PERFORMANCE EVALUATION OF A CENTRIFUGAL COMPRESSOR OPERATING UNDER WET GAS CONDITIONS

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ABSTRACT

This paper presents the results of performance testing of a single-stage centrifugal compressor operating under wet gas conditions. The test was performed at an oil and gas operator's test facility and was executed at full-load and full-pressure conditions using a mixture of hydrocarbon gas and hydrocarbon condensate. The effect of liquid was investigated by changing the gas-volume fraction between 1.0 and 0.97, which covers the range encountered by the operator during regular gas/condensate field production in the North Sea. Other parameters that were evaluated include the

compressor test speed, the suction pressure, and two different liquid injection patterns. During the tests, the machine flowrate was varied from near surge to choke conditions; hence, the evaluation covered the entire operating range of the machine. Although the test was primarily intended to evaluate the effects of the wet gas on the thermodynamic performance of the machine, the mechanical performance was also investigated by measuring the machine vibration levels and noise signature during the baseline dry gas tests as well as during the tests with liquid injection.

INTRODUCTION

Centrifugal compressor packages utilized for upstream gas processing often must operate under wet gas conditions in which the fluid handled by the compression package contains a mixture of liquid and gaseous phases. Typically, the liquid components of the mixture are separated from the gas stream before they enter the compressor by the use of scrubbers and separators located upstream of the compressor inlet. These devices are very large and heavy, requiring a large “footprint” (amount of floor space) as compared to the gas compression package. A compressor with the ability to directly handle wet gas without the need for separation equipment is very attractive from an economic standpoint, as it would drastically reduce the size, weight, and cost of the gas compression package. For the case of future subsea compression systems, this capability is even more attractive because of the high costs of deploying a compressor train and all of its associated equipment under water.

Wet gas compression (WGC) technology represents new opportunities for enhanced, cost-effective production from existing and future gas/condensate fields. Many oil and gas operators face future challenges in tail-end production, unmanned operation, and improved recovery from topside and subsea wells. This emphasizes the need to develop more robust compression systems, which can be designed for remote operation in unmanned topside installations, or could be designed for subsea operation for reinjection and/or transport boosting. The use of this technology for subsea boosting represents a new and exciting application for rotating equipment, which will allow new gas/condensate field production opportunities as well as enhanced recovery of existing gas/condensate fields and cost-effective production from marginal gas fields.

As mentioned above, these wet gas compression systems could be based on the use of a liquid tolerant dry gas compressor, which could boost a coarsely separated (via a scrubber) well-stream, however, an even more attractive solution would be the development of compression systems that can boost the well-stream directly. Many research projects and product qualification programs are currently underway to develop such a system either by modifying existing multiphase pump technology or by the adaptation of currently available gas compression technologies (Scott, 2004). Regardless of the choice of concept, the compressor solution should be able to tolerate liquid ingestion for an extended time without failure. For the case of subsea applications, the high cost associated with the retrieval of the compressor from the sea floor accentuates the importance of a reliable design.

The work presented herein served as an initial test to verify the multiphase boosting capabilities of a centrifugal compressor as well as to provide an oil and gas operator with data to compare the performance of this technology with other available wet gas compression concepts. It is important to state that the test compressor used for this investigation was not originally designed for wet gas boosting, nonetheless it provided an economically viable test bed for centrifugal compressor technology.

DESCRIPTION OF TEST VEHICLE

The test vehicle used for this work was a barrel-type, single-stage compressor, manufactured by the coauthors’ company. Said compressor was equipped with a high-head impeller, with a diameter of 0.384 m (1.26 ft), and a design flow coefficient of 0.02380. The compressor was originally designed to handle an inlet flow of 4332 Kg/min [2167 Am^3/hr (76,526.88 ft^3/hr)] of dry hydrocarbon gas (molecular weight of 18.49), with an inlet pressure of 130.2 bar (1888.4 psi)and a discharge pressure of 161.8 bar (2346.7 psi). Figure 1 shows a cross-section of the test compressor; the inlet and discharge nozzles are located at a 45 degree angle with respect to the top dead center of the machine. The original design of this machine, which dates to 1986, was not intended for wet gas service, and hence the internal geometry was not optimal. Nevertheless, in order to increase the reliability of the machine, the original rotor design was modified to accommodate an electron-beam welded and vacuum furnace brazed impeller with a shrink fit to the shaft. The rest of the machine remained the same (i.e., casing and stationary components). This compressor was equipped with a vaneless diffuser configuration.

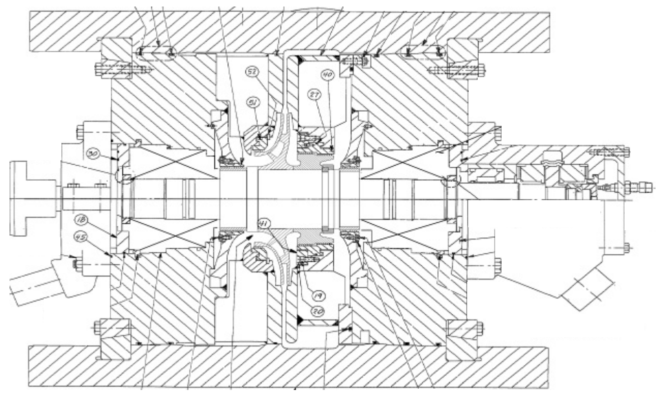

Figure 1. Cross-Section of the Test Compressor.

