CENTRIFUGAL PUMPS FOR CO_2 APPLICATIONS

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ABSTRACT

There is a renewed interest in pumping CO_2 in the liquid and supercritical states for many enhanced oil recovery (EOR), and new carbon capture and sequestration (CCS) projects. Referenced centrifugal technology currently available in the industry can cover discharge pressures up to 25 MPa (3600 psi). GE Oil & Gas has now developed and successfully tested pumps that are capable of providing solutions up to 60 MPa (8700 psi). This paper reports the activities performed for this development and the results of the tests.

INTRODUCTION

GE Oil&Gas has a 50-year history in manufacturing a wide range of API 610/ISO13709 compliant centrifugal pumps for hydrocarbon processing, refineries, water injection and pipeline services.

A large fleet of these multistage pumps is currently operating with liquefied gases. The centrifugal pumps used for liquefied gases already include most of the technology necessary for CO_2 pumping because of similarities in viscosity, density, and compressibility. Therefore, this reference fleet has been the starting point for the development of the high pressure CO_2 injection pumps. We then utilized experience from our very high pressure gas re-injection centrifugal compressors that discharge up to 80MPa (11000 psi).

PUMPS FOR LIQUID CO_2

The BB5 type multistage barrel pump has been selected for the very high pressure CO_2 applications. 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 than a balancing drum, thus reducing the internal leakages that become a significant source of power losses when operating with low viscosity fluids. In addition, for pumps with such high DeltaP and long rotors, it also provides improved rotor-dynamic damping, even in worn conditions.

The main differences between pumps suitable for CO_2 pumping and conventional centrifugal pumps for hydrocarbon or water applications are described in the following paragraphs.

- The low density of warm supercritical CO_2 forces the pump to make high differential heads required to reach high discharge pressures. With limited number of stages available due to shaft span, high speed is required to achieve reasonable specific speeds and efficiencies. Rotordynamic assessment is the most critical part of the analysis and design process. In this application, the use of swirl brakes is often necessary, and magnetic bearings for dense phase acid gas pumping at high rotating speeds is also possible.

- The high compressibility of supercritical CO_2 must always be taken into account during the selection of the pump as in the case of centrifugal compressors. Therefore, the correct parameter expressing the work per unit of mass provided by the pump is the polytropic head instead of the differential head. Also, the definition of the efficiency is not unique. The polytropic efficiency is preferred because it is the parameter used for the compressors. It should also be noted that the traditional equations to calculate polytropic head and efficiency, based on perfect gas assumptions, give high errors for supercritical CO_2 in the region near the critical point (Pc=7.4MPa (1029psi), Tc=31°C). In this region the specific heat and the isentropic exponent cannot be used as average values between inlet and outlet because they do not vary monotonically.

- Another effect of compressibility relates to the stall characteristics of the impellers and the overall system pressure pulsations.
For high-speed pumps, the formation of potentially dangerous standing waves must be considered because the speed of sound is much lower than in liquid hydrocarbons or water.

- The equations of state used for the thermodynamic calculations of the flow internal to the pump must have an adequate level of precision. In the cases where the CO_2 is not pure, they may not be accurate enough, so several models must be used and a sensitivity analysis performed. An experimental program has been conducted by GE to increase the accuracy in those cases where contaminants significantly alter the physical behavior of the gas mixture.

- Another requirement, peculiar to CO_2 pumps, is the optimization of the overall train comprising the compressor and pump, including the selection of the intermediate pressure between the last compression stage and the pump inlet. The objective function to be minimized can be the total absorbed power, the total life cycle cost, or the cost per unit mass of CO_2 re-injected underground. The constraints to be considered in the optimization, can be the available cooling power, the total footprint, the cooling temperature when heat recovery is possible, and the operating range and different duty points that may apply in the future.

- For enhanced oil recovery applications with very high suction and discharge pressures, the structural design and material selection criteria have been derived from proven experience with CO_2 compressors.

- Three sealing technologies are available for selection depending on the type of process. When operating with pure CO_2, single face quasi-gas seals are the simplest solution. When the pump suction pressure must be increased significantly above the critical pressure and the process fluid is contaminated, dual or triple mechanical seals are preferred. Recent developments on wet seals (provided with light oil as buffer fluid) made them very robust to process upset conditions but have a higher power consumption and larger footprint for the auxiliaries compared with dry gas seals. Dry gas seal systems, specifically tuned for this application, provide the same reliability and MTBM of those installed on the compressors upstream of the pump.

- With the objective of maximizing MTBF and availability, a monitoring and smart diagnostic system has been developed that includes algorithms specific for CO_2 applications.

