Centrifugal pumps for upstream oil and gas applications
Abstract

Upstream oil and gas applications present technical challenges to OEMs in terms of materials, pressure, flow rate, installation, integration and general constraints. This article describes, from a deep technical perspective, the main experiences of GE Oil & Gas with upstream applications such as CO₂ re-injection, BB3 pump-produced water re-injection, and BB5 pumping for treated sea water re-injection.

Introduction

For 50 years, GE Oil & Gas has manufactured a wide range of API 610/ISO13709-compliant centrifugal pumps for hydrocarbon processing, refineries, water injection and pipeline services, with almost 20,000 pump units installed worldwide. In the last 20 years, by leveraging our vast experience with R&P pumps and the research and design practices applied to high-pressure sour gas centrifugal compressors, GE Oil & Gas has executed centrifugal pump projects in upstream applications whose complexity was particularly related to the adoption of special materials like superduplex/nickel alloys, discharge pressure beyond 500 bar, and a composite balance of plant integration.

CO₂ Pump Re-Injection for Tupi Pilot

The BB5-type multistage barrel pump was selected for very high pressure CO₂ re-injection by floating, production, storage and offloading (FPSO) systems in the Tupi pilot field. The opposed impeller back-to-back rotor configuration provides good overall efficiency compared to the inline rotor configuration. This is because the central balancing bushing has a smaller diameter when compared to a balancing drum, thus reducing the internal leakages that become a significant source of power losses, especially when dealing with low-viscosity fluids. The configuration also provides improved rotodynamic damping, even in worn clearance conditions, for pumps with such high deltaP and flexible rotors.

Impeller design

The impeller was designed to accommodate the variation of the volumetric flow rate through the pump caused by the dense phase compressibility so that the stream could be elaborated with only one identical impeller. (CFD results are shown in Figure 2.) The design of the new impeller family has a more streamlined pattern flow with subsequently higher stall tolerance.

Rotor dynamics

The low density of warm supercritical CO₂ implies low damping and stiffening effects of internal wear ring leakages. Therefore, rotodynamic assessment is a critical aspect of the design process. It’s especially important to assess the residual imbalance derived from the balancing capability of a slender shaft that is experiencing the actual damping and stiffening effect of the low density, low-viscosity supercritical CO₂.

Performance predictions

The selection of the pump – like that of a centrifugal compressor – must always account for the high compressibility of supercritical CO₂. Therefore, the polytropic head is the correct parameter expressing the work per unit of mass provided by the pump. Moreover, the polytropic efficiency also is preferred for compressors.

Validating the equation of state, as it’s tailored to stream components and related percentage, becomes vital to providing the most accurate performance prediction. While this is normal for compressors, machinery manufacturers have never had to handle a supercritical mixture in dense phase that contains a
A high amount of contaminants. The Tupi pilot re-injection stream had a very high percentage of contaminants at pressures and temperatures never experienced by compressor applications. Therefore, a research institute was commissioned to perform density and heat capacity tests that reproduced the same main conditions of the stream in terms of contaminants percentage, pressure and temperature. The identified discrepancies between the mathematical prediction based on mixed rules and the test results enabled the fine tuning of the polytropic head and performances. Figure 3 illustrates a rough operative envelope of the CO₂ pump. The pump adoption also depends on the density variation predicted.

**Pressure boundary and materials**

The structural design and material selection criteria were derived from the proven experience of CO₂ compressors. For example, the mechanical seals assembly with pump case was made with special shear rings that GE designed to withstand almost 1,000 kN of axial thrust produced by a barrier fluid pressure equal to 550 bara. Given the negligible viscosity of supercritical CO₂, the wear rings were built with self-lubricating composite materials. All outer barrel flanged connections were designed to meet an API 6A standard 10000 rating.

**Mechanical seals**

The selected seals, which were specifically engineered, were triple seals coupled to the pump case with shear rings, in accordance with the 53B API 682 plan, and a barrier fluid pressurized at 550 bara. The barrier pressure is segmented with an additional seal toward the atmospheric side so that the barrier pressure could be halved in an additional level. In this way, the seal faces design was safer and more robust, with a deltaP on each seal of approximately 275 bar. The mechanical seals were qualified at the supplier’s workshop and later tested with the pump on GE’s CO₂ test bench.

**CO₂ Pumping Testing**

The CO₂ test campaign conducted at GE’s Bari site had three main phases:

- A low speed (3,580 rpm) campaign with the same Tupi pilot mechanical seals and seal system, using a mixture containing 83 percent CO₂ and 17 percent N₂. Suction conditions were the same on all four installed pumps.
- A high speed (up to 7,600 rpm) campaign with dry gas seals, pure CO₂, and a suction pressure and temperature of 10 MPa and 15°C, respectively.
- A high speed (up to 7,600 rpm) campaign with dry gas seals, pure CO₂, and a suction pressure and temperature of 10 MPa and 40°C, respectively.

