Carbon dioxide compression with rotodynamic machines

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CONTENT

1_Introduction

2_ Compression issues

- 2.1_Single phase flow Sonic and choke characteristics
- 2.2_Two phase flow Sonic characteristics
- 2.3_Hydrates
- 2.4_Flow surging
- 2.5_Compressor operability
- 2.6_Materials
- 3_ Thermodynamic properties
- 4_ One compression and injection application case
- 5_ Two phase flow compression
- 6_ Conclusion

1_Introduction

Compression of carbon dioxide may present some difficulties in various areas. Carbon dioxide does not behave as an ideal gas; therefore, good knowledge of its thermodynamic properties is required. Attention should be paid to the relative gas velocity at the blade edges to avoid sonic losses and inside hydraulic cell channels to limit choke losses. Attention is required at low temperature to avoid the formation of carbonate hydrates and not to cool the gas at a too low temperature to avoid liquid formation especially during the compression of the fluid in a single phase condition. Above the critical point, the fluid behaves in terms of compressibility in between a gas and a liquid phase; therefore, hydraulic cells with adapted internal geometry are required to get an optimum pressure coefficient. Operability of a compression train may be of concern in terms of flow and running speed range around the design point. Special materials are required to take into account the aggressiveness of wet carbon dioxide (sour gas), the presence of hydrogen sulphide and also the occurrence of very low temperature following a sudden depressurisation of the compression facilities.

The benefit of a two phase compression system is sometimes questioned. This document investigates on the relative advantage of this mode of compression.

2_ Compression issues

2.1_Single phase flow - Sonic and choke characteristics

2.1.1_ Sonic effect characteristics

The speed of sound is the distance travelled per unit time by a <u>sound wave</u> as it propagates through an <u>elastic</u> medium. Its value depends strongly on the temperature as well as the medium through which the sound wave propagates. At 20 C, the speed of sound in air is about 343 metres per second.

Sound travels relatively slowly in gases, faster in liquids and considerably much faster in solids. For example, sound travels at 1,481 m/s in water (almost 4.3 times as fast as in air) and 5,120 m/s in iron (almost 15 times as fast as in air).

In fluid dynamics, the speed of sound in a fluid medium (gas or liquid) is used as a relative measure for the speed of an object moving through the medium. The ratio of the speed of an object to the speed of sound in the fluid is called the object's Mach Number. Objects moving at speeds greater than *Mach1* are said to be traveling at <u>supersonic</u> speeds.

Concerning rotodynamic compressors, internals have to be designed according to the local relative gas velocity to limit losses occurring when this gas velocity approaches the speed of sound. These losses are designated by sonic losses.

The speed of sound "C" is given by the Newton-Laplace equation:

$$C = \sqrt{\frac{K_s}{\rho}}$$

where K_s is a coefficient of stiffness (the modulis of bulk elasticity for gases) and ρ is the volumetric mass of the gas. Following this equation it may also be written as:

$$C = \sqrt{\left(\frac{\partial P}{\partial \rho}\right)} s$$

Where P is the pressure and s the entropy. This formula indicates that the derivative applies at constant entropy. It is then finally expressed for an ideal gas by:

$$C \ ideal = \sqrt{\gamma \frac{P}{\rho}}$$
 also $C \ ideal = \sqrt{\frac{\gamma RT}{Mw}}$

Where R is the molar gas constant (8 314 J/°K/Mole) and Mw is the molar mass of the gas. For a real gas $C \ real = \sqrt{\frac{\gamma ZRT}{Mw}}$ where Z is the compressibility factor.

For further details see: <u>https://en.wikipedia.org/wiki/Speed_of_sound</u>

The sound velocity for carbon dioxide is represented on figure 2.1 versus the temperature (abscissa) and for several pressure values.



Figure 2.1 – Sound velocity for carbon dioxide in a gas phase

The graph shows that the sound velocity reduces considerably with the temperature. For instance, at 25 bar abs, it is 200 m/s at minus 20 °C and 320 m/s at 170 °C. The sound velocity varies to the opposite of the pressure with a variation particularly important at low temperature but relatively small at high temperature.