The compressor was driven by a 2.8 MW synchronous electric motor, through a speed increasing gearbox, with a gear ratio of 6.607. A variable speed drive permitted the operation of the compressor within its speed range of 6000 to 13,000 rpm.

The test compressor is utilized in the coauthor’s closed loop test facility, and was equipped to simulate the conditions expected for a centrifugal compressor operating under wet gas conditions. Figure 2 shows a schematic diagram of the test loop that was used for the evaluations. The major components of the test loop included a scrubber, the test compressor, a pump, a cooler, and a liquid injection module (mixer). The scrubber, here called guard separator, was used to separate the dry gas (saturated hydrocarbon mixture) from the liquid (hydrocarbon condensate) in order to permit accurate measurement of the massflow of each stream (liquid and gas). The liquid stream was measured with a Coriolis flowmeter while the gas stream was measured with a calibrated orifice plate.

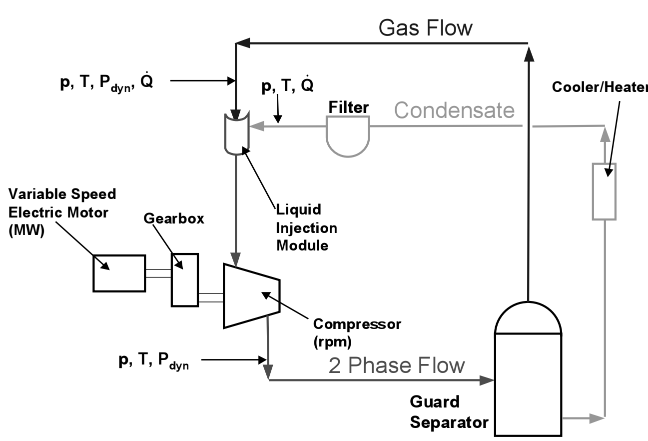

Figure 2. Schematic Diagram of the Wet Gas Test Loop.

A screw pump was used to handle the hydrocarbon condensate exiting the guard separator in order to increase its pressure to adequate values for injection into the gas stream. The liquid injection module permitted the introduction of the condensate into the gas upstream of the compressor inlet. The liquid could be introduced into the gas with two different patterns: as a uniformly distributed droplet mist or as a liquid film uniformly coating the wetted surface of the inlet pipe. This was obtained by injecting the liquid through specially designed nozzles for the case of droplet flow or through a circumferential slit for the case of liquid film flow. This required pressurization of the liquid phase and the use of a cooler/heater to regulate the temperature of the condensate to assure that it was injected into the gas stream at the same temperature as the gas. The liquid temperature was measured using calibrated PT-100 elements and static pressure measurements were made using calibrated pressure transducers.

The temperature and static pressure of the gas stream entering and exiting the compressor was measured using calibrated thermocouples and pressure transducers installed in the pipeline in accordance with the recommendations of ASME PTC 10 (1997). The measurements made on the gas exiting the compressor corresponded to that of the wet gas mixture, while the measurements of the liquid and dry gas streams at the inlet were made independently for each phase. The compressor discharge piping had a 45 degree slope upwards and also a diameter change from 0.203 to 0.305 m (.666 to 1.0 ft). Due to the risk of liquid accumulation in this piping, liquid hold up was monitored using a gamma ray densitometer. The same measurement was also performed upstream of the antisurge valve to detect any liquid accumulation. If this was detected, the test would have to be stopped due to the risk of injecting a liquid slug into the compressor.

The gas composition of the hydrocarbon mixtures utilized as test gas and liquid are shown in Table 1. The gas corresponds to an "export quality" lean hydrocarbon mixture (composed mostly of methane), which is typically commercialized for the European market, while the liquid corresponds to the condensate received from the Sleipner field, which lies in the Norwegian North Sea.

Table 1. Gas and Liquid Compositions (Compositions Shown as Molar Percentages).

Component	Export gas	Condensate	Composition in loop at 70 bar and 35 °C		
			Gas phase	Liquid phase	Total
N_2	0.756		0.854	0.089	0.531
CO_2	1.828		1.524	0.956	1.284
C1	90.373		90.933	25.920	63.474
C2	6.074		4.103	4.489	4.266
C3	0.844	0.024	0.341	0.955	0.600
I-C4	0.045	1.059	0.124	0.651	0.347
N-C4	0.064	7.690	0.654	4.632	2.334
I-C5	0.006	10.373	0.481	6.662	3.091
N-C5	0.006	12.015	0.454	7.856	3.581
C6	0.004	20.387	0.348	13.897	6.071
C7		18.616	0.137	12.932	5.541
C8		9.487	0.035	6.637	2.824
C9		4.392	0.008	3.084	1.307
C10+		15.957	0.005	11.239	4.750
Mole weight	**17.77**	**98.222**	**18.483**	**73.52**	**41.728**

Based on the volumes of gas and liquid and the filling temperature and pressure of the test loop, the composition of the gas and liquid streams and their associated thermodynamic states were evaluated using a thermodynamic property package in combination with the measured pressure and temperature. The thermodynamic property package is a precursor of a commercially available gas property package, which allows the combination of reliable fluid characterization procedures with robust and efficient algorithms to match fluid descriptions to experimental pressure, volume, and temperature (PVT) data. To increase the data accuracy, the gas and liquid densities were determined with a commercially available thermodynamic calculation software. The thermodynamic data at the inlet and discharge of the compressor were obtained and displayed online for each one of the measurement series and stored together with all of the measured and calculated parameters in one data file.