GE was able to validate the integration of the pump with the mechanical seals and seal system during the first phase of the testing campaign. The mechanical running was completed successfully with no signals out of the allowable range. Temperature/pressure rise and pump efficiency were validated in the four suction conditions, which were reproduced through subsequent venting, filling, heating and cooling. From the inlet pressure of 30 MPa, to the final outlet pressure of 54 MPa, the gas temperature rose from 40°C to 76°C.

The test was instrumental in adjusting the impeller trimming laws, based on the polytropic head required. The same pump, retrofitted with dry gas seals, also was tested with pure CO₂ at the full design speed of 7,600 rpm. With this configuration, the differential pressure reached 45 MPa (6,525 psi) and the flow rate was 35 kg/s. The stability of the rotor would have allowed the speed to be increased if there had been no limitations on the driver.
Figure 5 shows graphs of the measured discharge pressure and total efficiency. The pump was operated at 7,600 rpm with 10 MPa (1,450 psi) suction pressure, whereas the suction temperature was varied between 15°C and 40°C – the typical seasonal range of water-cooled CO₂. It’s clear that the same suction pressure yields different discharge pressures at different average inlet densities. The performance chart shows that the values and the shape of the Q-H and efficiency curves vary. Increasing the suction temperature progressively decreased the efficiency.

As expected, the two high-speed test phases confirmed that it was possible to regulate the discharge pressure acting on the water cooling flow rate of the cooler installed upstream from the pump. Even though speed regulation was confirmed as the most efficient and rapid way to regulate, suction temperature regulation is a competitive way to regulate discharge pressure if there are no requirements for system time response.

One of the most important results of the test is the data on the behavior of dry gas seals with liquid CO₂. The assessment referred to the need to heat the CO₂ bled by the pump discharge and used as buffer fluid so that the primary seal of the tandem arrangement wouldn’t freeze. This assessment is shown in Figure 6, which reports the thermodynamic scheme of a tandem dry gas seal.

**BB3 CO₂ pump**

GE’s BB3 pump (Figure 7) design has been upgraded to provide a competitive product fit for low-pressure CO₂ re-injection (<200 bara) applications. The main upgrade, involving the mechanical seal chamber, was done to avoid large protruding mechanical seals when there are double or engineered seals, and to guarantee a safe sealing of the gaskets at the outer diameter, which was accomplished by assembling a one-piece sleeve in the seal chamber together with a robust axial gasket at the end of the chamber. The arrangement, which was tested and validated, is shown in Figures 7 and 8.

**BB3 pump for produced water re-injection**

If the total injection flow rate is split into more than two or three injection lines, the required impeller diameter allows the selection of a BB3 pump instead of a BB5 pump – a choice that improves competitiveness and shortens maintenance lead time. Even though API 610 suggests that BB3 pumps only be used in applications with design pressure lower than 100 bara, GE’s BB3 pump can operate with a 10-inch impeller diameter to withstand pressures that are typical of well re-injection applications (>200 bara). In remote areas with natural gas-rich oil fields but poor electrification, high-pressure re-injection pumps can be driven by reciprocating gas engines. In the North Rumaila field in Iraq, GE provided a full package of 10 trains comprised of a gas engine, air starting system, jet-pulse air intake filters, station fuel treatment systems, air coolers, high-pressure BB3 pumps and 10 produced water booster pumps.
The main challenge for such a train is ensuring robust and reliable torsional behavior under all operating conditions. Building and simulating such a complex rotor scheme requires significant effort and joint responsibility between pump and engine OEMs, all of whom were within GE for this project (see Figure 10).

Total fluctuating torque, generated by gas pressure and the inertia of engine cylinders, was modeled in stationary and transient conditions. Particular care was taken to address the possibility of misfire combustion in the 16-cylinder engine, a condition in which major harmonic orders could undermine rotor and visco-elastic coupling stability (see Figure 11).

The complete train was string tested successfully during four hours of mechanical running. GE’s comprehensive expertise and products simplified the significant integration effort on the package. That said, the produced water re-injection pump itself (Figure 12) certainly presents the greatest technical challenges. The main pump features are summarized in the following table:

<table>
<thead>
<tr>
<th>Feature</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pump model</td>
<td>3x9 MSN10stgs</td>
</tr>
<tr>
<td>Speed</td>
<td>5725 rpm</td>
</tr>
<tr>
<td>Rated power</td>
<td>2.2 MW</td>
</tr>
<tr>
<td>Design pressure</td>
<td>350 bar</td>
</tr>
<tr>
<td>Max flow rate</td>
<td>320m3/h</td>
</tr>
<tr>
<td>Hydro-test pressure</td>
<td>530 bar</td>
</tr>
<tr>
<td>Efficiency</td>
<td>74.2 percent</td>
</tr>
<tr>
<td>Process fluid</td>
<td>Water with H2S and very high percentage of chlorides</td>
</tr>
<tr>
<td>Materials</td>
<td>Super duplex/nickel alloys</td>
</tr>
<tr>
<td>Full NACE compliance</td>
<td></td>
</tr>
</tbody>
</table>
Because of the moderate H2S content of the produced water, the required pump material class was D2 in accordance with API 610 and compliant with NACE. In addition, because of the hardness limit, the NACE compliance excluded ASTM A276 S32760 in strain-hardened conditions for bolts and nuts (considered as wetted parts). On the other hand, ASTM A276 S32760 in annealed conditions doesn’t have mechanical properties compatible with design pressure of 350 bara (hydro test pressure 530 bara). Therefore, these conditions induced the adoption of bolts in Inconel 718 instead of Annealed Super Duplex ASTM A276 S32760. Because of the big difference between yield strengths of casted super duplex and Inconel 718 (σ/Inconel ≈ 2σ/super-duplex), and because of the predicted torque applied to the bolts based on ~90 percent of yield of Inconel 718 yield strength, a tightening test was performed to validate the design of threaded joining and the actual axial preload provided by the torque wrench to isolate the torque friction component.

A M80 bolt sample in Inconel 718 was strain-gauged and assembled with a part in casted super-duplex that reproduced the threaded joining. The test was made with three lubrication configurations – Molikote paste, Molikote spray, and no lubrication – with all possible combinations between nuts and bolts.

This test helped define two important conditions: the function law that correlates preload and torque with a specific lubrication configuration on the threads of nut and bolt and the washer contact area, and the max torque that’s possible without any potential damage to the threads of casted super-duplex. Figure 15 illustrates one example of function law of strain and preload versus tightening torque.

This pre-assessment also made it possible for the tightening procedure to successfully conduct a hydro test at 530 bara without any leakage between the mating surfaces of the two pump halves.

High-power BB5 pump for treated sea water injection

In 2009, GE Oil & Gas supported increased oil production from the M’Boundi field in Congo Brazzaville by supplying a complete package for sea water injection, including intake transfer, booster and main pumps. All the pumps were in the API 610 D2 material class. The M’Boundi field is 80 km from the coast, where the installed intake and export pumps feed the sea water pipeline up to the injection pumping system that mainly is comprised of a booster pump and main injection pump. Figure 16 summarizes the features of the main injection pump:

<table>
<thead>
<tr>
<th>Feature</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pump model</td>
<td>6x15 DDHF/6stgs</td>
</tr>
<tr>
<td>Speed</td>
<td>4200 rpm</td>
</tr>
<tr>
<td>Rated power</td>
<td>6.3 MW</td>
</tr>
<tr>
<td>Design pressure</td>
<td>255 bara</td>
</tr>
<tr>
<td>Max flow rate re-injection</td>
<td>800m3/h</td>
</tr>
<tr>
<td>Hydro-test pressure</td>
<td>380 bar</td>
</tr>
<tr>
<td>Efficiency</td>
<td>78.5 percent</td>
</tr>
<tr>
<td>Process fluid</td>
<td>Treated sea water</td>
</tr>
<tr>
<td>Materials</td>
<td>API 610 D2 material class</td>
</tr>
</tbody>
</table>
The BB5 pump was tested successfully, with vibrations widely within API 610 limits (see the spectra in Figure 17).

Spectra measured by the remaining probes raise analogue rotor dynamic and hydraulic considerations. Prior to June 2013, the main re-injection pump accumulated almost 19,000 hours, with just one interruption for mechanical seals (API 682 plan 11) substitution due to unexpected sand content.

Even though the main pump represents the core of the project scope, the project’s complexity and balance-of-plant integration make it unique. The project scope was the widest ever performed by GE as a pumping system manufacturer, since only civil works, piping connection, and cable routing weren’t covered by it. The scope architecture is displayed in the following single-line scheme:

Here is a list of the main items of balance-of-plant:
- VSDS unit
- Transformers
- Monitoring system and programmable logic
- Switch gears
- VSDS chilling units
- HV motor control centers
- LV motor control centers

Conclusions

In the coming decades, upstream oil and gas production certainly is one area where OEMs will have their technology and project management skills put to the test. The essential flexibility demanded by the diverse features of the plant, combined with the nature of the well, give OEMs an unprecedented chance to promote technology solutions that transcend their existing design comfort zone. (High power subsea installation is the best example of this challenge.) Since OEMs in the upstream market have a huge learning curve to travel, only the ones who invest in new, safer, and more reliable technology will form closer partnerships with oil field operators.
Reciprocating compressor cylinder’s cooling: a numerical approach using computational fluid dynamics with conjugate heat transfer

By Carlo del Vescovo, GE Oil & Gas, Pasquale Manicone, GE Oil & Gas, Lorenzo Bergamini, GE Oil & Gas

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