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QUALIFICATION TESTING OF A LIQUID CO_2 TURBOPUMP FOR CARBON CAPTURE AND SEQUESTRATION APPLICATIONS

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ABSTRACT

In order to reduce the amount of carbon dioxide (CO_2) greenhouse gases released into the atmosphere, significant progress has been made in developing technology to sequester CO_2 from power plants and other major producers of greenhouse gas emissions. The compression of the captured carbon dioxide stream requires a sizeable amount of power, which impacts plant availability, capital expenditures and operational cost. Preliminary analysis has estimated that the CO_2 compression process reduces the plant efficiency by 8% to 12% for a typical power plant. The goal of the present research is to reduce this penalty through development of novel compression and pumping processes. The research supports the U.S. Department of Energy (DOE) National Energy Technology Laboratory (NETL) objectives of reducing the energy requirements for carbon capture and sequestration in electrical power production. The primary objective of this study is to boost the pressure of CO_2 to pipeline pressures with the minimal amount of energy required. Previous thermodynamic analysis identified optimum processes for pressure rise in both liquid and gaseous states. At elevated pressures, CO_2 assumes a liquid state at moderate temperatures. This liquefaction can be achieved through commercially available refrigeration schemes. However, liquid CO_2 turbopumps of the size and pressure needed for a typical power plant were not available. This paper describes the design, construction, and qualification testing of a 150 bar cryogenic turbopump. Unique characteristics of liquid CO_2 will be discussed.

INTRODUCTION

In the effort to reduce the release of CO_2 greenhouse gases to the atmosphere, sequestration of CO_2 from Integrated Gasification Combined Cycle (IGCC) and Oxy-Fuel power plants is being pursued. This approach, however, requires significant compression power to boost the pressure to typical pipeline levels. According to Herzog [1] the power penalty for carbon capture can be as high as 27 to 37% for a traditional pulverized coal (PC) power plant and 13-17% for a typical IGCC plant. The compression represents a significant percentage of this total.

The goal of this research is to reduce this penalty through novel compression and pumping concepts by developing concepts to boost the pressure of CO_2 to pipeline pressures with the minimal amount of energy required. Fundamental thermodynamics were studied to explore pressure rise in both liquid and gaseous states. In addition to compression options, liquefying CO_2 and liquid pumping were explored as well. Thermodynamic studies by Moore and Nored [2] indicated that a reduction in power up to 35% is possible by pumping the CO_2 including the cost of liquefaction when combined in series with isothermal compression. A turbopump in this pressure and flow range required for sequestration is not readily available in the market place. Therefore, this paper describes an experimental test loop that was designed and constructed to perform qualification tests of a multi-stage turbopump originally designed for liquid nitrogen and liquefied natural gas (LNG) service. Performance and mechanical test data were gathered and will be presented according to ASME PTC 8.2 guidelines [3].

NOMENCLATURE

C_d	=	Orifice plate coefficient of discharge [unitless]
d	=	Orifice plate bore diameter [inches]
D	=	Outside diameter of pipe [inches]
E	=	Quality factor from Table A-1A or A-1B of [4]
g	=	Acceleration due to gravity [in/s^2]
g_c	=	Unit correction factor = 32.174 lbm-ft/s^2/lbf
ΔH_a	=	Actual pump head [ft]
N_c	=	323.279 = Unit conversion for English units
ΔP	=	Orifice differential pressure [inches H_2O]
P	=	Internal design gage pressure [psi]
P_d	=	Discharge pressure [psi]
P_s	=	Suction pressure [psi]
q_m	=	Mass flow rate [lbm/sec]
S	=	Stress value for material from Table A-1 of [4]
T	=	Pressure design thickness [inches]
v_d	=	Fluid velocity at discharge [ft/s^2]

v_s = Fluid velocity at suction [ft/s^2]
Y = Expansion factor (= 1.0 for liquid), unitless
Y = Coefficient taken from Table 304.1.1 of [4]
β = Diameter ratio, unitless
$\rho_{t,p}$ = Density upstream of the orifice [lbm/ft^3]
ρ_d = Discharge density [lbm/ft^3]
ρ_s = Suction density [lbm/ft^3]

DESIGN OF TEST FACILITY

The test rig consists of a newly constructed liquid CO_2 test loop and a commercial multi-stage turbopump. The pump is the smallest frame size for this product line but still retains the same configuration as large scale pumps that would be used for power plant applications. The smaller size was selected to keep the cost of the facility to a minimum, yet still provide valid performance and mechanical data. A schematic of the loop is shown in Figure 1.