The sound velocity is approximately equal to 270 m/s at the critical point (circle symbol on the graph - 73 bar in the legend).

Attention must therefore be paid when operating a compressor with carbon dioxide at low temperature and medium pressure. At 50 bar, the sound velocity is increased approximately from 210 m/s at 20 $^{\circ}$ C to 245 m/s at 50 $^{\circ}$ C.

2.1.2_ Choke flow

Choked flow is a compressible flow effect. When it occurs, the fluid velocity (the flow rate) is considerably limited.

Choked flow is a fluid dynamic associated with the <u>venturi effect</u>. When a flowing fluid at a given pressure and temperature passes through a constriction (such as the throat of a convergent – divergent nozzle, a valve, an orifice plate or the channel of the rotodynamic cell of a compressor) into a lower pressure environment, the fluid velocity increases. At initially subsonic upstream conditions, the <u>conservation of mass</u> principle requires the fluid velocity to increase as it flows through the smaller cross-sectional area of the constriction. At the same time, the venturi effect causes the static pressure, therefore the density, to decrease at the constriction. Choked flow is a limiting condition where the mass flow will not increase with a further decrease in the downstream pressure environment for a fixed upstream pressure and temperature.

For homogeneous fluids, the physical point at which the choking occurs, for adiabatic conditions, is when the exit plane velocity is at sonic condition; i.e., at a <u>Mach number</u> of 1. At choked flow, the mass flow rate can be increased only by increasing density upstream the choke point.

Assuming ideal gas behaviour, steady-state choked flow occurs when the downstream pressure falls below a critical value P*. That critical value can be calculated from the dimensionless critical pressure ratio equation:

$$\frac{P*}{P0} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$$

Where γ is the isentropic factor or the heat capacity ratio Cp/Cv (case of an ideal gas) of the gas and P0 the total (stagnation) upstream pressure.

For a gas with a heat capacity ratio of 1.4 (nitrogen, for instance) then $P^*=0.528$ Po. For a heat capacity ratio varying between 1.09 (heavy hydrocarbon) and 1.67 (monoatomic gas) the critical pressure ratio varies in the range $0.487 < P^*/P0 < 0.587$. In the case of carbon dioxide, the isentropic factor varies with the pressure and the temperature. See thermodynamic section.

When the gas velocity is chocked, the gas mass flow rate is given by:

$$m_{c} = A C_{d} \sqrt{\gamma \rho_{0} P_{0} \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}}$$

Where A is the discharge hole sectional area (m²), Cd a dimensionless discharge coefficient and ρ_0 the gas volumetric mass (kg/m3) at total pressure P0 and total temperature T0.

For further details see: <u>https://en.wikipedia.org/wiki/Choked_flow</u>

In the case of a rotodynamic compressor (axial or centrifugal), the chocked flow limit may be designated by **overload**.

2.2_ Two phase flow - Sonic characteristics

Several numerical expressions are available to calculate the sound velocity for a two phase mixture. Nguyen has proposed a model where the gas phase is homogenously dispersed in the liquid phase.

$$c = \frac{1}{(1-\beta)\sqrt{\frac{1-\beta}{c_l^2} + \frac{\beta\rho_l}{\rho_g c_g^2}} + \beta\sqrt{\frac{\beta}{c_g^2} + \frac{(1-\beta)\rho_g}{\rho_l c_l^2}}}$$

Where β is the gas volume fraction, ρ the volumetric mass and c the sound velocity. Indices « g » and « I » refer to gas and liquid.

For further details see: <u>https://pastel.archives-ouvertes.fr/pastel-00005234/document</u> (« Développement d'une mesure de la célérité du son en écoulement diphasique - Application aux écoulements cavitants »).



Figure 2.2 – Sound velocity for a two phase carbon dioxide mixture (gas – liquid). The sound velocity is represented versus the gas volume fraction for several gas volumetric mass.

When the gas volume fraction tends, respectively, towards 1 and 0, the two phase sound velocity tends, respectively, towards the gas and the liquid sound velocity. In between, the sound velocity is reduced (smaller than the lowest – the gas - sound velocity).

