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Compressors in Energy-Related Applications

While their use in the movement of natural gas through pipelines is perhaps the most common purpose, compressors are used frequently in upstream and downstream applications.

Tertiary Recovery Project Requires Compression Selection

The project

Engineering design of reciprocating and rotary screw compressor capability and sizing was demonstrated on a unique tertiary recovery and life-of-field extension project in the southeast New Mexico area of the Permian Basin. Located near Hobbs at an elevation of 3,615ft (1,102.58m) above sea level, the site included facilities for a carbon dioxide (CO_2) and water flood in a 13 sq-mile (33.67 sq-km) field with a design life of 40 years. The flooding of reserviors with CO_2 involves injecting the gas into the wellbore of existing and new wells. In the reservoir, CO_2 acts as a solvent, allowing oil to be removed from porous rock and flow more freely to nearby producing wells. This procedure is designed to increase the oil recovery rate. The water injection will alternate with the CO_2 gas injection (or WAG).

The re-injection project entailed the construction of distribution, injection, collection and treating facilities, along with gas, water and oil gathering lines, oil treating and a 100,000 bbl water treating and injection facilities expansion. Seven production test satellites, a new CO_2 re-injection compressor facility and a water injection station, along with about 180 miles (289.62 km) of the necessary flowlines, injection lines and production transfer lines will be installed. To optimize the economics of the project, Mustang Engineering utilized its proprietary SYDES hydraulic flow modeling software. The tool assists in predicting the steady state flow conditions of gasses and liquids, including pressures and temperatures. It takes into account the equipment (pumps, compressors, heaters, etc.), the physical properties of the gas or liquid and heat exchange with the environment. Among other things, the modeling results aid in the proper sizing of gas and transmission lines, compressor power consumption and facilities design.

As part of the project, a 75 MMscf/d produced gas/CO_2 dehydration package will be installed along with two 25 MMscf/d re-injection compressors. The compression facility for re-injecting the gas is expandable to 75 MMscf/d by adding a third compressor in the future.

Figure 1. Clean air panel

Operating criteria and compressor selection

Low-Pressure Compressors—The wet CO_2 gas gathered from the production test satellites will be compressed and brought into the re-injection compressor facility from three remote tank batteries through a low-pressure fiberglass pipe gathering system. For that purpose, a Cooper Superior (manufactured by Howden in the U.K.) oil flooded rotary screw compressor was chosen for each of the battery compression requirements. This compressor type has a history of operation in severe wet land-fill gas service, making it highly suitable for this application. The three units were designed to operate at 20 psi suction with a 300 psi discharge, loaded to 534 hp at a rate of 3.0 MMscf/d.

With a worst case scenario calling for a 40-psi suction and simultaneous failure of the hydraulic actuated capacity slide valve, the motor demand would increase to 728 hp. Therefore, a 750 hp Toshiba motor was specified. Because of the high H_2SO_4 level coming from the batteries, all associated vessels, piping and air coolers on this low pressure compressor package were specified for construction in either 304L or 316L stainless steel. The 304L suction scrubber, a vertical separator, was specified with an internal 316 stainless steel vane pack. The discharge vessel was a vertical filter/coalescer that assists in separating the oil injected during compression from the process gas. This vessel also is being supplied in 304L stainless steel. The compressors will operate at a discharge pressure of 300 psi, allowing for a 25-psi flowline pressure drop before entering the suction side of the main re-injection compressors from the dehydrators. Once separated and dehydrated, the water (H_2O) content was largely removed (about 2 lb H_2O/MMscf/d), leaving the gas at slightly more than 90% CO_2 with less than 5,000 ppm hydrogen sulfide. With the gas dried to that extent, no special metallurgy was required for the line coming from the dehydrators to the scrubbers on the suction side of the re-injection compressors.

Re-injection Compressors—For the re-injection of CO_2, the client initially suggested the use of a high-speed compressor driven by an electric motor because of the favorable cost of electricity. Original consideration was given to a 900-rpm, four-throw reciprocating compressor. Mustang ran preliminary sizing calculations based on the projected throughput and used manufacturers sizing programs to determine the number of compressors required and the best cylinder configuration. The ultimate economic and power consumption analyses favored a slower speed reciprocating compressor option.