In addition to the instrumentation described above, the test loop was also equipped with dynamic pressure transducers, installed at the inlet and discharge piping of the compressor. These transducers were utilized to measure the fluctuating pressure components at the inlet and discharge of the machine. The signals from these instruments were displayed and recorded during the test in the form of frequency spectra, which enabled the test engineers to monitor the pressure signals in the process loop directly upstream and downstream of the compressor. These measurements were correlated to the noise level of the machine and permitted the comparison of this parameter while the machine operated under dry and wet gas conditions. In addition, the probes could be used to assist in correlating any possible subsynchronous rotor vibrations with pulsations in the gas stream (Marshall and Sorokes, 2000).

In order to minimize the complexity of the instrumentation for the initial test, and since the primary mission was the study of the thermodynamic performance change with liquids introduced to the gas stream, it was decided to forego installation of additional instrumentation, which would have given more insight into the mechanical reactions taking place in response to the various liquid loadings. As such, the installation of strain gauges on the impellers with their attendant installation complexities, as well as converting to active magnetic journal and thrust bearings, were held off for future test programs. The installation of a magnetic bearing shaft exciter (Moore, et al., 2002) onto the free-end of the compressor shaft would have provided a means to assess any variation of the rotor natural frequencies, as well as to determine any change to the rotordynamic stability of the compressor due to the addition of liquids into the gas stream. The use of this device was also left to a future test program.

THEORETICAL FOUNDATION

Single-Phase (Dry Gas) Performance

For a centrifugal compressor the primary variables of interest are the amount of flow delivered, the pressure rise produced, and the required power. The pressure rise and the efficiency of the gas compression are normally nondimensionalized to allow comparison of different geometries and operating conditions (Stahley, 2000; Colby, 2004). The polytropic compression process is selected for industrial compressors as it is better suited to handle the wide range of gases used in industry (Schultz, 1962). The equations for polytropic head coefficient, flow coefficient, efficiency, and power are shown below (ASME PTC-10, 1997).

$$\mu_p = \frac{W_p}{U^2} \tag{1}$$

$$U = \frac{\pi \cdot D \cdot N}{60} \tag{2}$$

$$W_p = \frac{n}{n-1} \cdot \left(p_2 \cdot v_2 - p_1 \cdot v_1 \right) \tag{3}$$

$$n = \frac{\ln\left(\frac{p_2}{p_1}\right)}{\ln\left(\frac{v_1}{v_2}\right)} \tag{4}$$

$$\phi = \frac{\dot{Q}_1}{2 \cdot \pi \cdot \frac{N}{60} \cdot D^3} \tag{5}$$

$$\eta_p = \frac{W_p}{h_2 - h_1} \tag{6}$$

$$P = \dot{m} \cdot \left(h_2 - h_1\right) \tag{7}$$

This formulation assumes a single-phase gas. If the compressor inlet stream contains both gas and liquid (i.e., wet gas), these equations must be modified. The primary area of interest is the definition of polytropic work, which impacts the polytropic efficiency and polytropic pressure coefficient.

Two-Phase (Wet Gas) Performance

The calculation procedure to estimate the performance of a machine operating under wet gas conditions is not described in any standard, as is the case for dry gas. However, the thermodynamic approach used for a single-phase gas, as stated above, can still be applied to a two-phase fluid. For the case of the single fluid model, the required modifications are shown below:

$$n_{TP} = \frac{\ln\left(\frac{p_2}{p_1}\right)}{\ln\left(\frac{v_{TP1}}{v_{TP2}}\right)} \tag{8}$$

$$W_p\Big|_{TP} = \frac{n_{TP}}{n_{TP} - 1} \cdot \left(p_2 \cdot v_{TP2} - p_1 \cdot v_{TP1}\right) \tag{9}$$

where the two-phase specific volume is based on the homogeneous model:

$$v_{TP} = \frac{1}{GVF \cdot \rho_g + \left(1 - GVF\right) \cdot \rho_l} \tag{10}$$

The gas-volume fraction (GVF) is defined by:

$$GVF = \frac{\dot{Q}_g}{\dot{Q}_g + \dot{Q}_l} \tag{11}$$

The fluid power was derived from electric power readings, using adequate calibration curves available from previous testing.

A different approach would be to consider a two fluid model where each phase is treated individually. The polytropic head is then calculated as:

$$W_p\Big|_{TP} = x_1 \cdot \frac{n}{n-1} \cdot \frac{R_0}{MW_{g1}} \cdot Z_1 \cdot T_1 \cdot \left(\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right) + \left(1 - x_1\right) \cdot v_{l1} \cdot \left(p_2 - p_1\right) \tag{12}$$

where the fluid quality is defined as:

$$x = \frac{\dot{m}_g}{\dot{m}_g + \dot{m}_l} \tag{13}$$

For the case presented in this work, the phase transition component was small due to a low pressure ratio through the machine and stable fluids. However, for higher pressure ratio multistage compressors, the phase transition contribution cannot be neglected. For the case at hand however, the effect of phase transition is only accounted for in the above expressions by changes in the value of the polytropic exponent due to a lower discharge temperature.

For this work and both of the performance calculation models presented above, the two-phase head coefficient and two-phase flow coefficient may be expressed as:

$$\mu_p\Big|_{TP} = GVF_1 \cdot \frac{v_{g1}}{v_{TP1}} \cdot \frac{W_p\big|_{TP}}{U^2} \tag{14}$$

$$\phi_{TP} = \frac{\dot{Q}_{Tot1}}{GVF_1 \cdot 2 \cdot \pi \cdot \frac{N}{60} \cdot D^3} \tag{15}$$

The efficiency for both cases is then expressed as:

$$\eta_p\Big|_{TP} = \frac{W_p\big|_{TP}}{P_{cs}} \tag{16}$$

where P_{cs} is the specific compressor shaft power, defined as the power consumed by the compressor per unit mass of wet gas.