At a low pressure (low gas volumetric mass), the two phase sound velocity may reduce to 20 m/s (that is 10 times smaller than the gas sound velocity). At medium pressure (of the order of 50 bar) the sound velocity is increased to 100 m/s. The two phase sound velocity approaches the gas sound velocity for a pressure greater than 150 bar.

2.3_ Hydrates

Carbon dioxide <u>hydrate</u> or **carbon dioxide clathrate** is a snow-like crystalline substance composed of water ice and carbon dioxide. It is normally a Type I gas <u>clathrate</u>. In the case of pure carbon dioxide and at low pressure, clathrate formation occurs below 283K (10 C). See <u>https://en.wikipedia.org/wiki/Carbon_dioxide_clathrate</u>.

The temperature threshold (temperature at which hydrate formation occurs) increases slightly with the pressure: 10°C at 50 bar abs and 12°C at 150 bar abs.

The temperature threshold increases with the fraction of associated components: 13°C at 50 bar abs and 16°C at 250 bar abs in the case of a mixture containing 5 per cent of methane.

2.4_ Flow surging

Rotating stall is a local disruption of airflow permitting the compressor to continue to provide a positive gas flow but with reduced effectiveness. The rotating stall arising at an air foil (or blade) may propagate to the air foils around it.

The propagation mechanism can be described by considering three blades 1, 2 and 3 among a blade cascade. A flow perturbation causes blade 2 to reach a stalled condition before the other blades. This stalled blade does not produce a sufficient pressure rise to maintain the flow around it. As a consequence, an effective flow blockage or a zone of reduced flow develops. This retarded flow diverts the flow around it so that the angle of attack increases on blade 3 while it decreases on blade 1. In this way, a stall "volume" may move along the cascade in the direction of the blade lift. The stall propagates downward relative to the blade row at a rate about half the rotational speed with the diverted flow stalling the blades below the retarded-flow zone and unstalling the blades above it. The retarded flow or stall "volume" moves from the pressure side to the suction side of the blades in the opposite direction to that of rotation. The stall zone may cover several blade passages. The radial extent of the stall zone may vary from just the tip to the whole blade length.

For further details see: <u>https://www.sciencedirect.com/topics/engineering/rotating-stall</u>.

Also Nuovo Pigone paper: https://core.ac.uk/download/pdf/188800045.pdf

Sometimes, a rotating stall is assimilated to an approach to a surge flow condition as a result of a significant increase in rotor vibrations.

Compressor surge is a form of aerodynamic instability in axial or radial (centrifugal) compressors. The term describes a violent air flow oscillating in the axial direction of a compressor. The axial component of fluid velocity varies periodically and may even become negative.

The surge flow may be described by considering the shape of the pressure coefficient curve of a single compression stage (hydraulic cell). By reducing the flow from the design point, the pressure coefficient increases until it passes through a maximum then reduces if the flow is further decreased. By decreasing the flow below the maximum of the pressure coefficient, the compression stage losses its capability to push the flow forward (toward compressor exit) resulting in a backward flow. The flow is then violently increased and progressively reduced (to reach the operating condition) until it generates a second backward flow. This generates a pumping system (oscillating phenomenon) with a frequency of the order of 1Hz. See figure 2.3.

This phenomenon can be very destructive particularly for compressors operating at high pressure, high rotating speed and handling gases with large a molecular weight. The compressor may be protected by using an anti-surge line recycling a part of the transmitted flow.

It has to be pointed out when the overall compressor curve may show at the surge line a possible increase in pressure increase following a flow reduction. This would be a wrong interpretation as the surge line is based on the occurrence of surge for a single stage while other stages are still remote from the surge condition. See figure 2.3.