As a general rule of thumb, the slower the speed, the less horsepower consumed and the lower the power costs.

The recommendation was for three 514 rpm six-throw double-acting compressors, each capable of handling about 25 MMscf/d gas. The compressors chosen were synchronous motor-driven Cooper Superior model WG76, with a 7-in. stroke. Compressed CO_2 gas entered the compressors at about 270 psi and was discharged at almost 1,900 psi. The calculated power requirements indicated 2,628 hp. Based on the 25 MMscf/d discharge flow and using a standard safety factor of 10%, the required motor horsepower was 2,891 hp, hence the selection of a 2,900-hp motor. The lower speed design required two additional compressor cylinders

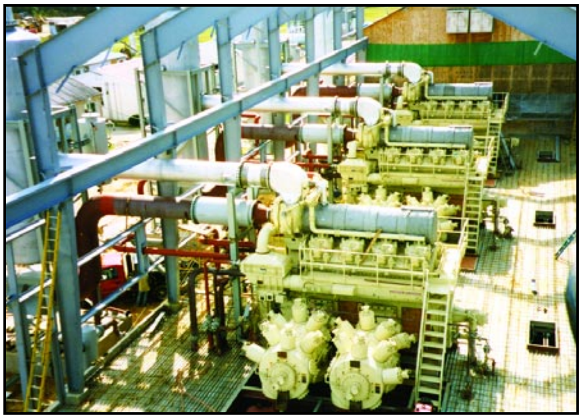

Figure 2. Reciprocating engines

Figure 3. Gas-fired engine driving centrifugal compressor (background)

Figure 4. Centrifugal compressor

to achieve the required displacement. The three first stage cylinders were 12in. and the second stage cylinders were 7 1/4in. Motor-driven horizontal air-cooled heat exchangers were utilized to cool the compressed gas between stages and after cooling.

The decision to go from a 900-rpm, four-throw frame to a 514-rpm, six-throw model had an approximate 20° discharge temperature reduction and a 331-hp savings per compressor. Even though the slower speed units were a more expensive purchase, the combined 662-hp savings translated into substantially reduced electrical costs and a 3.1-year payout on the additional capital cost of the two units ordered. The slower speed units are expected to have less maintenance and more uptime. Suction and discharge valve life should be approximately doubled. Based on the manufacturer's estimated valve costs of almost U.S. $1,000 each, and with 72 valves needed for the two purchased units, the reduction in valve costs on a bi-annual basis will be a minimum of $70,000, not including labor and downtime costs.

The compressors were equipped with variable volume clearance pockets for added load control. These adjustable pockets allow for clearance adjustments to be made to the head-end of each compressor cylinder. This modification allows for capacity reduction as needed and minimizes the amount of higher pressure discharge gas recycling. For this project, a discharge gas meter is monitored from the compressor control panel's programmable logic controller (PLC). This PLC provides an output to the head-end pneumatic suction valve unloaders, which cause cylinder head-ends to progressively unload as recycle gas volumes increase. The system tries to match the gas compressor capacity to system input rates.

Studies conducted

The compressor packager contracted Southwest Research Institute to perform pulsation studies on the reciprocating compressor piping and pulsation bottles. The studies were undertaken in accordance with the American Petroleum Institute 614 "design approach 3." The analog studies conducted on the suction and discharge pulsation bottles and their supports assisted in the correct sizing and configuration of orifice baffles and choke tubes for regulating flow in the bottles.

The client ran separate power studies. The economics of a unity power factor favored the decision to recommend the synchronous motor.

The compressor manufacturer completed a torsional study. The study, normally undertaken with large compressor packages, determined the flywheel mass and coupling design needed for the compressor's drive torsional response. The study results confirmed a flywheel was needed and provided sizing input to the packager for coupling selection.

Soil studies were conducted to determine its stability for foundation support. While this southwest region is largely comprised of limestone and is less susceptible to soil shifting, a civil engineer should be part of the team for designing the foundation and pipe supports. For this particular project, the decision to use the slower speed, six-throw compressors resulted in an increase in the weight of the final skid package, placing a substantially increased loading on the equipment pad.

