizing the excitation forces, increasing the stiffness, increasing the damping or a combination. Minimizing the excitation forces is in most of the times only possible for the pulsation-induced shaking forces as explained in section J.3.2. The API Standard 618 [1] and the API RP 688 [17] include guidelines for reducing the pulsation-induced shaking forces and increasing the stiffness of the pipe system. Pulsation-induced shaking force guidelines are given for pulsation bottles and piping. However, there is no guideline for pulsation-induced shaking forces in separators.

Increasing the damping ratio will have the largest effect on vibrations at resonance conditions. However, increasing the damping ratio is not a straight forward solution for most of the systems. The most effective method is therefore by shifting the resonance frequencies far enough from the frequency of the excitation forces.

One approach to minimize the transfer of the dynamic loads (free forces and moments, crosshead guide forces and cylinder vibrations), which are transferred from the compressor to the separator, is to remotely locate the separator from the compressor. However, the definition of "remotely locate" is difficult to quantify. In general, it means that the separator is far enough away from the compressor, so the dynamic loads are not transmitted through the piping, skid or foundation to the separator. One reference [23] states that a component can be considered "remote" when the component has at least 6 meters of piping between it and the bottle flange. This length is a reasonable criterion if the separator is located on a skid and foundation that is independent from the reciprocating compressor.

It is emphasized that the amplitude of the pulsation-induced shaking force in the piping between the damper and separator strongly depends on the length of the piping. The pulsation-induced shaking force on this pipe might be larger if the separator is installed remote from the compressor. The pulsation-induced shaking forces shall be calculated with a pulsation analysis according to e.g. the API Standard 618 [1].

J.4.2. Detuning and over-tuning
Minimizing separator vibrations and the risk on fatigue failure can be achieved in most of the cases by increasing the stiffness of the separator structure so that the excitation frequency does not coincide with a mechanical resonance frequency. This is also one of the requirements as stipulated in the API Standard 618 [1]. It states that the lowest MNF shall be minimum 20% above two times the maximum compressor speed (2.4X) and shall be separated minimum 20% from the frequency of the dominant excitation forces. This is to ensure that the frequency of the dominant forces, which are in general the free forces and moments and the pulsation-induced shaking forces, will not coincide with the lowest MNF's of the separator.

Over-tuning the separator mechanical natural frequency to meet the minimum guideline according to the API 618 [1] and the API 688 [17] of 2.4 times the compressor speed (2.4X) will generally result in acceptable separator vibrations and a low risk of fatigue failure.

A separator MNF that is tuned between the fundamental (one times compressor speed, 1X) and two times the compressors speed (2X) is said to be inter-tuned. A separator MNF that is tuned below the fundamental speed is said to be detuned. The risk with the inter-tuned separator design is that the static stiffness and damping ratio may be too low resulting in unacceptable vibrations due to normal dynamic loads.
The risk with the detuned design is that there may more mechanical natural frequencies that are lowered and become resonant at two times the compressor speed by detuning the system. The

example as shown in Figure J-7 demonstrates a single mechanical natural frequency for a compressor with a speed of 1200 rpm.

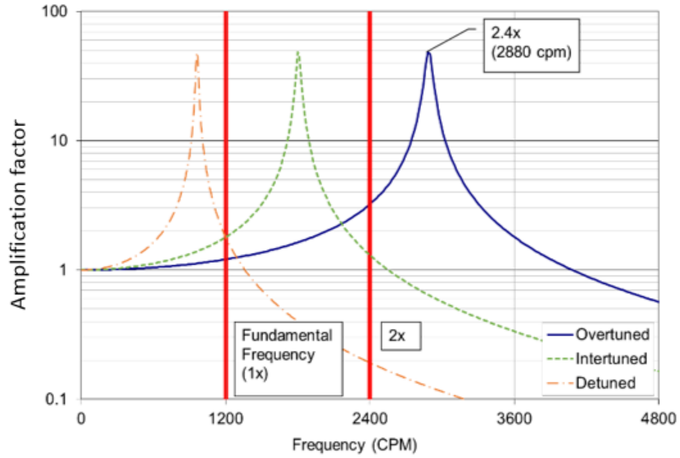


Figure J-7 Separator Mechanical Natural Frequency Tuning

J.4.3. Variable speed compressor

Ensuring a 20% separation margin is more difficult, and often not practical, for variable speed reciprocating compressors. Consider a case where the compressor has an electric motor Variable Frequency Drive (VFD) operating from 600 to 1200 rpm. The required frequency range for the fundamental frequency is 600-1200 rpm and 1200-2400 rpm for two times the compressor speed. There is no frequency between the fundamental and two times the compressor speed where the separator MNF can be inter-tuned. This is the red area as shown in Figure J-8 . It is still possible to detune the separator, but the frequency will be very low resulting in either a very low static stiffness (or heavy mass) or an increased risk of a second MNF being lowered into a resonance condition. Designing separators to minimize vibration in a variable speed application is a significant challenge.

