CO_2 compression challenges

Klaus Brun, Brian Pettinato, Stephen Ross and Todd Omatick, Elliott Group, and Joseph Thorp, Aramco Ventures, introduce a promising technology for large-scale carbon storage compression applications.

The objective for carbon capture and sequestration is not to isolate turbomachinery component design from the thermodynamic cycle, but to design the turbomachinery as an integrated system so as to increase the efficiency of the entire process rather than just the machine. This requires customised solutions that are often not commonly available with off-the-shelf or turbomachinery products.

Design challenges include:

- System performance (efficiency) and operability (variability).
- Cyclic operation.
- High pressures and temperatures.
- Hostile environment.

From a purely thermodynamic perspective, carbon dioxide (CO_2) is relatively easy to compress. This means that the pressure ratio per compressor impeller stage is high, and compressors with low stage counts or speeds are required. Because of the high pressure ratio per stage, however, CO_2 also has a rapid specific volume decrease with pressure and a very high heat of compression. Consequently, CO_2 will heat up when compressed, which requires stage intercooling to maintain the gas temperature at reasonable levels so as to not damage the seals and bearings of the compressor. Furthermore, because of its rapid density change with pressure, there is a significant flow volume reduction that requires a wide range of aerodynamic high-to-low flow compression stages.

Since both reciprocating and screw compressors are severely flow limited, they cannot be practically used for large-scale industrial carbon sequestration applications. Viable technology options rely on proven centrifugal compressor or pump impellers but use different layouts and stage arrangements. Two of the most promising options for typical 30 - 2100 psia carbon storage compression applications are an intercooled centrifugal barrel compressor feeding a dense phase pump and integrally geared compressor with inter-stage cooling.

Compressor design

The target for a centrifugal compressor design is to compress the CO_2 gas from near atmospheric suction pressure to above the critical point (1070 psia) and into the supercritical gas regime to about 1100 - 1150 psia. Above that pressure, the gas is in dense phase (either as a supercritical fluid or a liquid), and it is advantageous to utilise a pump for any further pressure increase to the required discharge pressure. To reduce cost, it is desirable to maintain the compression process from suction pressure to above the critical point in a single compressor casing.

A case study was devised for a CO_2 compressor design for a typical carbon separation and sequestration application at a 500 MW natural gas combined cycle plant. Depending on its efficiency and loaded operational hours, this plant will produce 1 - 2 million tpy of CO_2.

If a pre-combustion carbon separation process is used, the CO_2 would be available at approximately 30 - 60 psia, while for a flue gas clean-up, the pressures would be 15 - 35 psia. On the discharge side, it is assumed that the CO_2 will be injected into a pipeline operating at 2100 psia. Therefore, the following design conditions were selected for a hybrid CO_2 compressor and pump:

- Ps = 30 psia.
- Pd = 2215 psia.
- Ts = 100˚F (37˚C).
- Q = 1 million tpy (70 lb/s or 52 million ft^3/d).
- Gas = 100% of CO_2.

The compressor is designed for a 30 - 1150 psia pressure rise, and the downstream supercritical pump for a 1100 - 2115 psia pressure rise. A conservative 50 psia pressure drop is assumed to account for discharge cooler, pipe, and valve losses between the compressor and the pump.

Figure 1 shows a six impeller stage barrel-type centrifugal compressor layout and cross-section with multiple nozzles for intercooling or side-streams. Figure 2 shows a multi-stage horizontally split compressor with nozzles for two intercoolers driven by an electric motor through a gear-box. This is a typical design only, but it is representative for the application. Typical CO_2 carbon separation operating conditions for these machines are suction pressures from <50 psia to discharge pressures of >1100 psia. At 1100 psia, the compressor discharge gas will be in the supercritical state, and after cooling, it can be fed directly into the dense phase pump for higher pressure pipeline transport or storage injection.

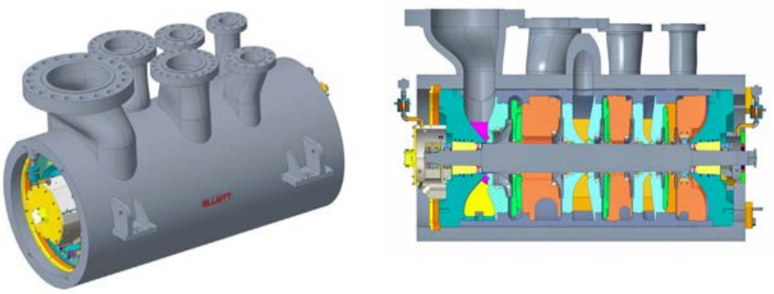

Figure 1. Multi-stage barrel centrifugal compressor design for CO_2 (Source: Elliott Group).