Compressor accessory recommendations

Wherever possible, it is recommended the frames/cylinder configuration be designed so rod loads on both stages are balanced. A good conservative compressor design also suggests the actual rod load be about 80% of the manufacturers' maximum rod load ratings. For this project, using the manufacturer's rated compression rod loads of 70,000 lb, the first and second stage loads were almost identical at slightly more than 56,000 lb.

Similarly, it is recommended the compressors be sized so the discharge temperatures for all stages of compression be below 275°F (134.87°C) to 280°F (137.64°C). In this instance, the designed operating discharge temperature for stage one was about 245°F (118.22°C), while the discharge temperature on the second stage was about 268°F (130.98°C). While 350°F (176.49°C) is the stated maximum cylinder discharge temperature of many compressor manufacturers, it is never a good design to purposely exceed 280°F.

It has been a long-standing rule of thumb that one can compress each stage of compression by three compression ratios (based on absolute pressure). Usually, most designs proposed by compressor packagers will only have compression ratios between 2.5 and 2.8. Going three full ratios usually results in excessive temperatures or rod loads.

Because of its geographical location, the field experiences minimum winter temperatures nearing 0°F (-17.76°C). Therefore, crankcase immersion heaters will be installed on each unit. These auxiliary heaters will keep the crankcase oil warm, lower oil viscosity and facilitate compressor startup in cold weather.

In addition, an electric pre-lube pump was recommended for each of the compressors. These pumps will pressure up the compressor oil header, with oil being distributed evenly to every bearing surface, prior to rolling the crankshaft. This optional auxiliary equipment is designed to prolong bearing life.

For these units, second-stage discharge pressure cylinders were equipped with a packing water cooling and pump assembly to help cool the compressor cylinder packing rods. These units have independent water-cooling pumps and heat exchangers and assist in reducing the need for maintenance on the piston rods in high-pressure service.

As with these compressors, more operators are requiring dual oil filters so change out of the filter does not necessitate a unit shutdown. Instead, an isolation valve blocks and isolates the filter needing to be changed.

It is suggested the gas coolers be designed to have a maximum of 20°F (-6.66°C) approach above ambient. With this design, discharge gas needs to be cooled between stages to a temperature of no more than 20° above the stated maximum ambient design temperature (110°F – 43.29°C – in this case). While this design is more costly than a 30°F (1.11°C) solution, it reduces the suction temperature for the second stage, conserving horsepower and reducing power demand.

Other recommendations

Mustang identified those external lines where temperatures could be elevated. For those lines, a thermal stress piping engineering study will be conducted. Long piping runs of about 100ft (30.5m) between the compressors and the gas coolers represent one such area. The studies will indicate whether expansion joints or bellows need to be placed on the piping runs to provide for thermal growth and reduce stress on the equipment nozzles in those locations.

Steel skids supporting the compressor and engine units are subject to potential instability and severe vibration. Customarily, grout is poured around the skid's perimeter and some unit packagers will pour concrete or grout under the compressor. The client for this project required the skids have additional openings in the deck plate and at the multiple joint spacers traversing the deck. These added spaces allowed the application of extra epoxy grout to form better adherence between the steel frame of the skid and the concrete pad on which it rests.

Summary

As with most compressor projects, there are tradeoffs in the design specifications of equipment to provide the most optimal solution at the lowest possible cost. Sometimes the initial capital equipment cost is higher but can easily be justified by the annual operating cost savings that provide a rapid payout. As a general rule of thumb, the slower the speed, the less horsepower consumed and the lower the power costs.

Operating temperature is of great importance in designing compression. Ideally, if it can be reduced and balanced between stages, it will produce horsepower savings and lead to more efficient operation. It also is important when designing compression to take into consideration the variation in seasonal ambient temperatures and the corrosive nature of the input gases. These factors can have a direct impact on the equipment capacity as well as on associated piping. Auxiliary equipment can be added, as necessary, to improve operation.

Consideration also should be given to the soil conditions and foundation of the equipment pads and pipe supports. Proper engineering of these facets of the project can reduce vibration and improve general operating conditions of the compression facility.