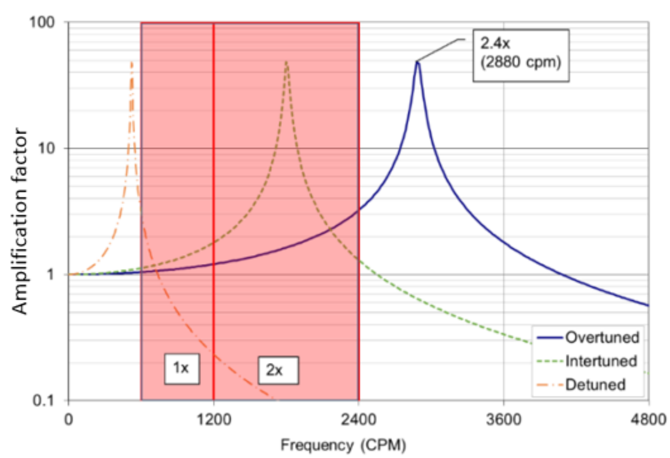


Figure J-8 Detuning a separator for a variable speed compressor

J.4.4. Mechanical response analysis

The separator mechanical natural frequency is an indicator of the potential for high vibration due to resonance. A separator may be resonant at 1x or 2x compressor speed, but the vibration and cyclic stress levels will exceed the allowable level. The separator vibration strongly depends on the damping ratio and stiffness. A mechanical response analysis is required to calculate if the maximum vibration and cyclic stress levels do not exceed the allowable levels. In the API Standard 618 [1] and the API RP 688 [17] a detailed description is given as to how to carry out such an analysis. The allowable vibration and cyclic stress levels are also given in these API documents.

J.4.5. Attached equipment to separators

The previous sections were focussing on frequency avoidance and tuning the separator vessel's MNF to achieve acceptable vibrations and to avoid risk of fatigue failure. Another significant risk for fatigue failure on a separator includes the instrumentation and other small diameter nozzles that are included on the separator as part of its normal operating requirements. Even in the case of acceptable vibrations of the separator shell, the attachments on the separator may experience high vibrations and the risk on fatigue failure. These connections may include:

- nozzles with blinds that are used for inspection,
- liquid level sight glasses or gauges, liquid level sensors,
- relief valve connections,
- temperature gauges, and
- liquid drains.

Figure J-9 shows an example of the attachments that are frequently installed on a separator which is mounted on a reciprocating compressor package. These attachments can have excessive vibration at frequencies that are above the 2.4 times the separation margin. This is that case if there are MNF's of attachments above 2.4 times the separation margin that are excited by forces with a frequencies higher than 2 times the maximum compressor speed.

Figure J-9 Example of Separator Attachments

The main dynamic forces that causes vibration of the separator attachments are pulsation-induced shaking forces in the pipe between the damper and separator and the compressor cylinder gas forces. The pulsation-induced shaking forces can be reduced with proper pulsation design practices. However, the cylinder gas forces cannot be reduced in amplitude with design practices like the pulsation shaking forces. The design of the separator attachments must consider the cylinder gas forces as vibration excitation sources.

The cylinder gas forces occur at all multiples of compressor speed, so frequency avoidance is not always easy to achieve and in many cases even not practical, particularly for variable speed compressors. Figure J-10 shows an example of the cylinder gas forces for multiples of 1X to 5X for a range of operating points. An operating point represents a pressure ratio, suction temperature, gas composition and cylinder load step. From Figure J-10 it can is shown that the cylinder gas forces are high at 2X, 3X and 4X compressor speed. The dynamic stiffness of the separator attachments shall be increased considerably to avoid resonance at these multiples of the compressor speed. to ensure that the vibrations levels will not exceed allowable levels.

It is also important to note how these forces change depending on the operating point. A separator attachment that is resonant at 3X compressor speed may have acceptable vibrations when operating at point 10. However, the cylinder gas forces at 3X will be much higher for operating point 1 as shown in Figure J-10 Vibrations will increase and may be excessive when operating at point 1. This change in cylinder gas forces must be considered when assessing vibrations on reciprocating compressor packages at either the design stage or when taking site measurements of vibration.