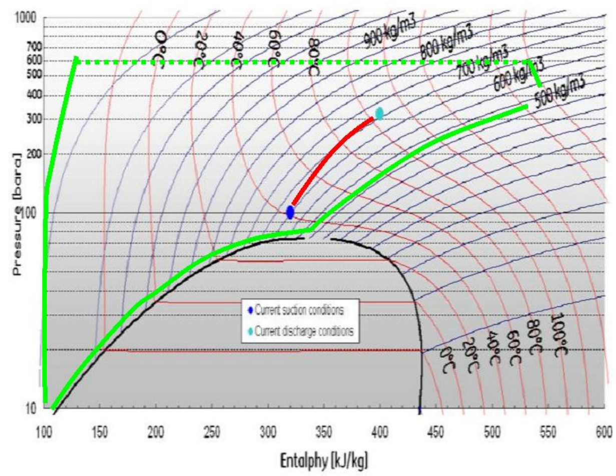

Figure 1. Operating Envelope of CO_2 pumps.

In Figure 1 the pressure vs. enthalpy (Mollier) diagram of pure CO_2 is shown. The envelope in which CO_2 pumps usually operate is also shown on the diagram. The envelope is bounded by the melting line on the left (about –50°C), the saturated liquid limit curve on the bottom and the minimum operating density on the right. The value of the minimum operating density, between 500 and 400 kg/m^3, depends on rotor-dynamics of the pump and the pressure ratio between discharge and suction pressure. As the ratio is increased the variation of the density can decrease the efficiency of the pump and the operating range unless differentiation of the stages is adopted. There is also a small segment on the upper right corner due to the max design operating temperature. The discharge pressure upper limit is not a physical limit but rather, is based only on available references. To define the minimum suction pressure a margin is added to the saturated liquid limit curve. This margin depends on the NPSH requirements of the pump considered, the accuracy of the control of the temperature of flow stream at pump inlet and on the expected variations in the composition of the gas. Within the envelope, there is one example of the polytropic transformation obtained during the tests.

PUMP FOR TUPI PILOT FPSO

This project is the first of its kind. It is a very high pressure gas reinjection train for enhanced oil recovery. The pump, shown in Figure 2, has a suction pressure of 30 MPa (4350 psi) and a rated discharge pressure of 54 MPa (7830 psi). The design pressure of the barrel casing is 67 MPa (API 10000 rating class). Because of the very high suction pressure, the mechanical seals are the most critical component. A triple seal arrangement was necessary to split the total pressure differential into smaller steps.

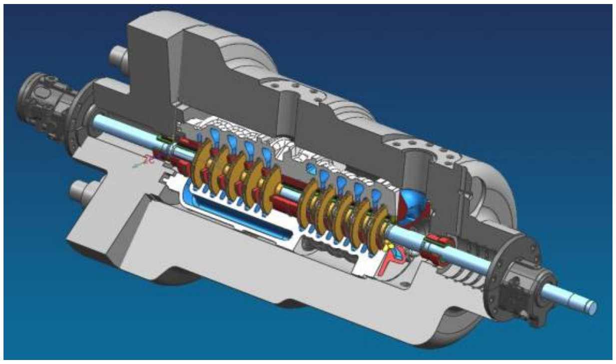

Figure 2. Cross section of the CO_2 injection pump for the TUPI pilot project.

The main development activities conducted for this project have been:
- Qualification of the prototype mechanical seals capable to operate up to 48 MPa of pressure difference.
- Modification of the impellers to improve the stall characteristics with the compressible mixture (see Figure 3) and avoid any surge phenomena.
- A detailed rotordynamic assessment using the experience, the tools and additional criteria developed for the centrifugal compressors.
- An experimental campaign to validate the thermodynamic properties (density, specific heats and speed of sound) since the gas mixture that this pump operates with contains, in addition to CO_2, up to 23% molar percentage of hydrocarbons.

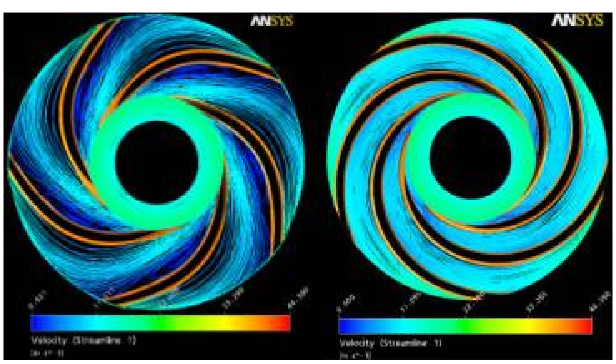

Figure 3. CFD's of Standard and New Impeller.

The graph in Figure 4 is part of the rotordynamic stability analysis. The damping factors achieved are high enough to guaranty stable operation even at the critical speeds thanks to the introduction of swirl brakes and additional damping devices. This was also confirmed by the tests.