The compressor two-phase efficiency calculated with the use of the single fluid model was found to be virtually equal to the one calculated via the two-phase fluid model (with a maximum deviation of 0.8 percent), so the results described in this work will be based on the single fluid model.

The performance of a wet gas compressor must be compared to the alternative, which would involve the separation of the fluid stream into individual phases (dry gas and condensate), and the subsequent boosting of the streams in separate compressor and pump units.

TEST PURPOSE AND VARIABLES

The wet gas testing presented in this work had several objectives. The first objective was to investigate the heat transfer rate between the gas and liquid condensate through the compressor and determine the state of thermal equilibrium. Another objective was to evaluate the compressor performance (power consumption, pressure ratio, and temperature ratio) and determine the effects of directly handling a wet gas mixture as opposed to dry gas compression. The impact of liquid ingestion on the compressor mechanical behavior and the pressure pulsations in the loop was also of interest, as was the liquid tolerance capacity and robustness of the compressor. Finally, the testing would create a foundation to evaluate the benefits and/or drawbacks of centrifugal compression technology as opposed to other multiphase boosting concepts.

To achieve the test goals, the performance of the machine was evaluated under several combinations of key parameters such as suction pressure, flowrate, rotational speed, gas-volume fraction, and liquid injection pattern following the data presented in Tables 2 and 3. The test program was completed in a time frame of about four weeks, during which the machine accumulated about 300 hours of operation under wet gas conditions.

Table 2. Key Test Parameters with Range of Variation.

Key Test Parameter	***Values***	***Units***
Suction Pressure	30 70	bar
Machine Speed	9651 10723	rpm
Gas Volumetric Flowrate	1600 1800 2000 2200 2400	Am^3/hr
Gas-volume Fraction	1.0000 0.9994 0.9950 0.9900 0.9800 0.9700	N/A
Liquid Injection Pattern	Uniform Droplet Fluid Film	N/A

Table 3. Test Matrix Agenda.

Test Number	Suction Pressure	Machine Speed	Gas Flowrate	Gas-Volume Fraction	Liquid Injection Pattern
1	30	9651	All	1.0	N/A
2	70	9651	All	1.0	N/A
3	70	10723	All	1.0	N/A
4	70	9651	All	All	Droplet
5	70	9651	All	1.0, 0.995, 0.98 2200 Am^3/hr :All	Fluid Film
6	70	10723	All	1.0, 0.995, 0.98 2200 Am^3/hr :All	Droplet
7	30	9651	All	All (*)	Droplet
8	30	9651	All	1.0	N/A

(*) The test point at 2400 Am^3/hr and GVF = 0.97 was not completed due to test loop limitations.

The quality (x) of the wet gas mixture being injected into the compressor was dependent on the predefined gas-volume fraction as well as the suction pressure at which the test was being executed. Table 4 presents the values of quality for each GVF used for the tests for suction pressures of 30 and 70 bar (435.1 and 1015.3 psi).

Table 4. Quality of the Wet Gas at the Compressor Inlet.

GVF	Quality at 70 bar	Quality at 30 bar
1.0000	1.0000	1.0000
0.9994	0.9931	0.9824
0.9950	0.9454	0.8699
0.9900	0.8959	0.7706
0.9800	0.8100	0.6254
0.9700	0.7377	0.5218

RESULTS AND DISCUSSION

Single-Phase (Dry Gas) Performance

Prior to the introduction of liquids into the test loop, the thermodynamic performance of the machine was evaluated to establish the baseline performance while it was operating under single-phase (dry gas) conditions (tests 1, 2, and 3 of Table 3). This baseline would be used for comparison with the results obtained during the operation of the machine under wet gas conditions. In addition, the baseline performance would be used to evaluate if the injection of liquids during the multiphase testing had produced any noticeable effects (performance changes) after the tests were concluded. This would be done by running another series of dry gas performance tests and comparing the results to the initial baseline.

Figure 3 shows the results of the single-phase performance tests that were executed before and after the evaluation with two-phase (wet gas) flow. As seen in the figure, the performance levels of the machine (polytropic head coefficient and efficiency) have remained unchanged, that is, there is no evidence to suggest that the ingestion of liquid produced any significant variation in the machine's performance levels. This implies that the compressor flowpath was not subjected to any significant damage during the wet gas tests. A boroscopic inspection of the inlet and impeller eye areas executed after the tests confirmed that there was no evidence of internal damage.

Two-Phase (Wet Gas) Performance

Figures 4, 5, and 6 show the performance of the compressor exposed to liquid with up to 3 percent of the inlet volume flow (GVF of 0.97). The wet gas tests are shown together with dry gas results for comparison. Figure 4 presents a comparison of the compressor performance at two different speeds, while operating at a suction pressure of 70 bar, with the liquid being injected with a uniform droplet pattern. Figure 5 shows the performance of the compressor at two different suction pressures [p_1 = 30 and 70 bar (435.1 and 1015.3 psi)], while operating at the same speed (9651 rpm) and with the same liquid injection pattern (uniform droplet). Figure 6 shows a comparison of the machine performance for both liquid injection patterns (droplet and fluid film), with the machine operating at the same suction pressure [70 bar (1015.3 psi)] and the same speed (9651 rpm).