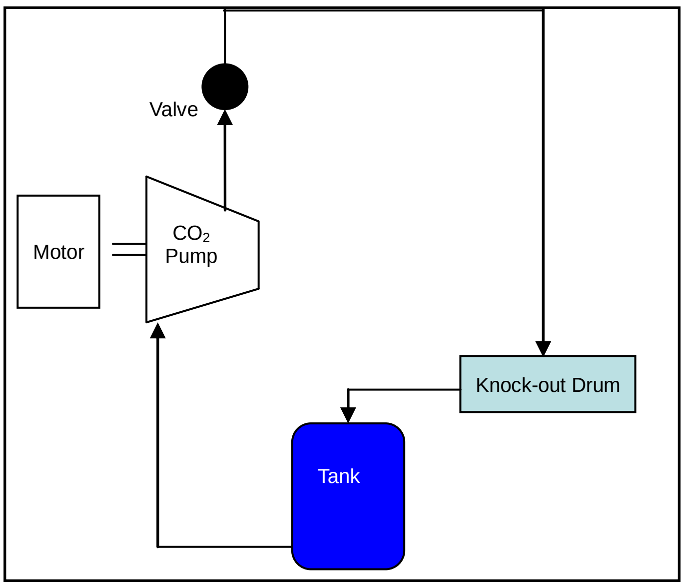

Figure 1. Schematic of Liquid CO_2 Pump Loop

The pump loop consists of a 12-stage pump driven by a variable speed electric motor. The pump parameters are shown in Table 1 and demonstrate the motivation for the smaller scale pump unit. Notice that the flow is about 1/9 of the full scale while the head requirements are identical.

Table 1. Summary of Sub-Scale Pump Parameters

		Full Scale	Test Scale
Power	hp (kW)	1807 (1348)	200 (149)
Flow	gpm (lpm)	968 (3663)	107 (405)
Head	ft (m)	4230(1290)	4230(1290)

The size of the loop components and the power requirements are driving the test scale design. The pump is fed from a 1,000-gallon pressurized vessel that maintains liquid CO_2 at its boiling temperature of -12°F at 250 psia. The discharge of the pump will feed an orifice flow meter run followed by a control valve that will drop the pressure from 2,215 psia down to 250 psia. The control valve will discharge into a knock-out drum for liquid/gas separation, since some flashing of the gas back to the vapor phase will occur. Finally, the liquid CO_2 will be returned to the main vessel through a drain line and the remaining gaseous CO_2 will be vented to the atmosphere through a back pressure control valve. Figure 3 shows a process and instrumentation diagram (P&ID)

The pump test will measure pump performance to quantify the power requirements. Also, mechanical performance, including vibration, temperatures, and seal flows, will be quantified. The test rig will be monitored for any sign of cavitation. A net positive suction head (NPSH) test will be performed in accordance with the ASME PTC 8.2 performance test code.

PRESSURE DESIGN OF SUCTION AND DISCHARGE PIPING

The material picked for both suction and discharge piping is SA333-6 seamless and welded steel for low temperature service, with a rated minimum allowable operating temperature of -50°F. Based on the chosen material properties and operating conditions, the minimum wall thickness of both suction and discharge piping was calculated according to the ASME B31.3 [4] standard using the equation below:

$$t = \frac{P*D}{2(S*E+P*Y)} \quad (1)$$

The minimum wall thickness calculated for suction (2-1/2 inch diameter) and discharge (1-1/2 inch diameter) piping using the B31.3 standard is 0.018 inches and 0.102 inches, respectively. Schedule 40 and 80 wall thickness pipes were chosen for the suction and discharge piping, respectively, which have a wall thickness of 0.203 inches and 0.200 inches. The pipe welds were X-ray inspected and all received a hydro test per ASME guidelines.