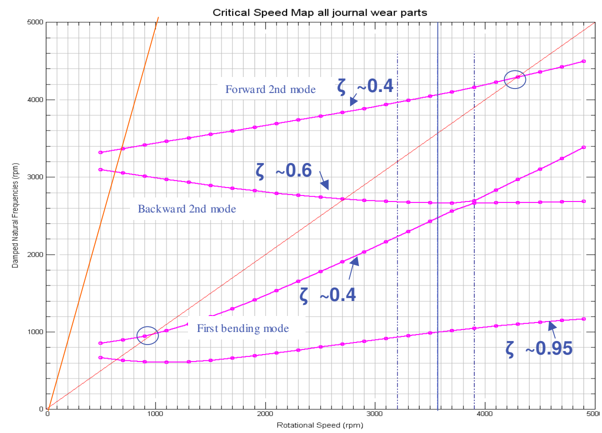

Figure 4. Damped Critical Speed Analysis.

From the inlet, at 30 MPa, to the final outlet pressure of 54 MPa, the gas temperature rises from 40°C to 76°C. With this particular gas composition, the temperature increase makes the total polytropic transformation very close to an iso-density (or isochoric) transformation, limiting the variation of the volumetric flow rate between the inlet and outlet.
From June to December 2010 the pump has been tested in a dedicated test loop with a mixture of CO_2 and N_2 in order to simulate the densities of the field process gas.
The same pump, retrofitted with dry gas seals, has been also tested with pure CO_2 at full design speed of 7600 rpm. With this configuration the differential pressure reached 45 MPa (6525 psi) with a flow rate of 35 kg/s. The stability of the rotor was such that the speed could have been increased if there were not limitations on the driver.
Figure 5 and 6 shows, respectively, a photo of the tested pump and the measured discharge pressure and total efficiency as a function of the volumetric flow rate at inlet conditions. The pump was operating at 7600 rpm with 10 MPa (1450 psi) suction pressure, whereas the suction temperature was varied between 15°C and 40°C, typical seasonal range of water cooled CO_2. It is apparent that, with the same suction pressure but different inlet density, the discharge pressure varies accordingly with variation of average density. A modification of the values and shape of the curve of the efficiency can be also noted. Further increase of suction temperature led to progressive decrease of the efficiency.

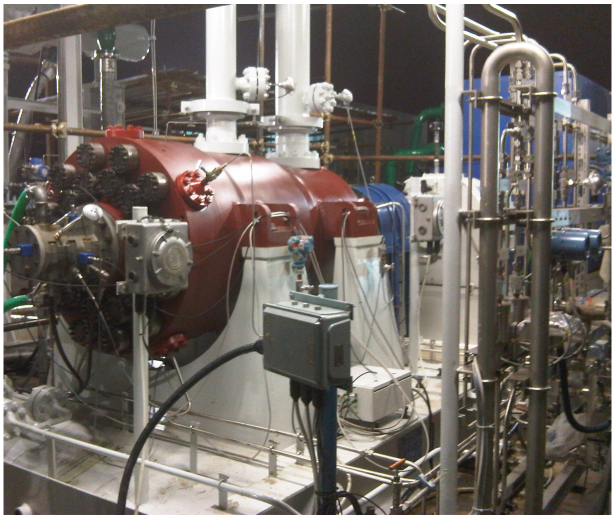

Figure 5. Tested Pump.

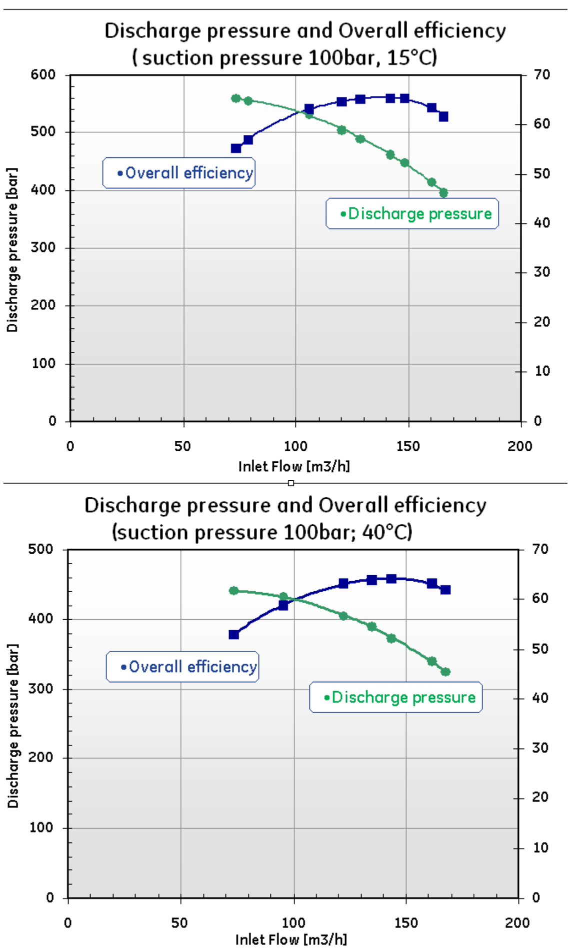

Figure 6. Experimental Test Results.