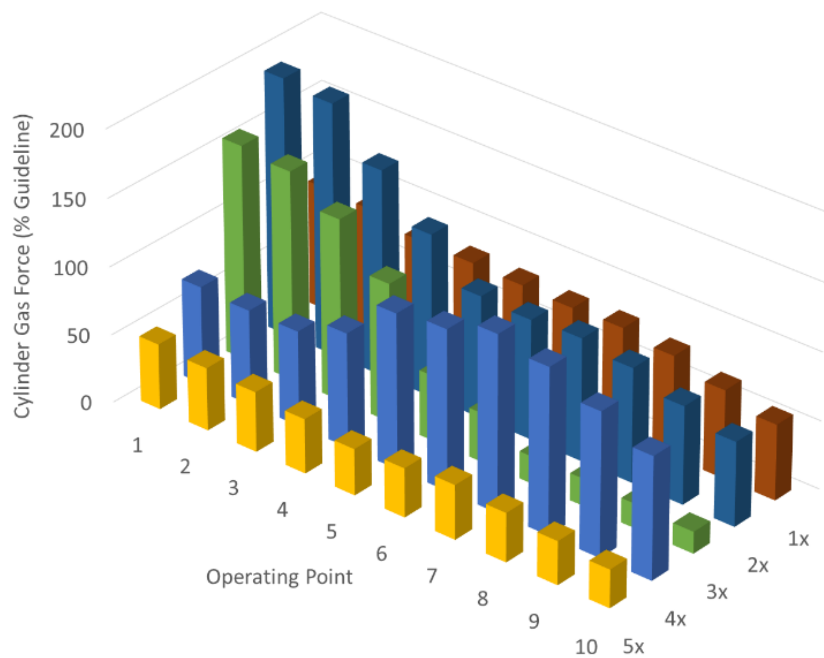


Figure J-10 Example of compressor cylinder gas forces

The challenge to avoid fatigue failures on separator attachments is to increase the dynamic stiffness of these attachments so that the MNF's are higher than the frequencies where the response (vibrations) due to cylinder gas forces is significant. The separator connections may include heavy flanged valves or include redundant valves and instrumentation that have a long projection from the vessel shell. The combination of the heavy weight and long projection results in significantly lower dynamic stiffness. There are many best practices that will reduce the likelihood of failure on the separator attachments that will be outlined in the following section.

J.5. Industry best practices

This section summarizes the most important design procedures and best practices for minimizing vibrations and the risk on fatigue failure on the separator shell.

J.5.1. Pre-selection of separator

Section 4.2 describes the process of the pre-selection of a separator for handling liquids in the gas stream. The pre-selection of the separator design and general vessel sizes shall also consider factors that will minimize vibration. Section J.4 described the impact of having a separator close-coupled to the reciprocating compressor versus remotely located as well as installing the separator in an application that is fixed speed versus variable.

These and other factors need to be considered in the pre-selection of the separator. For factors controlling the separator's vibration it is recommended to consult an engineering specialist company in an early stage of the design process.

A typical project timeline is shown in Figure J-11. A project begins with a planning or FEED (Front End Engineering and Design) stage where the initial specification and general equipment selections are done. The pre-selection of the separator design is often done in this phase. The detailed design phase is then started if approval is given. Equipment suppliers and packagers will be awarded contracts and begin their detailed design. These suppliers will hire vibration consultants and other engineering specialist to help with the detailed engineering. It is normally at this point in the design where vibration problems with the separator are identified. However, the equipment package design has progressed, and budgets have been set. Making a significant change in the separator design to avoid a vibration problem is in general very difficult or not feasible anymore. Remedial measures to control vibration may result in a less than desirable final design. For example, large braces may be needed to minimize separator vibration. The braces can restrict access for maintenance and pose safety hazards.

The recommended timeline for involvement of the vibration consultant is illustrated in Figure J-12. The vibration consultant should have input very early in the project planning stage. Pre-selection of the separator design with consideration for vibration control will avoid costly delays and redesign later in the project. A second important factor that shall be considered in the pre-selection of the separator is achieving a balance of roles and responsibilities between the vendor and purchaser. Decisions to minimize separator vibration may increase capital costs or result in layout changes. The purchaser or owner must be involved in making key decisions that will impact the life-cycle costs of the equipment.