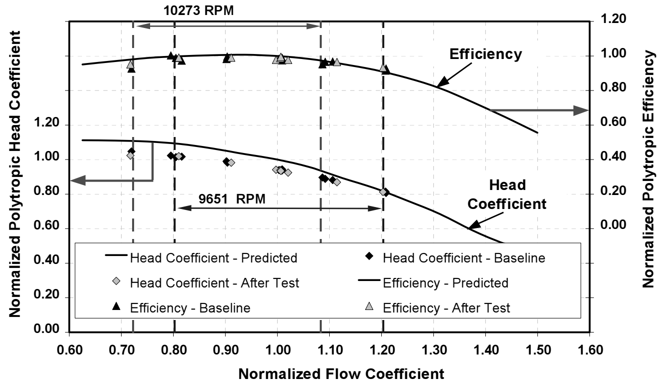

Figure 3. Single-Phase Thermodynamic Performance Evaluation. Baseline Versus After-Test Conditions.

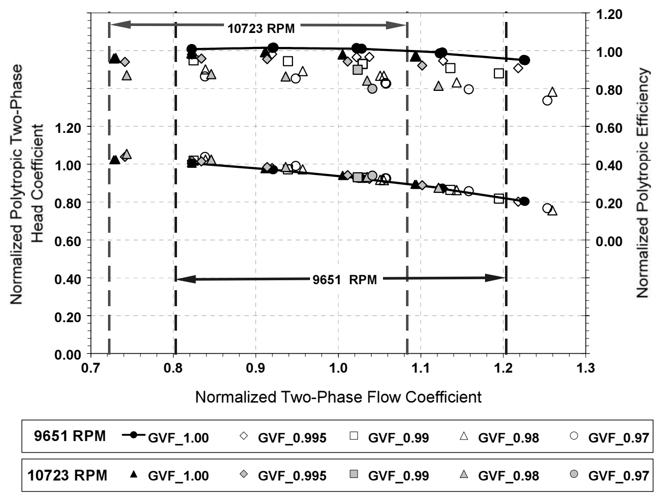

Figure 4. Two-Phase Thermodynamic Performance Evaluation. Effects of Test Speed (p_1 = 70 bar, Droplet Injection Pattern).

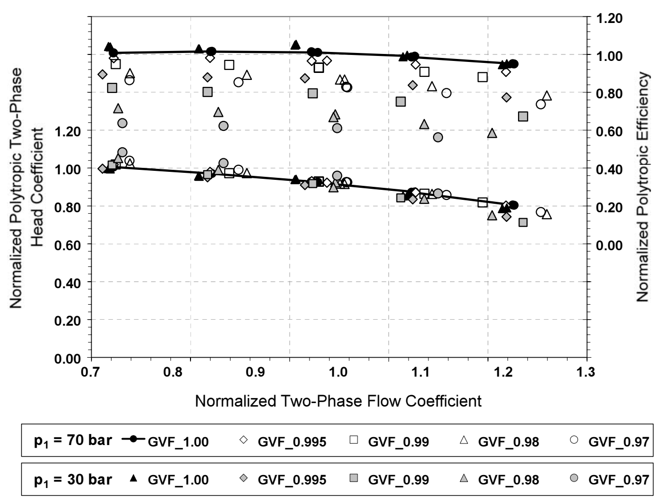

Figure 5. Two-Phase Thermodynamic Performance Evaluation. Effects of Suction Pressure (9651 rpm, Droplet Injection Pattern).

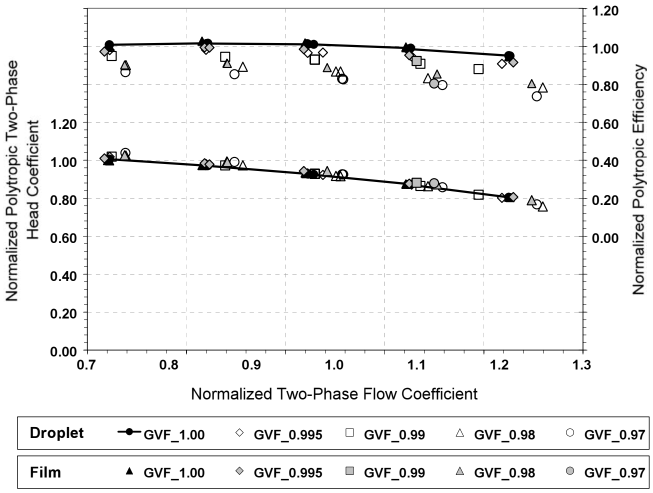

Figure 6. Two-Phase Thermodynamic Performance Evaluation. Effects of Liquid Injection Pattern (9651 rpm, p_1 = 70 bar).

As seen in Figures 4 through 6, the redefined polytropic head and flow coefficients [Equations (14) and (15)], valid for two-phase flow, are capable of merging the data from the various wet gas operating conditions with those corresponding to the dry gas operation. Furthermore, these figures show that the efficiency drops when the amount of liquid is increased and that this effect is much more pronounced at lower pressures. The more pronounced effect at lower pressures is due to the increasing density difference between the gas and the condensate when the suction pressure is reduced while maintaining a constant GVF. The increasing density difference leads to a considerable increase in the mass fraction of liquid entering the compressor. As shown in Table 4, at a GVF of 0.97, the mass fraction of liquid (1-x) increases from 0.2623 at 70 bar (1015.3 psi) to 0.4782 at 30 bar (435.1 psi).