COMPRESSION + PUMPING

The usual starting sequence of a train composed by a compressor and a pump is: start the compressor with its recycle valve open; pressurize the pump until the minimum density is achieved; start the pump with its recycle valve open; close at the same time the two recycle valves while opening the discharge valve. There are different possible options for the control of the flow rate of a train composed by one or more centrifugal compressors and a centrifugal pump. The most flexible solution, but also the more expensive in terms of CAPEX, is a fully variable speed solution (both compressor and pump). On the other extreme (lower CAPEX and higher OPEX) there is the fixed speed solution with the frequent use of recycle and a discharge pressure control valve to maintain constant the discharge pressure. In this last case the differential head of the pump must be oversized.

An example of a complete train for enhanced oil recovery is shown in figure 7. It includes a compressor and a pump. The peculiarity of this case is that, without the need for a variable speed drive system, the pump can maintain a constant discharge pressure despite a wide variation in the gas mixture composition. This result is achieved by controlling the pump suction temperature and without an increase in the overall power as would be the case for a solution consisting of only a centrifugal compressor.

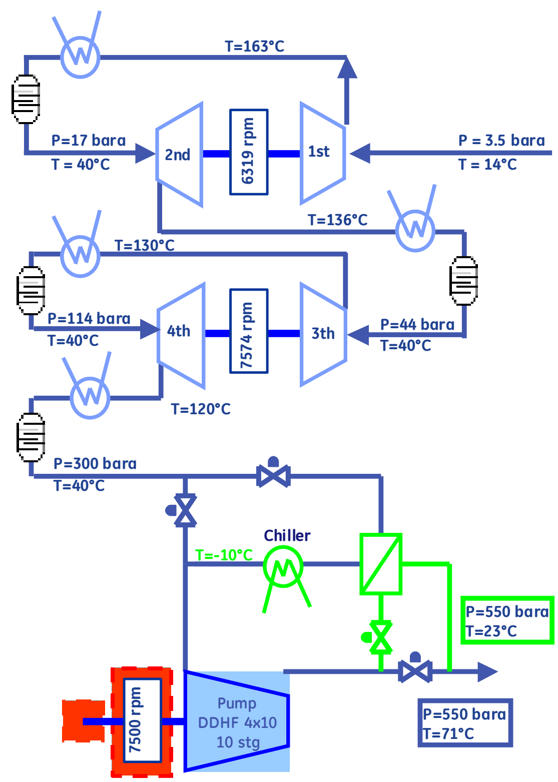

Figure 7. Example of Complete Compression+Pump Train

Figure 8 shows a map of compressor/pump train configurations for the range of CO_2 compression applications as a function of the inlet mass flow rate. Up to a discharge pressure between 10 and 13 MPa (1450-1885psi), the best technical solution, considering both CAPEX and OPEX, is achieved with integrally geared compressors because of the possibility to increase the number of intercoolers and better approximate the ideal isothermal compression. Above this discharge pressure, depending on the availability of sufficient cooling, centrifugal pumps lead to savings in the overall absorbed power between 5% and 15%.

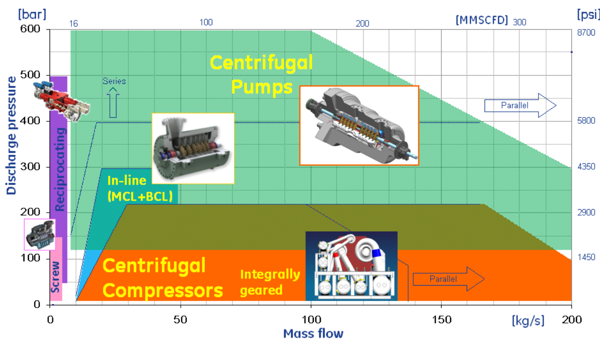

Figure 8. Technology map.

CONCLUSIONS

The re-injection of CO_2 underground, to where the carbon originated, is one of the possible mitigation actions available to counteract the greenhouse effect and the associated impact on climate change. The key parameters impacting the validity and sustainability of this technology are the cost and the energy that must be expended per unit of mass of re-injected CO_2. These parameters are strongly dependent on the overall power absorbed by the compression train. The use of centrifugal pumps as soon as possible in the overall compression process, combined with various forms of heat recovery, is the most favorable solution for this application and is now a proven technology ready for a variety of types of injection wells.