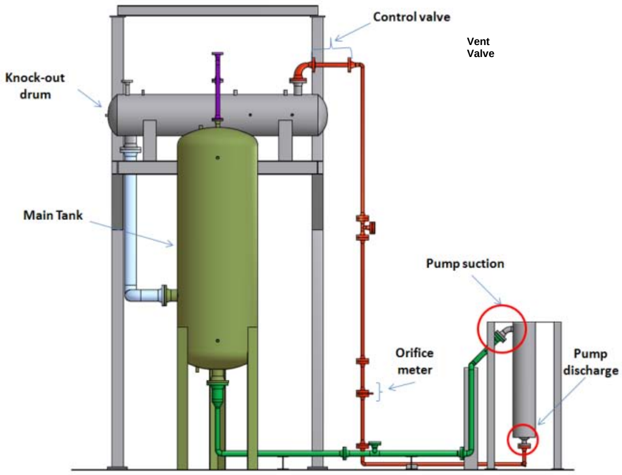

Figure 2. Front View of Loop Solid Model

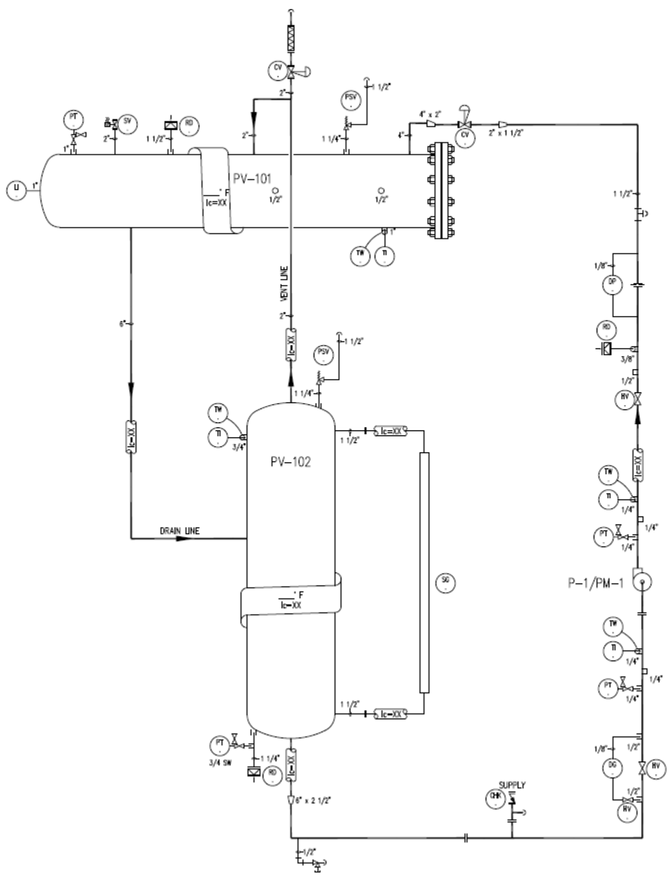

Figure 3. Test Loop Process and Instrumentation Diagram

PIPING THERMAL STRESS ANALYSIS

Due to the cold temperatures of the working fluid, a finite element model of the suction piping and discharge piping was created using ANSYS. For the suction model, the piping from the main tank liquid CO_2 outlet nozzle to the pump suction nozzle was included. A thermal stress analysis was performed to determine the displacements, stresses, strains, and forces in the piping caused by thermal growth and gravity. The loads and the response of the piping were assumed to vary slowly with respect to time. For the suction pipe, a maximum displacement and stress of 0.119 inches (3.0 mm) and 6,004 psi (41.4 MPa) were predicted as shown in Figure 4. Such predicted levels are well below the yield strength of the material and were deemed acceptable.