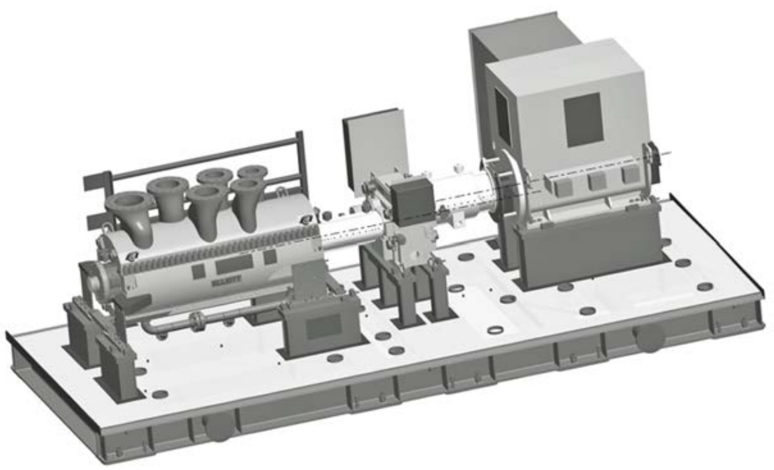

Figure 2. Centrifugal compressor with multiple intercooling nozzles (Source: Elliott Group).

Supercritical pump design

Pumps for supercritical CO_2 pumping for pipeline boost have been demonstrated in transport and enhanced oil recovery applications and are derived from standard products. For the required operating conditions of 1100 - 2115 psia, a 10-stage pump running at 3600 revolutions per minute (rpm), using 15 in. impellers, and requiring 1850 hp, could satisfactorily provide the service. Figure 5 shows a schematic of this supercritical CO_2 pump. The efficiency of this pump is around 70%. Thus, depending on the application, total compression system efficiency is also 65 - 75%. In a non-adiabatic system, however, efficiency is of limited significance.

Performance

Polytropic and isentropic efficiencies are of limited value for processes that have intercooling or other heat losses or addition. In these cases, it is more practical to look at total power consumption for a quantitative evaluation of the effectiveness of the process. The total power demand of a CO_2 compression process is strongly path dependent and can vary widely based on the number of stages of cooling and cooler performance. It is also a function of the compressor and pump performance. For this case study, a total power consumption of approximately 15 000 hp was estimated with the compressor requiring about 13 000 hp and the pump about 2000 hp. The compressor speed is approximately 8500 rpm, and the pump speed is 3600 rpm. These performance numbers are a strong function of ambient cooling conditions, valve/pipe losses, gas purity, cooler design, and other system variables.

CO2 compression rotordynamic challenges

The main causes of lateral vibration are unbalance, misalignment, and instability. The primary methods for ensuring acceptable vibration are eliminating sources of excitation such as unbalance and misalignment, and desensitising the rotor to sources of excitation through proper design and rotordynamic analysis.

Rotordynamic requirements are set by API standards and include achieving separation margin from critical speeds and achieving required logarithmic decrement stability values. Separation margins from lateral natural frequencies are largely determined by rotor sizing, bearing selection and coupling selection. Logarithmic decrement stability is similarly dependent on rotor sizing/configuration and bearing selection and support system for what is known as a basic log decrement (δb). Destabilising excitation sources, such as aerodynamic cross coupling and gas seals, act to further reduce the rotor stability. Stabilising mechanisms such as shunts, swirl breaks, damper seals, and/or squeeze film dampers (SFDs) act to improve stability. Analysing the full model, including all mechanisms influencing the rotor, will result in a final log decrement (δf).

Specifically, in the case of beam-style CO_2 compressors, one or more intercoolers are often applied to improve performance. Applying one or more intercoolers to a beam-type compressor requires multiple sections, which can be easily handled using multiple compressor bodies, but at a considerable cost. A far less expensive solution is to place multiple sections in a single beam machine, but this typically requires a long bearing span resulting in high rotor flexibility, which can be a difficult problem with respect to rotordynamics, both in terms of the critical speed separation margin as well as the stability. In addition, the gas density for compressed CO_2 is also high, which tends to create a higher risk with respect to rotordynamic stability.