Figure 2.3 – Representation of the surge line on an overall compressor performance map. Figure from <u>https://en.wikipedia.org/wiki/Surge_in_compressors#/media/File:Compressor_map.gif</u> from <u>https://en.wikipedia.org/wiki/Surge_in_compressors</u>

For general information on stall and surge see: <u>https://en.wikipedia.org/wiki/Compressor_stall</u> and <u>https://en.wikipedia.org/wiki/Surge_in_compressors</u>

For other unsteady flow fields see: https://hal.archives-ouvertes.fr/hal-01296905/document

2.5_ Centrifugal compressor operability

In a rotodynamic compressor (axial or centrifugal), the pressure ratio and the temperature ratio are given by:

$$\frac{Td}{Ts} = \frac{Pd^M}{Ts} = \frac{HMMw}{ZRT} + 1$$

Where T is an absolute temperature, P an absolute pressure, H a manometric head, M an isentropic exponent and Z a compressibility factor. Indices "s" and "d" refer to compression suction and discharge.

In the case of an isentropic compression, $M=\gamma^{\gamma-1}/\gamma$. An isentropic compression is defined by a compression operating at constant entropy: no losses - a reversible process.

In the case of a polytropic compression, $M=^{\gamma-1}/_{\gamma\eta}$ where η is the polytropic efficiency . A polytropic compression is characterised by an increase in entropy: occurrence of losses - an irreversible process.

The manometric head is calculated from: $H = \mu U^2 / 2$ where μ is a pressure coefficient and U is the impeller peripheral velocity. The pressure coefficient is dependent on the impeller geometry and the ratio U/C.

Other parameters have been defined in above sections.

The performance of a compressor is therefore dependent on the suction temperature, isentropic factor and highly dependent on the molecular weight. The dependence on the molecular weight is illustrated on figures 2.4 to 2.6.

The compression of hydrogen (molecular weight = 2) is presented on figure 2.4 in the case of a 10 stage compressor and for a rotating speed variation ranging 33 % above and below the design running speed (100 %). The compression ratio for each stage being relatively small, the manometric head and pressure ratio curves versus the relative volume flow are relatively flat providing an extremely large operating range in terms of rotating speed and volume flow.

The compression of methane (molecular weight = 16) is presented on figure 2.5 in the case of a 10 stage compressor and for a rotating speed variation ranging again 33 % above and below the design running speed (100 %). The compression ratio for the overall compressor is considerably larger with methane (8 at the design point) compared to hydrogen (1.25 at design point) the manometric head and pressure ratio curves versus the relative volume flow being sharper. The operating range in terms of rotating speed and volume flow is considerably reduced as it can be seen on figures 2.4 and 2.5.





The compression of carbon dioxide (molecular weight = 44) could not be presented similarly to the hydrogen and methane cases due to a too large pressure ratio. It is presented on figure 2.6 in the case of a 5 stage compressor (number of compression stages divided by 2) and with a rotating speed range of 22 % (2/3 of previous speed ranges). Despite the significant reduction in the number of stages, the pressure ratio is still greater with carbon dioxide compared to methane. The temperature ratio is also greater in the carbon dioxide case, the temperature limitation occurring at 5 % above the design speed (25 % in the methane case).

The flow range is also considerably smaller with carbon dioxide than with methane at maximum and minimum speed even after a significant reduction in the speed range (2/3 in carbon dioxide compared to the methane case). This significant speed range reduction can be explained in the following way. At high running speed, the compression ratio is so large

that the last stages of the compressor are in surge much earlier than the first stage. Therefore a considerable flow increase is required to remove the successive stages out from the surge condition. However, when all the stages are remote from the surge condition, the first stage is close to the overload condition. This is typically the case at the maximum running speed. This is roughly the opposite with the minimum operating speed: when surge is provided by the first impeller stage, following a relatively small flow increase, the overload condition is provided by the last compressor stage.



Figure 2.5 – Manometric head and pressure ratio for a 10 stage compressor with 33 % speed range around design running speed and temperature limitation. Gas: methane



Figure 2.6 – Manometric head and pressure ratio for a 5 stage compressor with 22% speed range around design running speed and temperature limitation. Gas: carbon dioxide

In conclusion, an increase in the gas molecular weight considerably increases the pressure ratio and the temperature ratio and significantly reduces the operating range.

2.6_ Materials

Some details concerning the material requirements may be found in the SPE 36 600 document. It is titled "Sleipner West CO2 disposal, CO2 injection into a shallow underground aquifer". Authors are from Statoil and Sintef.