The reduction in the machine efficiency as the mass fraction of liquid increased is due to larger internal losses in the compressor. The test vehicle was not instrumented internally, so the available data were insufficient to identify the source of the increased losses. The compressor manufacturer plans to evaluate this issue by performing two-phase computational fluid dynamics (CFD) simulations of the compressor at the test conditions. Another way to provide insight into this phenomenon would be to run additional tests with internal instrumentation.

Figure 6 shows very little difference between the performance of the machine when subjected to uniform droplet and fluid film injection patterns. Consequently, it is thought that the compressor inlet serves as a mixing element and makes the flow pattern inside the impeller relatively independent of the injection method. This effect will also be evaluated via two-phase CFD calculations. The distance between the point of liquid injection and the center of the impeller was limited to approximately three times the internal diameter of the compressor suction nozzle. This was done in an effort to ensure that the two-phase flow pattern was maintained from the point of injection up to the impeller inlet.

In general, for the figures discussed above, there is a tendency of a larger departure from a common head coefficient curve at 30 bar (435.1 psi) when the deviation between the operating flowrate and the impeller design flowrate increases (GVF < 0.99).

The compressor specific power consumption is shown in Figure 7. The data shown in the figure correspond to the tests with the compressor operating at 9651 rpm, with a suction pressure of 70 bar (1015.3 psi), and the liquid being injected in a uniform droplet pattern; nevertheless, the same behavior was observed for the other wet gas test conditions. As shown in the figure, when liquid is injected, the required specific power is reduced. To properly evaluate the specific power associated to wet gas compression, these data have to be compared with the data that would be obtained if the same amount of liquid and gas were to be transported between the same two pressures (as independent streams). A separate gas and liquid boosting case is included in the figure assuming a GVF of 0.97. As can be seen the specific power consumption is lower than the values obtained for wet gas compression. The separate boosting data were based on the same pressure difference. However, when wet gas compression is utilized, the pressure drop in the scrubber may be avoided and the required pressure boost is less than the one depicted in Figure 7. Furthermore, the possibility of simplifying the compressor system by avoiding the scrubber and the appurtenant instrumentation must also be considered when the system is evaluated.

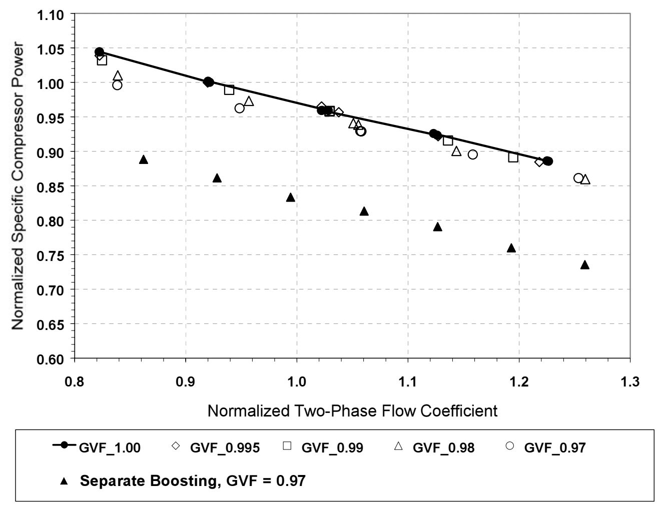

Figure 7. Two-Phase Thermodynamic Performance Evaluation. Compressor Specific Power Consumption as a Function of GVF (p_1 = 70 bar, 9651 rpm, Droplet Injection Pattern).

Figures 8 and 9 show the effects of liquid injection on the pressure and the temperature ratios (discharge/inlet) through the compressor. Once again, the data shown in the figures correspond to the tests with the compressor operating at 9651 rpm, with a suction pressure of 70 bar (1015.3 psi), and the liquid being injected in a uniform droplet pattern; nevertheless, the same behavior was observed for the other wet gas test conditions. The pressure ratio increased due to the increased density (and molecular weight) of the fluid processed by the compressor. The temperature ratio slightly decreased for the case of liquid injection. This observation is explained by two mechanisms:

- Increased internal energy of the liquid phase, and
- A certain degree of liquid evaporation has occurred as the liquid passed through the machine.

These results will also be evaluated by the compressor manufacturer via two-phase analytical simulations, as they will be of great importance when designing multistage machines for wet gas operation. The change of phase of the liquids inside the compressor may cause a mismatch between the subsequent stages of the machine, which may lead to performance shortfalls.

Dynamic Pressure Measurement

Figures 10 through 15 show the frequency spectrum of the dynamic pressure signals measured close to the inlet and discharge flanges of the machine under three different test conditions (note that the scales on all plots are the same). Figures 10 and 11 correspond to the machine operating at 9651 rpm with a suction pressure of 70 bar (1015.3 psi) and show the effects of the liquid being injected in a uniform droplet pattern. Figures 12 and 13 correspond to the same compressor operating condition but with the

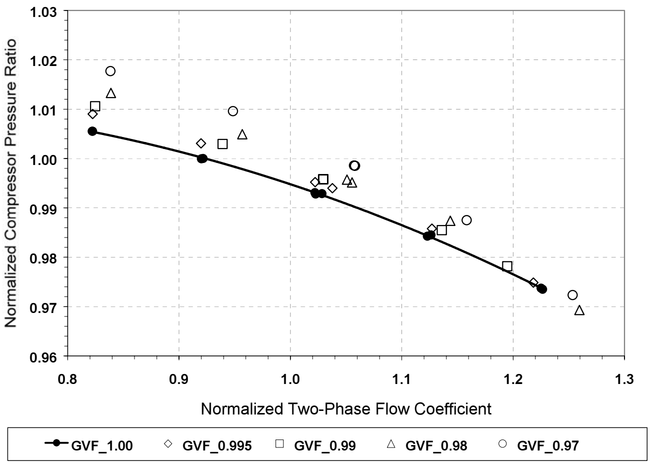

Figure 8. Two-Phase Thermodynamic Performance Evaluation. Compressor Pressure Ratio as a Function of GVF (p_1 = 70 bar, 9651 rpm, Droplet Injection Pattern).