As an example, Figure 3 shows the rotordynamic model of a beam compressor suitable for compressing CO_2 up to a cavern storage pressure of roughly 1900 psia (130 bara). The compressor has two sections in a back-to-back arrangement with three stages in each section. A centre seal is used to seal the two sections from each other. The centre seal tends to have considerable influence on the rotordynamics of the machine as well as the overall performance.

Centrifugal compressor rotordynamics are heavily dependent on the aerodynamic path. Most OEMs offer a series of aerodynamic stage solutions to solve problems such that the required compression can be performed in a small number of compressor bodies. In this case, the smaller bore coincides with the first section of the machine having the highest flow coefficient wheels, whereas the larger bore coincides with the second section having the lowest flow coefficient wheels. These wheels are somewhat optimised with respect to stage space to reduce the overall rotor length.

Plotting the critical speed ratio (operating speed/critical speed on stiff supports) vs the average gas density of this compressor indicates that the compressor lay within API

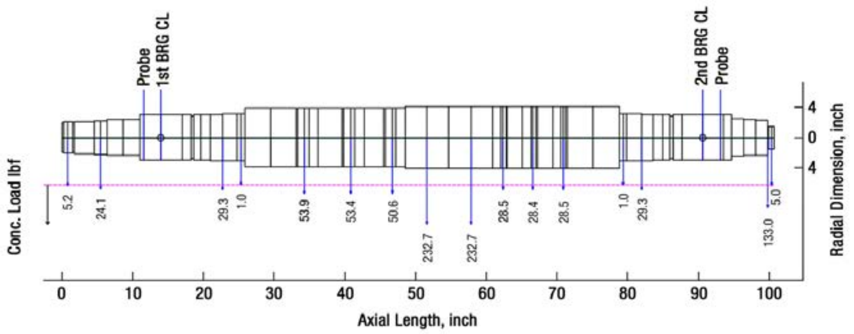

Figure 3. **Small bore CO_2 compressor rotor with centre seal.**

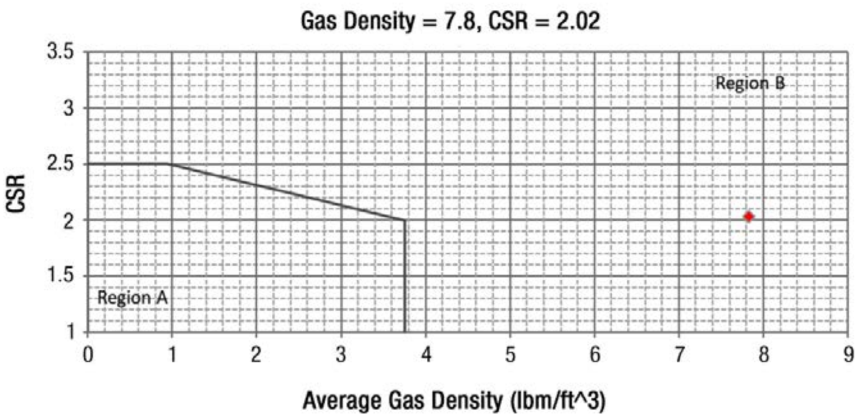

Figure 4. **API stability experience plot.**

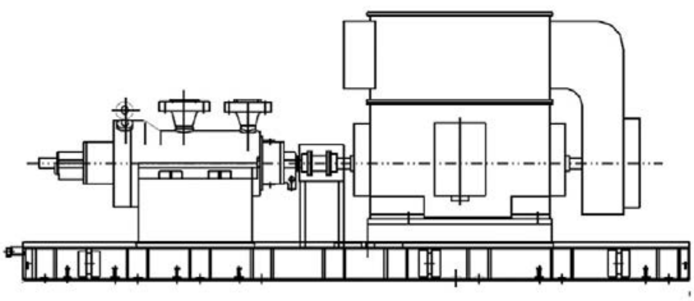

Figure 5. **2000 hp supercritical pump CO_2 with 10 stages (Source: Ebara Corporation).**

Region B of the stability experience plot (Figure 4), further indicating a rotordynamic stability challenge.