Materials are selected on the following basis:

- Corrosion potential of wet carbon dioxide
- Low yield requirement in presence of hydrogen sulphide
- Material brittleness in case of sudden compression loop depressurisation: risk of temperature lowering below minus 50°C.

3_ Thermodynamic properties

Knowledge of thermodynamic properties is required to calculate the sonic velocity, the occurrence of a choke flow and the compression features of a fluid.

Thermodynamic calculation was performed by using the CoolPack software. It has been developed by the <u>Department of Mechanical Engineering (MEK)</u>, Section of <u>Thermal Energy</u> (<u>TES</u>) at the Technical University of Denmark (DTU): <u>https://www.ipu.dk/products/pack-calculation-pro/</u>

The specific heat cp is the heat (calorie) required to rise the temperature of a fluid of one mass unit of 1 °Kelvin at constant pressure (isobaric expansion – external work production). Its value is relatively constant and of the order of 1 kcal/kg/°K at a pressure smaller than the critical pressure. Its value slightly increases with the temperature. Figure 3.1 – Left.

The specific heat cv is the heat (calorie) required to rise the temperature of a fluid of one mass unit of 1 °Kelvin at constant volume (isochoric – Internal energy increase). Its value is smaller than cp and is relatively dependent on the temperature and the pressure. Figure 3.1 – Right.

From above definitions:

H = CpT where H is the enthalpy and U = CvT where U is the internal energy



Figure 3.1 – Specific heat cp and cv (kcal/kg/°K) versus temperature for several pressure values. Gas: carbon dioxide

For an ideal gas, the isentropic factor is: $\gamma = H/U$ or $\gamma = Cp/Cv$

In the case of a real gas, there are several methods for calculating the isentropic factor. Can be quoted the methods of Chandrasekar or Edmister.

Figures 3.2 show the difference between the actual isentropic factor (real gas – right figure) and a calculation based on the ratio of cp/cv (left figure). Strangely, the actual data vary very little in the gas phase (below 70 bar and above 30°C) compared to the ratio of cp/cv with a very strong dependence of the pressure.



Figure 3.2 – Specific heat cp / cv ratio and isentropic factor versus temperature for several pressure values. Gas: carbon dioxide

The compressibility factor (figure 3.3 - right) has been calculated from the actual volumetric mass (figure 3.3 - left). The compressibility factor is relatively close to 1 at low pressure (below 25 bar) then starts to decrease as the pressure is increased. The compressibility factor is around 0.2 at the critical point tending towards 0 in the liquid phase (temperature lower than 31 °C and pressure greater than 73 bar).



Figure 3.3 – Volumetric mass and compressibility factor versus temperature for several pressure values. Gas: carbon dioxide



The pressure - enthalpy diagram is used to perform compressor calculation together with the "Refrigerant Calculator" of the CoolPack Software. See figures 3.4 and 4.2.

Figure 3.4 – Pressure versus enthalpy diagram with entropy (S) specific volume (v) and isotherm lines (T). Diagram plotted by using CoolPack software.

4_ One compression and injection application case

A compression calculation has been performed on a carbon dioxide flow leaving an amine treatment unit. It is assumed that the amine treatment provides a gas at a pressure slightly above 1 bar abs and that it is required to bring it at a pressure of the order of 150 bar

The gas is suctioned by the first compressor section at a temperature of approximately 30°C. The gas outlet temperature is limited to 180 °C in the present study. Following each compression stage, the gas is cooled down to a temperature of 30 °C to provide sufficient margin with the temperature corresponding to hydrate formation, particularly, in the case of a gas mixture containing methane (See hydrate section).

Between each compression stage, a pressure loss of 10 % of the actual pressure is considered to take into account, the gas discharge from the compressor, the gas cooling, water separation in a dedicated vessel, the gas entrance into the compressor and the intermediate pipe sections.