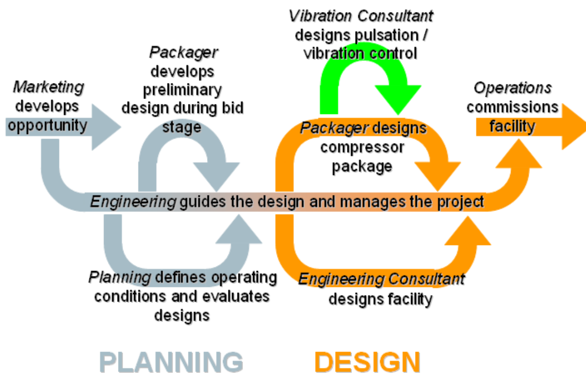


Figure J-11 Typical project timeline

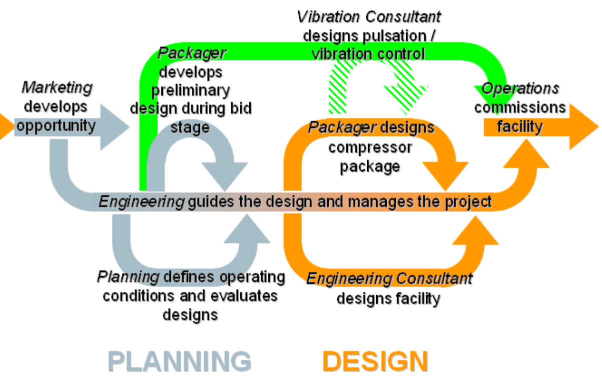


Figure J-12 Recommended project timeline

J.5.2. Separator Base Design

The design of the base for the separator is one key to over-tuning the separator MNF. Separators may be mounted directly to a concrete foundation or steel skid. Figure J-13 shows a typical configuration for a separator which is mounted on a concrete/grout structure. Figure J-14 shows a typical base design of a separator mounted on a skid.