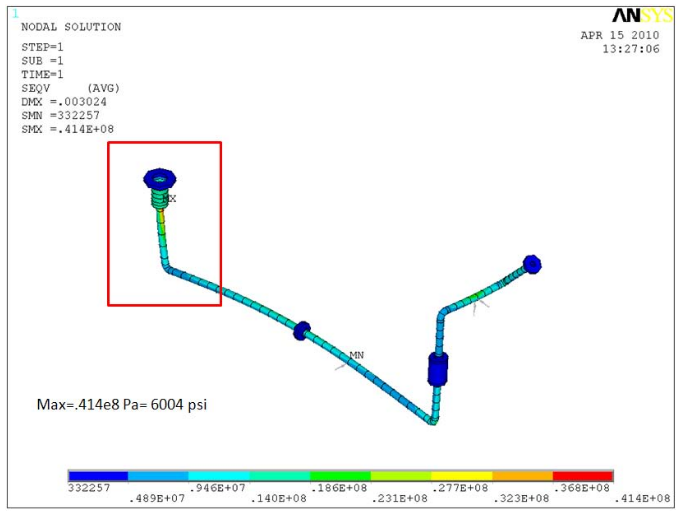

Figure 4. Equivalent Stress of the Suction Piping

A similar finite element model of the discharge piping was also created using ANSYS. Such model included the piping from the pump discharge nozzle to the knock-out drum nozzle. Again, a thermal stress analysis was performed to determine the displacements, stresses, strains, and forces in the piping caused by thermal growth of the piping and gravity. The applied loads and response of the piping were also assumed to vary slowly with respect to time. Maximum displacement and stress of 0.134 inches (3.4 mm) and 9,732 psi (67.1 MPa) were predicted as shown in Figure 5. Although higher stress than the suction, the stress is well below the yield strength for this material. The forces and moments acting on the pump nozzles were within the limits set by the manufacturer.

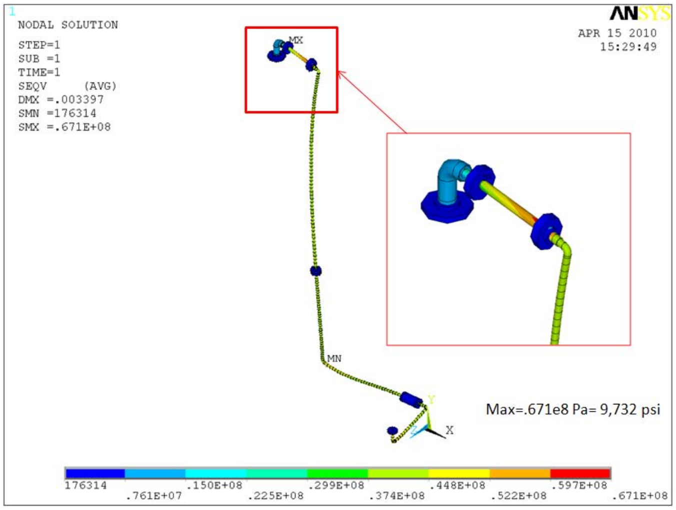

Figure 5. Equivalent Stress of the Discharge Piping

DATA ACQUISITION (DAQ) SYSTEM AND INSTRUMENTATION

The data acquisition and control system utilized the National Instruments CDAQ© system and programmed using Labview©. The purpose of this is to not only acquire data on demand but also provide control to the test loops. Currently, the data acquisition system is loaded with prescribed channel matrices for the pump loop tests and can acquire up 80 channels of data continuously at a sample rate of 1,000 Hz. Single-click data acquisition and continuous data logging at a prescribed interval has been implemented as well. Properties for CO_2 have been included using property tables for CO_2 for performance calculations. The interface accepts user input to control pump speed, throttle valve position, and vent valve position. Rather than controlling vent valve position manually, an automatic controller was developed to provide a constant tank pressure. Finally, mechanical monitoring data for the test rig (e.g., bearing temperature readings, seal gas pressure/temperature/flow, loop pressure, tank level) are included.

CONSTRUCTION OF TEST FACILITY

The pilot scale test facility consists of a newly constructed liquid CO_2 test loop and a custom developed test-scale multi-stage pump. Twenty percent of the welds in the test loop were subjected to and successfully passed X-ray testing. The test loop discharge piping was fabricated using threaded joint connections. The test loop piping and major components were insulated to reduce the amount of heat transfer to the test loop and reduce hazard due to the expected low operating temperature. The pump was installed and electrical connections were made to an 850 kW variable frequency drive (VFD) and all instrumentation is installed. Figure 6 shows a picture of the test loop. A seal control panel (Figure 7) is being used to provide conditioned CO_2 for the gas seal and provide conditioned purge gas for the protection chambers inside the pump.