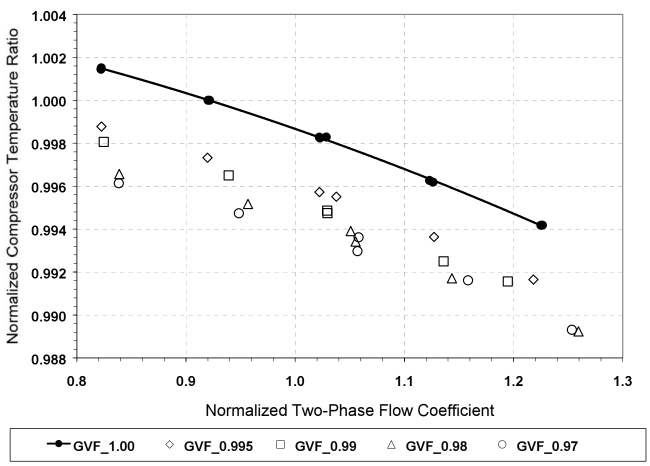

Figure 9. Two-Phase Thermodynamic Performance Evaluation. Compressor Temperature Ratio as a Function of GVF (p_1 = 70 bar, 9651 rpm, Droplet Injection Pattern).

liquid injection being performed as a fluid film in the periphery of the pipe. Finally, Figures 14 and 15 correspond to the same compressor speed, with a suction pressure of 30 bar (435.1 psi) and the liquid being injected in a uniform droplet pattern.

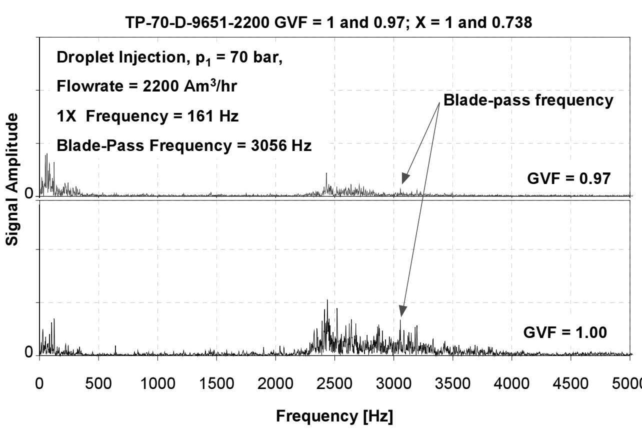

Figure 10. Effects of Two-Phase Flow over the Dynamic Pressure Measurements at the Machine Inlet (p_1 = 70 bar, 9651 rpm, Droplet Injection Pattern).

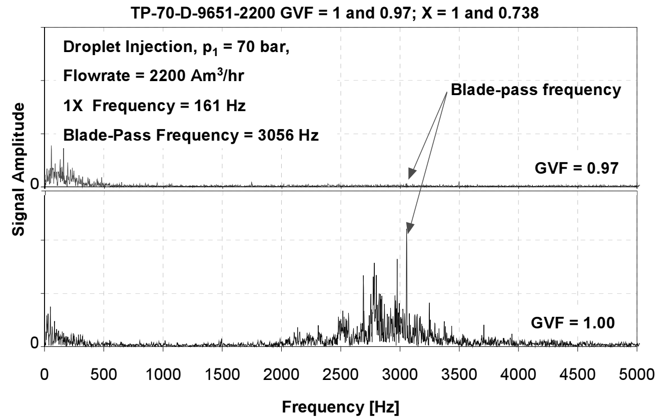

Figure 11. Effects of Two-Phase Flow over the Dynamic Pressure Measurements at the Machine Discharge (p_1 = 70 bar, 9651 rpm, Droplet Injection Pattern).

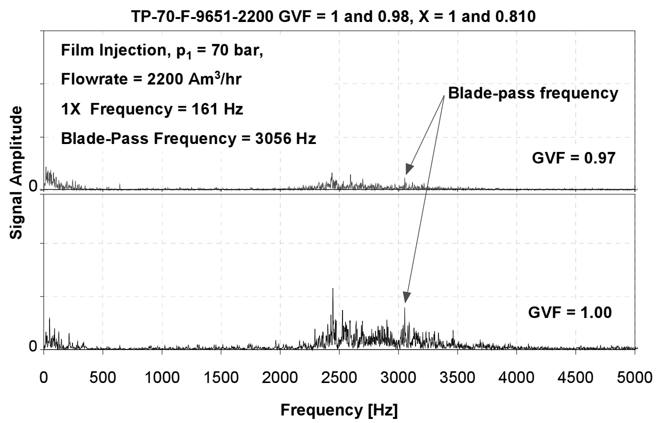

Figure 12. Effects of Two-Phase Flow over the Dynamic Pressure Measurements at the Machine Inlet (p_1 = 70 bar, 9651 rpm, Film Injection Pattern).