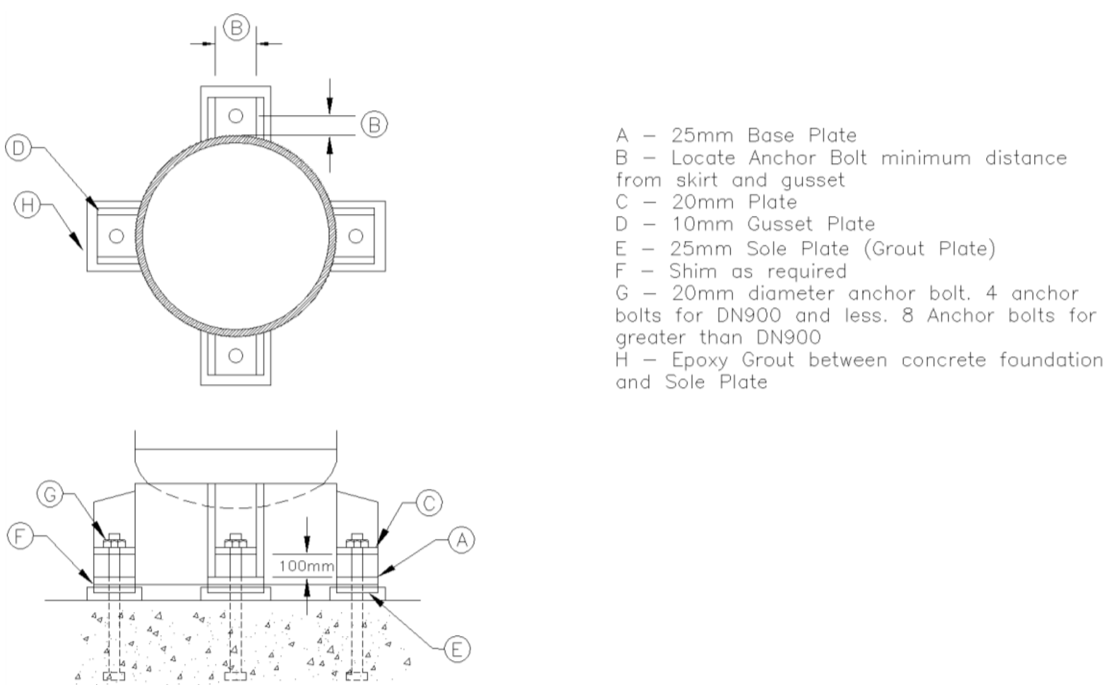


Figure J-13 Separator base for mounting on a concrete foundation

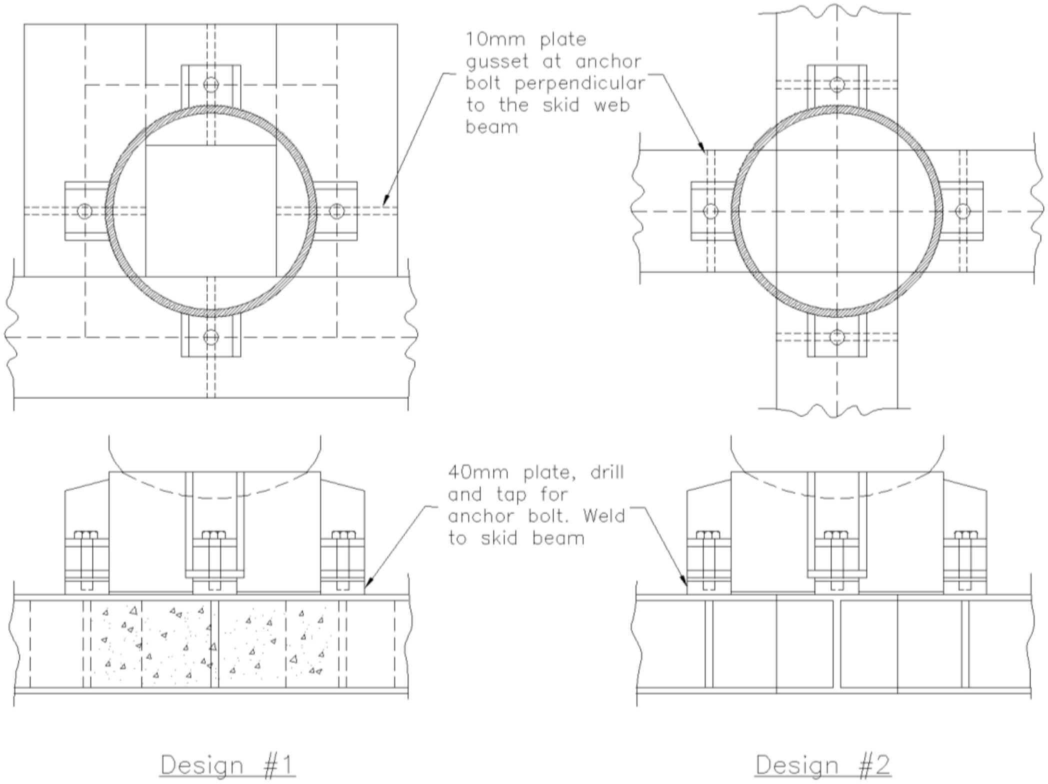


Figure J-14 Separator base for mounting on a skid

The following recommendations are important in the base design:

1. If separators are mounted on a concrete foundation it is recommended to use an epoxy grouting on top of the concrete. This improves the accuracy of the lining and will also avoid that oil, water etc. will intrude into the concrete and will avoids cracks.
2. Shims are recommended between the base plate and the sole plate attached to the foundation or skid. The shims are necessary to ensure there is full contact and transfer of stiffness between foundation and the separator.
3. A minimum free bolt length of 8 times the bolt diameter is recommended to lower the fatigue stress in the bolts.
4. A recommended bolt preload of 50-70% of the yield strength is recommended to avoid that bolts will become loose and it will also lower the risk on bolt fatigue.
5. If the separator is to be mounted on a steel skid, it is preferred to weld the separator skirt directly to the skid beams rather than using bolts. Welding the skirt to the skid beams ensures a direct transfer of stiffness. However, welding the skirt to the skid is not always possible due to requirements for shipping or maintenance.
6. For skid mounted separators, the arrangement of the beams below the separator is important in the base stiffness. An arrangement as shown in Design #2 of Figure J-14 is acceptable for cases where the separator skirt is welded to the skid beams. The weld will directly connect the web of the skid beam to the skirt. Alternatively, a “box” arrangement of beams can be used as shown in Figure J-15. The skid beam webs should be spaced at approximately 90% of the outside diameter (OD) of the separator to maximize the skid stiffness.

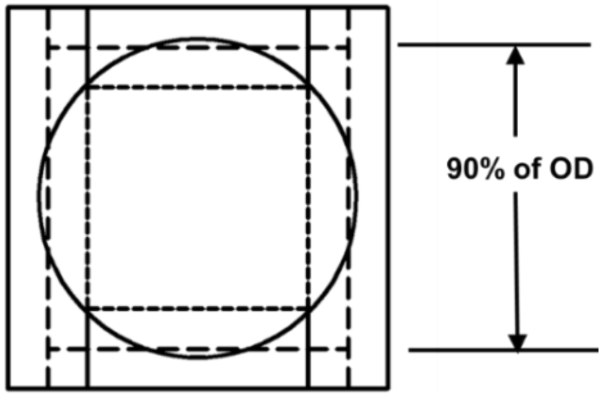


Figure J-15 Skid Beam Arrangement

7. A full skirt is the preferred method of supporting a separator in a reciprocating compressor application. General guidelines of the skirt are as follows:

- Avoid using a separator base fabricated from structural angle sections for reciprocating compressor systems. This construction can lead to fatigue failures as shown in Figure J-16. Apply as much as possible a full skirt and keep openings to a minimum.