Figure 6. Photo of CO_2 Pump Test Loop

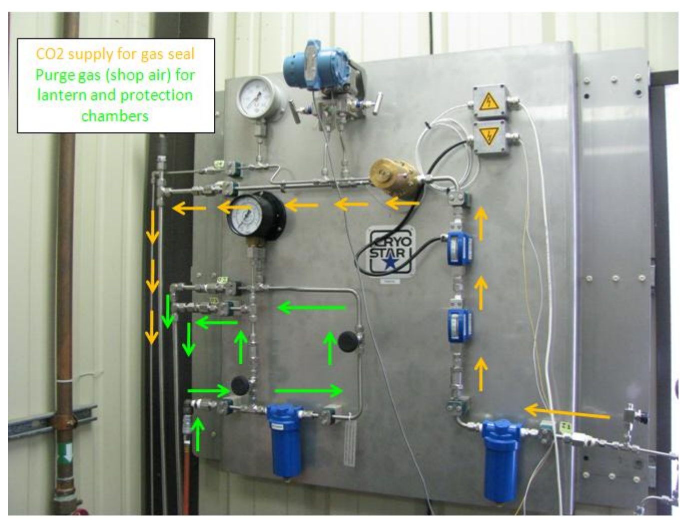

Figure 7. Seal Gas Control Panel

DESCRIPTION OF TEST PUMP

The pump used in this test program is a 12-stage centrifugal cryogenic turbopump mounted in a vertical orientation with a design speed of 3510 rpm with an impeller diameter of 240 mm. The pump is direct driven by a 250 kW induction motor. A dry gas seal is used to seal the shaft end and is supplied by gaseous CO_2 from bottles through the seal gas control panel. This panel regulates the seal gas at 3 bar above suction pressure and monitors the flow rate to the seal. The pump is also equipped with a heater to ensure that the pump bearing temperature remains within limits. Bearing temperature and seal gas delta-P are monitored by the data acquisition and control system. The pump has a 0.5 m net positive suction head (NPSH) requirement at the design point. The behavior of the pump at low NPSH values will be investigated. The pump comes with a support frame and is weather proof.

THERMODYNAMIC CALCULATIONS

This section provides results from two separate tests, which will be referred to as Tests 1 and 2. The flow calculations were performed per API 14.3 [5] for an orifice in liquid flow and head calculations were performed with static and dynamic head as shown below:

Mass flow is calculated in accordance with API 14.3 and ASME MFC-3M-1989 standards using the following equation:

$$q_m = \frac{\pi}{4} N_c C_d Y d^2 \sqrt{\frac{2\rho_{t,p}\Delta P}{1-\beta^4}} \qquad (2)$$

Since the pressure drop across the orifice plate is minimal compared to the absolute pressure and the stream is single phase, the constant density assumption used in Eq. 2 is valid (i.e. Y=1.0).

The pump head is calculated from pressure measurements at the pump suction and discharge locations. The calculation also involves densities obtained from liquid property data for CO_2 provided by REFPROP [6]. Once these have been obtained, the actual head produced by the pump is calculated as:

$$\Delta H_a = \frac{\frac{P_d}{\rho_d} - \frac{P_s}{\rho_s}}{g} \times g_c + \frac{v_d^2}{2g} - \frac{v_s^2}{2g} \qquad (3)$$

The discharge and suction fluid velocities are obtained from the mass flow rate as follows:

$$v_{d/s} = \frac{q_m}{\rho_{d/s} A_{d/s}} \qquad (4)$$

In Eq. (4), $A_{d/s}$ represents the cross-sectional area of the discharge and suction piping, respectively.

The hydraulic power (in hp) produced by the pump is calculated from the following equation:

$$HP = \frac{q_m}{\rho_d}\left(P_d - P_s + \frac{1}{2g_c}\left(\rho_d v_d^2 - \rho_s v_s^2\right)\right) \qquad (5)$$