Further rotordynamic details of this selection are summarised in Table 1. As seen in the table, the final stability of this rotor is heavily dependent on the centre seal type and main shaft diameter. The standard labyrinth seal, in combination with a small bore rotor, is predicted to be unstable. The use of a hole pattern centre seal improves the rotordynamic stability considerably, achieving a final logarithmic decrement (δf) that is greater than 0.1. A large bore rotor can also provide improved stability that also achieves an acceptable logarithmic decrement. Additional ways to reach an acceptable solution including SFD bearings, applying swirl brakes at each stage, or some combination of methods. The final selection will be a design that provides the best combination of overall performance and reduced cost.

Table 1. Rotor description and rotordynamic stability

	Small bore rotor	Small bore rotor with hole pattern seal	Large bore rotor
Centre seal type	Labyrinth	Hole pattern seal	Labyrinth
Bearing span, L (in.)	76.6	76.6	74.1
Main diameter, D (in.)	7.7	7.7	8.8
Shaft L/D (dim)	9.9	9.9	8.5
Nma (rpm)	6481	6487	6602
Nmc (rpm)	8587	8587	8747
FCSR (rpm)	4243	4243	5910
CSR (dim)	2.02	2.02	1.48
ρave (lbm/ft3)	7.8	7.8	7.2
δb	0.46	0.306	0.708
δf	-0.521	1.101	0.798

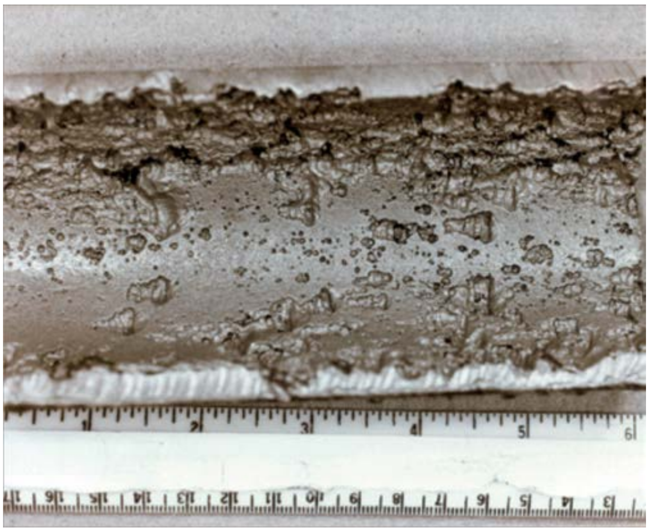

Figure 6. Blistering in elastomeric materials due to rapid decompression of CO_2.

CO_2 materials issues

CO_2 gas can present a corrosion risk to metallic components. When moisture is present, the CO_2 reacts with water to form carbonic acid which can result in a general corrosion attack occurring at the surface of a carbon on low-alloy steel grade. 'Sweet' corrosion occurs when carbonic acid is formed when CO_2 is contacted by water (weak acid).

This general corrosion attack creates a corrosion product which covers the entire exposed surface. Stress corrosion cracking and surface pitting are not concerns with CO_2 gas unless chloride or sulphide impurities are included. While carbon steels are susceptible to a general corrosion attack, stainless steels are resistant to corrosion from CO_2 and carbonic acid. This corrosion resistance applies to all forms of stainless steels including martensitic (400 series), austenitic (300 series) and duplex (200 series) grades of stainless steel.

In summary, basic material concerns with CO_2 compression thus are:

Base material

- Carbon steel (C-Mn) – suitable if stream is dry.
- Stainless steel – required if H_2O is unavoidable.
- Application may be subjected to rapid temperature swings due to isenthalpic effect.

Elastomers are susceptible to explosive decompression

- May result in blistering or rupture (Figure 6).

Conclusion

One promising technology for large-scale carbon storage compression applications is a hybrid combination of a centrifugal compressor to compress the gas to slightly above its critical point in series with a dense phase pump to reach the desired process discharge pressure. Both low-pressure CO_2 compressors and dense phase pumps are proven technologies, but their hybrid combination has not seen significant service in the industry.

A case study with representative operating conditions for a typical power plant carbon separation and sequestration application using a hybrid compressor pump package for CO_2 compression was presented. Rotordynamic and material constraints, as well as machine, package design, layout options, footprint, and performance parameters, were provided.

Many other viable compression paths and technologies exist. Some of these are commercially available, while others are still in development. Regardless of the type of compressor, it is important to be aware of the compression issues and to recognise that CO_2 compression has technical challenges that are often application specific and must be individually addressed for all new carbon sequestration technologies.

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