Figure J-16 Separator Base with too flexible leg supports

- Typically, the skirt outside diameter and wall thickness shall be the same as the main vessel.
- For high pressure service (> 100 bar) where a very thick wall vessel is required, it is recommended to apply a minimum skirt wall thickness of 25 mm.
- The legs are often too flexible resulting in high vibration that can lead to cracking of the structural section as shown in Figure J-16.
- It is recommended to evaluate the skirt/base design during the detailed design phase. This can be done e.g. with a detailed finite element model of the separator base including the skirt, base plate and (a part of) the skid. An example is shown in Figure J-17. It is noted that assuming the separator base rigid for a skid mounted separator will overestimate the MNF's and puts the separator design at risk of high vibration and fatigue failure, especially of the attachments.

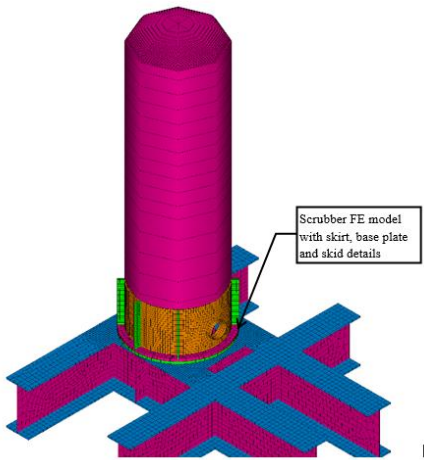


Figure J-17 Example of a separator base finite element model

J.5.3. Separator shell sizing

In section 4.3.1 engineering rules are given how to determine respectively the diameter and height of the vertical separator to achieve an optimal performance of the separator w.r.t. removing liquid from the gas stream. The selection of the diameter and height shall not be contrary with the “over-tune” design philosophy of the separator to avoid unacceptable vibrations.

The separator mechanical natural frequency (MNF) is proportional to the square of the outer diameter (OD), divided by the height (h), ignoring the effect of wall thickness, as represented in equation (J-5).

$$\boldsymbol{MNF} \propto \left(\frac{\boldsymbol{OD}}{\boldsymbol{h}}\right)^2 \quad \text{(J-5)}$$

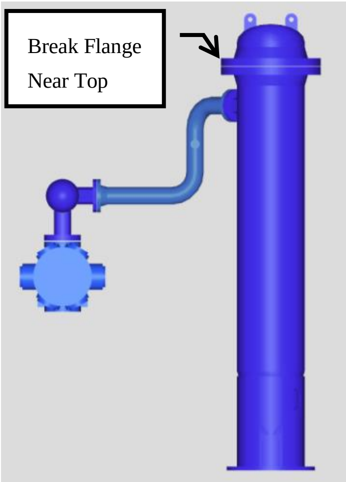
Slight changes in the diameter and height can have a significant impact on the MNF. Separators on high speed reciprocating compressor packages are often at risk of not being over-tuned as meeting the over-tuning criterion of 2.4X is a challenge. Typical design values are:

- Typically, a ratio of diameter to height of more than 0.2 to 0.25 will result in an over-tuned separator for compressor speeds from 1000 to 1200 rpm.
- The diameter to height ratio of 0.12 may be acceptable for low speed reciprocating compressors with a speed of 300-360 rpm. Note that the previously stated ratio assumes that the separator does not include a large manway, access flange and blind to changing vessel internals. Equation (J-5) further assumes that the separator has a suitable base as described in the previous sections and is fixed to a stiff supporting structure (concrete or skid). A larger diameter to height ratio may be required for installations that do not meet good design practices for vibration control in other areas.
- In general separators installed on the suction side of the reciprocating compressor system may have a relative thin wall thickness due to the low suction pressure. Thin wall vessels may be of risk for high frequency vibrations of the shell or attachments on the shell. A minimum shell thickness of 13 mm is recommended as a standard requirement [24]. The minimum 13 mm wall thickness requirement is also applied to stainless steel shell material where the pressure code requirement would be much less.