The thermodynamic efficiency of the pump is calculated from the actual enthalpy rise and the isentropic enthalpy rise. The actual enthalpy at suction (h_s) and discharge ($h_{d,a}$) conditions is calculated using measured temperature and pressure at the suction and discharge along with the property data. The measured suction temperatures were not used due to thermocouple errors and heat transfer to the environment through the body of the probe. Instead, saturation temperatures (and densities for saturated liquid) were obtained from REFPROP at each suction pressure. The isentropic discharge enthalpy ($h_{d,i}$) is calculated by first using tabular data to obtain the entropy of CO_2 at suction saturation conditions and then evaluating the enthalpy at the discharge pressure and suction entropy. The thermodynamic efficiency of the pump is equal to the ratio of isentropic enthalpy rise to actual enthalpy rise:

$$\eta = \frac{h_{d,i} - h_s}{h_{d,a} - h_s} \qquad (6)$$

TEST RESULTS

The head vs. flow performance behavior is shown in Figure 8 at three different speeds lines: 1578 rpm, 2500 rpm, and 3510 rpm. The pump vendor provided factory test results using LN_2 for the nominal speed of 3510 rpm whereas the other two speed lines were calculated using speed scaling laws. The comparison of both Tests 1 and 2 against the results (2)
provided by the pump OEM show very good correlation. The Pump performed well for both Tests 1 and 2, matching the measured performance during factory testing on LN_2. The comparison of predicted and measured head for the design point correlates within 3% for all tests. Notice that the pump was tested to flows well below that in the factory testing.

The pump efficiency was calculated on actual power derived from energy balance calculations and isentropic power, as shown in Figure 9. Tests 1 and 2 show very good repeatability and correlation to the factory measurements on LN_2. The error bars show the effect of using saturation temperatures and the actual temperature readings obtained from the installed thermocouples, as shown in Figure 10 and is more pronounced at the lower head values.

(3)

While the efficiency values are relatively low compared to centrifugal compressors, the required pump power is an order of magnitude less than a comparable compressor as demonstrated by Moore and Nored (2008). Therefore the net power savings is still attractive.

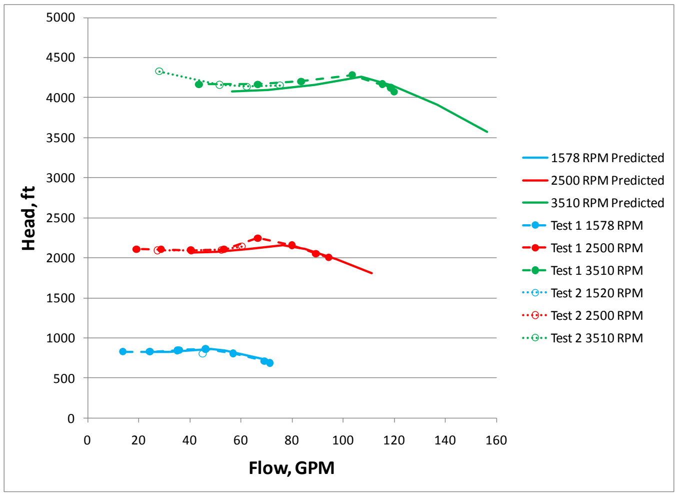

Figure 8. Pump Performance Plot (Head vs. Flow)