In these conditions, the compression ratio for each stage is of the order of 4 and four sections are required to bring the gas at a pressure of the order of 100 bar. At this pressure level, following cooling, the gas enters into a dense phase condition (intermediate between gas and liquid condition) with a volumetric mass of the order of 750 kg/m3. It has to be noted that following the first three cooling's, the volumetric mass is considerably smaller, respectively, of the order of 6, 24 and 75 kg/m3 at the outlet of, respectively, the first, the second and the third cooling stage. See figures 3.3 and 4.1.

Following the fourth compression stage, the fluid is pumped (liquid) or compressed (gas) using a specific impeller to take into account the small compressibility of the fluid. See figure 4.2.



Figure 4.1 – Pressure (Bar abs) versus volumetric mass (kg/m3) with evaporation and condensation lines also two isotherms (10 and 31°C) for carbon dioxide.



Figure 4.2 – Pressure (Bar abs) versus enthalpy (kJ/kg) with evaporation and condensation lines, two isotherms and compression line for carbon dioxide.

Following the fifth compression stage, the fluid volumetric mass is of the order of 800 kg/m3 corresponding to a pressure of 140 bar and a temperature of 30°C. The fluid is therefore injected as an almost liquid phase. Would the fluid be cooled down to 10°C with the injection of hydrate inhibitors, the volumetric mass would approach 1 000 kg/m3 therefore increasing the manometric head of the injection well by 20 %. This permits an equivalent pressure reduction at the injection wellhead.

5_ Two phase flow compression

As shown on figure 4.2, carbon dioxide compression is started in gas phase at a pressure slightly above 1 bar abs and a temperature of the order of 30 °C. Should the temperature be significantly lowered, there is no possibility for liquid formation (at least not below minus 50 °C).

Above 40 bar abs, the temperature has to be controlled carefully to avoid the formation of carbon hydrates (temperature to be above 20 °C providing 5 °C temperature margin in case of presence of methane in the carbon dioxide mixture) and also to avoid the liquefaction of carbon dioxide when using a compressor handling only a gas phase.

Carbon dioxide could be partially liquefied without any risk of hydrate formation at 20 °C in view of using a two phase pump. This temperature corresponds to a liquefaction pressure of approximately 60 bar abs (figure 4.1). The compression would operate in two phase flow up to 73 bar abs corresponding to the critical pressure above which the fluid is in dense phase condition (single phase). The pressure range in two phase flow is therefore relatively narrow.



Figure 5.1 – Two phase envelopes for a propane – butane mixture with propane mass fraction varying from 0 to 100 %.

In addition, there is a major difficulty in controlling the volume flow crossing the two phase flow pump as the liquefaction occurs at a single point condition represented by a single pressure and a single temperature condition (case of pure carbon dioxide). This would induce a lack of knowledge (therefore control) in the gas and liquid fractions therefore in the total volume flow. This total volume flow could vary in the volumetric mass ratio of the liquid and gas phases that is of the order of 6 (See figure 4.1).

If the fluid mixture was made of a least two pure components, the gas-liquid phase fractions would be represented by an envelope and not by a single line (pure component) as it is represented on figure 5.1 in the case of a butane – propane mixture. When the propane fraction is close to 0 or 100 %, the liquefaction line is represented by a single line. However, when the gas fraction tends towards 50 %, the liquid – gas phase envelope becomes wider allowing a better control in the liquefaction of a fraction of the mixture. For instance, at 3 bar abs, the temperature range is of the order of 7 $^{\circ}$ C.

In conclusion, the two phase compression of carbon dioxide does not present a major advantage, the domain of operation being extremely reduced and the liquefaction process being practically out of control.

6_ Conclusion

Sonic losses may be limited by controlling the gas velocity below the speed of sound at the blade leading and trailing edges. This is particularly important at low temperature.

Choke losses may be limited by controlling the gas velocity in the hydraulic cell channels particularly at low pressure and low temperature.

The operation of a multi section compression system is limited in compression range due to a limited rotating speed range. It is also limited in flow range due to, respectively, the surge and overload occurrence when the flow is slightly, respectively, reduced or increased.

The compression of carbon dioxide in two phase flow does not present a major advantage as the domain of operation in pressure is extremely narrow and the liquefaction process is practically out of control