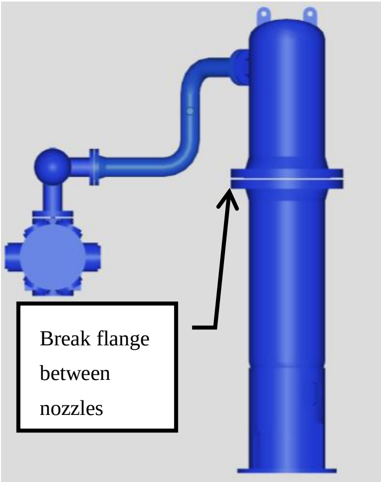
J.5.4. Contrary "liquid removal" and "vibration" design
If the values of the wall thickness and the height of the "liquid removal" design as summarized in section 4.3.1 and 4.3.2, and the detuned or over-tuned design are contrary, a decision shall be made by the vendor and purchaser about the preferred design. A field vibration test is recommended after start-up and commissioning of the compressor installation to check the separator vibration in cases where the vibration control design philosophy is not followed. Refer to the ISO 10816-8 standard for criteria for field vibration assessment.

J.5.5. Service considerations
Some separator designs require inspection or maintenance of the internal components. Typically access to the internal components is by a blind flange or flange set located at the top of the vessel. The flange set can be very heavy. The combination of the large mass and location at the top of the separator results in a very low MNF. Selecting a different separator design that does not require inspection or maintenance of the internal components shall be considered if the application will allow.

One option to consider is to lower the flange set as shown in Figure J-18. Lowering the flange set will significantly increase the dynamic stiffness. The separator shell OD/h ratio could also be increased beyond what is normally required when a flange for access is required. However, this shall not contrary with the design for an efficient removing of the liquid.



Original Design



Recommended Design

Figure J-18 Separator break flange locations

J.5.6. Instrumentation
The most common components that experience fatigue failures on separators are the small diameter attachments, that is, the sight glass, bridle, drip pot, temperature gauge and level transmitters. Generally, these components are most at risk when the separator is closely coupled to the compressor. Vibration is caused by a combination of low dynamic stiffness and significant levels of cylinder gas forces at frequencies above two times the compressor speed.

The low dynamic stiffness is typically the result of a large suspended mass extending a significant distance away from the vessel. The Gas Machinery Research Council [25] and the Energy Institute [26] have published guidelines for the design and evaluation of small diameter connections that can reduce failures of separator attachments. General recommendations that will minimize the potential for high vibrations and failures are as follows:

1. Minimize the projection of the nozzle from the shell by using studding outlets or RFLWN (Raised Face Long Weld Neck) which are flanges trimmed to a minimum length rather than RFWN (raised face weld neck) flanges.
2. Eliminate isolation valves between the nozzle and the attachment. If the attachment must be isolated, use either:
   a. Mono flange valves or,
   b. Mount the attachment directly on the separator shell or on a separate support column. Install a small male/female threaded, full port needle valve on nozzle. Use tubing or braided hose from the valve to the remotely located attachment.
3. Install a reinforcing pad and gussets in two planes for RFWN (Raised Face Welding Neck) nozzles. The nozzle must be stress relieved after welding.
4. Orientating the nozzle and attachments parallel to the direction of piston motion. Welding pads on the separator to allow for field braces are required.
5. Figure J-19 shows two acceptable arrangements for the sight glasses or level gauge. The sight glass shall be mounted as close to the separator as possible as shown by the studding outlet arrangement on the left. Some separator designs include a special mounting bracket on the separator shell and tubing or steel braid flex hose between the nozzle and sight glass. A remote mounted sight glass design is shown on the right which have relatively short and lightweight ball valves are used on the separator for isolation.