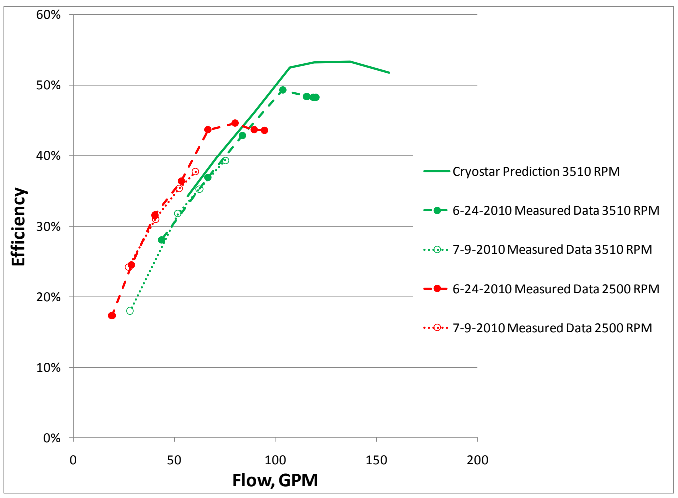

Figure 9. Pump Efficiency

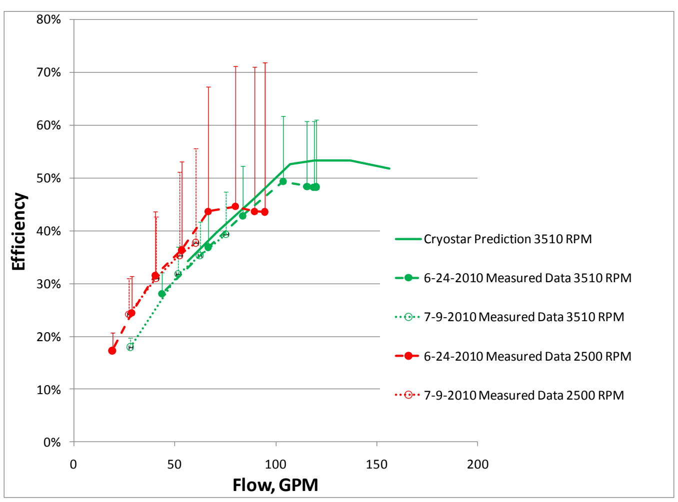

Figure 10. Pump Efficiency Showing Error Bars

The dynamic pressure measurement at the pump show little indication of unsteadiness for the design point as shown in Figure 11. The vibration levels observed during both tests were reasonable and did not exceed 0.2 inches per second (ips) (5.1 mm/sec) for the design flow operating point as shown in Figure 12. Some signs of cavitation were observed for the minimum flow operating point as seen in Figure 13. A sub-synchronous vibration occurred at minimum flow point as shown in Figure 14. The waterfall plot, shown in Figure 15, shows the behavior of the suction dynamic pressure when decreasing flow rate using the throttle valve. The subsynchronous pressure pulsations were not encountered until operating well below (30% of) the design flow of the pump near the minimum NPSH.

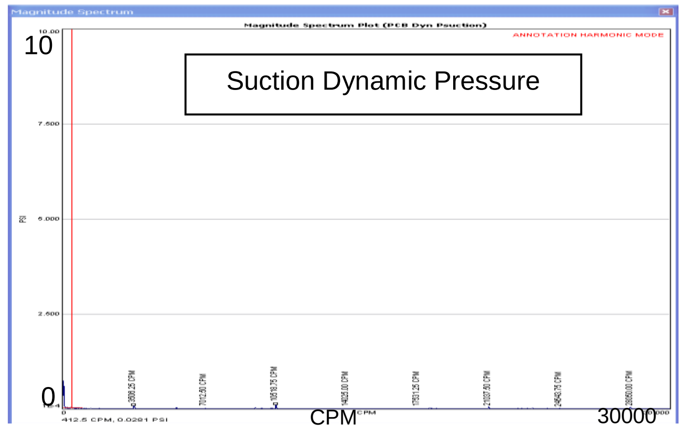

Figure 11. Suction Dynamic Pressure Spectrum for Design Operating Point

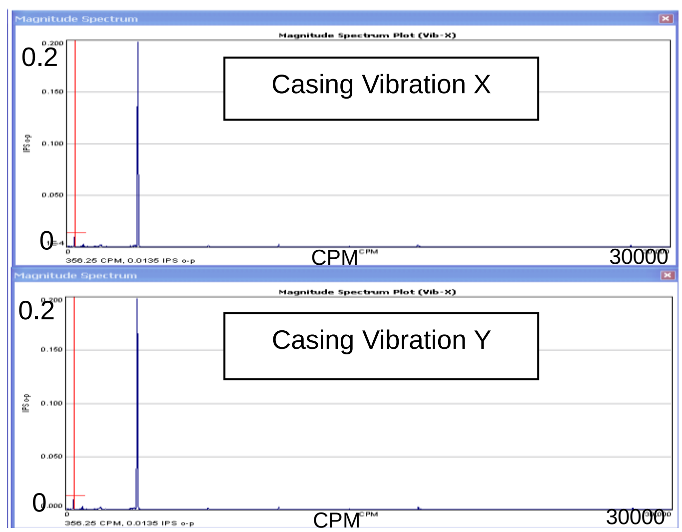

Figure 12. Pump Casing Vibration Spectrum for Design Operating Point

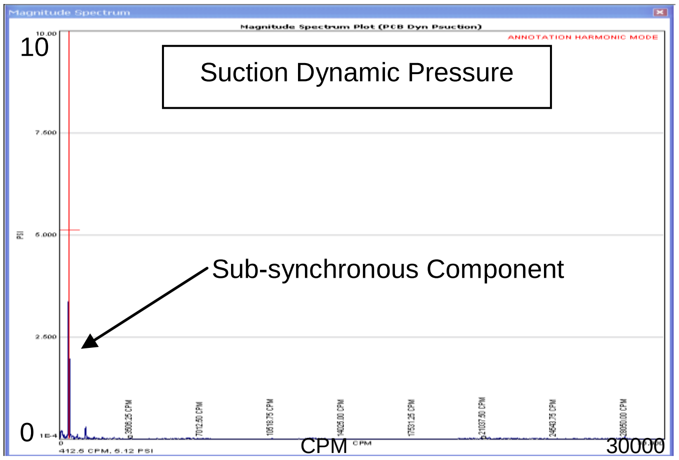

Figure 13. Suction Dynamic Pressure Spectrum for Minimum Flow Operating Point

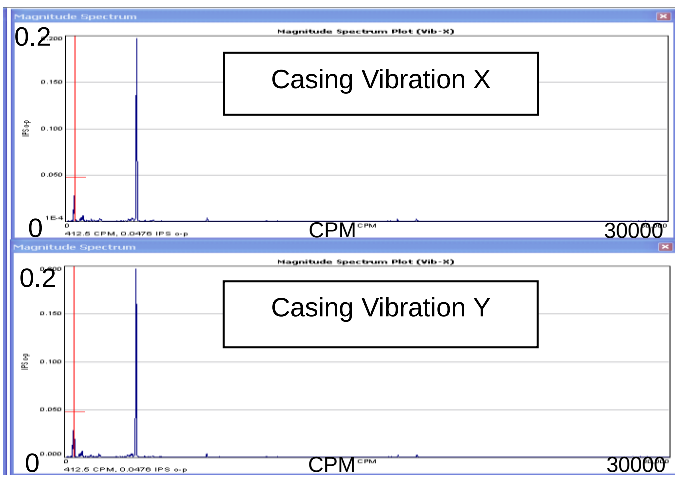

Figure 14. Pump Casing Vibration Spectrum for Minimum Operating Point

The pump was operated until the liquid level in the vessel reached the NPSH limit. No unusual pressure pulsations or vibration was observed.

CONCLUSIONS

An industrial cryogenic turbopump was adapted for liquid CO_2 service and tested in a newly developed closed loop test facility. Liquefaction followed by pumping a CO_2 stream is being proposed as a lower power alternative to traditional high pressure compression. Furthermore, once liquefied, volume reduction issues associated with high pressure ratio compressors are no longer a challenge when pumping due to the minimal specific volume change during pumping. The pump tests revealed close correlation to factory measurements on LN_2. Furthermore, the pump met all NPSH requirements showing no signs of cavitation despite being fed with a liquid at its boiling temperature. Only when the pump was throttled well below its design flow was any significant pressure pulsations and vibration observed. The pump is considered fully qualified for liquid CO_2 service.

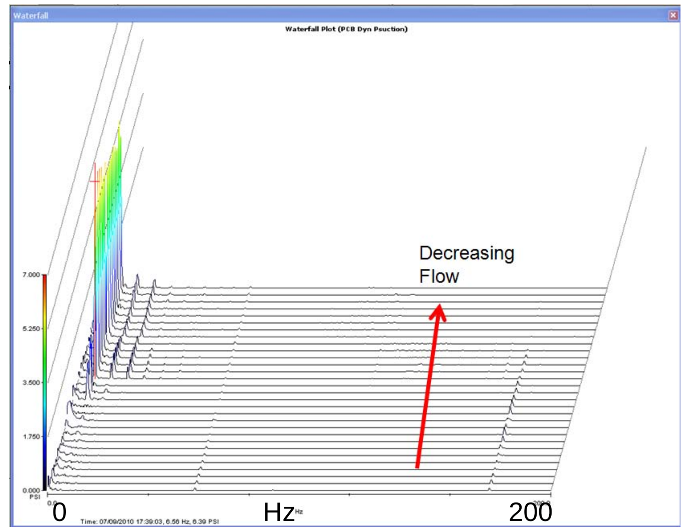

Figure 15. Dynamic Suction Pressure Waterfall While Throttling

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