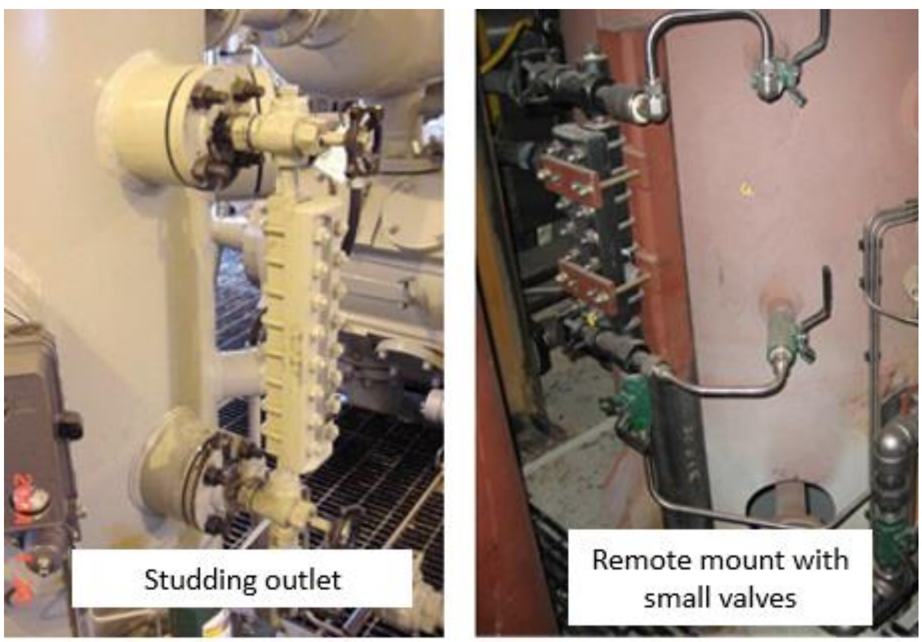


Figure J-19 Separator Sight Glass Layout Options

Bibliography

[1] API Standard 618, 5th edition 2007, Reciprocating compressors for petroleum, chemical, and gas industry services

[2] Amsitco catalogue for sizing mesh and vane mist eliminators.

[3] Specification for Oil and Gas Separators, API Specification 12 J, 8th edition, 2009.

[4] H. P. Bloch, J. J. Hoefner, Reciprocating Compressors, Operation and Maintenance. Gulf Professional Publishing, 1996.

[5] S. P. C. Belfroid, G. J. N. Alberts, Separation efficiency in a pulsation flow, EFRC report, 2005.

[6] B. Kalis, Compressor suction drum, Amistco reports, 2004.

[7] M. Souders, G. G. Brown, Design of fractionating columns, entrainment and capacity, Industrial and Engineering chemistry, 38(1): 98-103, 1934.

[8] R. A. Swanborn, A new approach to the design of gas-liquid separators for the oil industry, PhD dissertation, 1988.

[9] P. Simmons, B. Nesbitt, D. Searle, Guide to European compressors and their applications, Professional Engineering Publishing, 2003.

[10] Sulzer catalogue (Sulzer chemtech) on liquid-liquid separation technology (http://www.sulzer.com/en/media/Documents/ProductsAndServices/Separation_Technology/Coalescers/Brochures/Liquid_Liquid_Separation_Technology.pdf)

[11] Ishii, Short courses Modelling and Computation of Multiphase flows, Zurich, 2003.

[12] J.M. Mandhane, G. A. Gregory, K. Aziz, A flow pattern map for gas-liquid flow in horizontal pipes, IJMF, 1, 537-551,1974.

[13] Y. Taitel, D. Barnea, A. E. Dukler, Modelling flow pattern transitions for steady upward gas-liquid flow in vertical tubes, AIChE J, 26, 345-354, 1980.

[14] Amistco, Mesh & Vane mist eliminators, Technical Bulletin 106, 2004.

[15] M.J.H. Simmons, T.J. Hanratty, droplet size measurements in horizontal annular gas-liquid flow, International Journal of Multiphase Flow 27, pp. 861-883, 2001.

[16] Coulson & Richardson's "Chemical engineering" Vol.6, 2nd ed., Pergamon Press, Oxford, 1993.

[17] API RP 688, 1st edition April 2012, “Pulsation and Vibration Control in Positive Displacement Machinery Systems for Petroleum, Petrochemical, and Natural Gas Industry Services

Informative references

[18] C. Verlaan, Performance of novel mist eliminators, PhD thesis, TU Delft, 1991.

[19] M. I. Comyn, F. Boer, R. Garza, J. S. Hansen, I. Bjorkevol, G. Wiken, Reliability experienced with offshore reinjection compressors, ,Proceeding of the 3rd EFRC conference, Vienna, 2003.

[20] F. Simpson, G. Lis, Liquid slugging measurements in reciprocating compressors. Proceeding of International Compressor Conference at Purdue, volume 2, pages 479–487, 1988.

[21] B. G. Shiva Parsad, Effect of Liquid on a Reciprocating Compressor, Journal of Energy Resources Technology, Vol. 124, 2002.

[22] T. Austrheim, L. H. Gjertsen, A. C. Hoffmann, Is the Souders-Brown equation sufficient for scrubber design? An experimental investigation at elevated pressure with hydrocarbon fluids, Chemical engineering science, 62(21): 5715-5727, 2007.

[23] GMRC, "Design Guideline for Small Diameter Branch Connections," GMRC , Dallas, 2011.

[24] GMRC, "Guideline For High-Speed Reciprocating Compressor Packages For Natural Gas Transmission & Storage Applications," GMRC, Dallas, 2013.

[25] GMRC, "Design Guideline for Small Diameter Branch Connections," GMRC , Dallas, 2011.

[26] E. Institute, "Guidelines for avoidance of vibration induced fatigue failure in process pipework," Energy Institute, London, 2008.