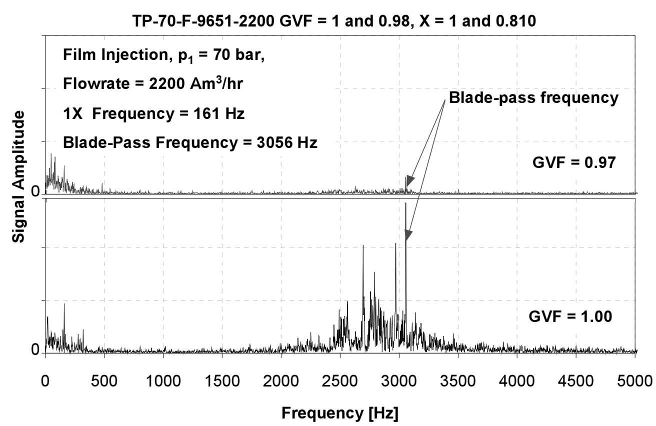

Figure 13. Effects of Two-Phase Flow over the Dynamic Pressure Measurements at the Machine Discharge (p_1 = 70 bar, 9651 rpm, Film Injection Pattern).

It is important to state that the low frequency amplitudes (noise) that are evident on the spectrum plots are due to the fact that the data acquisition and display system that was used to capture the data, and to generate these figures, did not have the capability to average several fast Fourier transform (FFT) samples. This hindered the ability to reduce the random noise components of the spectra. Furthermore, the pressure sensors were not installed flush to the pipe wall. They had a small recess [about 25 mm (.98

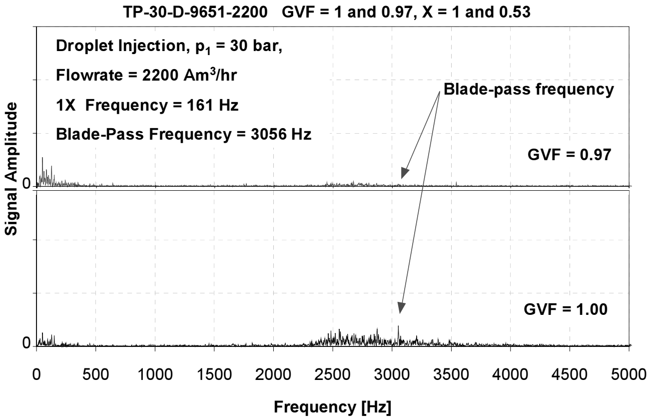

Figure 14. Effects of Two-Phase Flow over the Dynamic Pressure Measurements at the Machine Inlet (p_1 = 30 bar, 9651 rpm, Droplet Injection Pattern).

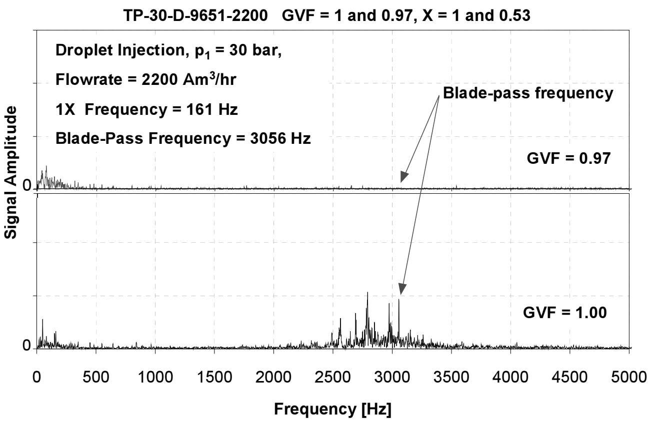

Figure 15. Effects of Two-Phase Flow over the Dynamic Pressure Measurements at the Machine Discharge (p_1 = 30 bar, 9651 rpm, Droplet Injection Pattern).

inches)] that may have contributed to the generation of some noise components in the FFTs.

As may be seen in the figures, for the case in which the compressor is handling dry gas (GVF of 1.0), the frequency spectrum plots show a variety of peaks in the vicinity of the impeller blade passing frequency. On the other hand, when liquids are injected into the flow (GVF = 0.98 or 0.97), the high-frequency components of the spectrum vanish. This behavior suggests that the liquids injected into the process gas dissipate the acoustic signals (evident by a reduction in the audible noise level) and dampen the pressure fluctuations. This phenomenon was observed for all of the test conditions that were evaluated as shown in the test matrix above (refer to Tables 2 and 3).

Rotordynamic Behavior

Although the main objective of the testing described in this paper was to establish the effects of wet gas conditions over the aero/thermodynamic performance of a centrifugal compressor, another objective of similar importance was to evaluate the effects of liquid ingestion over the rotordynamic behavior of the machine. For this, the shaft vibration was measured via eddy current proximity probes installed at both ends of the machine (i.e., at the driven and nondriven ends). The test compressor was originally supplied with a pair of proximity probes at each journal bearing to measure the shaft vibration in the horizontal and vertical directions. In addition, the axial position of the shaft was also measured via proximity probes at the free end of the machine. The signals from these probes were displayed in the control room and were linked to the process control system to provide appropriate machinery protection. Said signals were also connected to a proprietary inhouse data acquisition and reduction system, which permitted the frequency spectrum of the shaft vibration signals to be displayed during the tests. These frequency spectra were utilized in the evaluation of the rotordynamic behavior of the machine by comparison of the spectra obtained under dry and wet gas conditions.

The test compressor had a bearing span of 0.727 m (2.39 ft), with an impeller bore of 0.132 m (5.2 inches), and a journal diameter of 0.076 m (2.99 inches). The shaft was mounted on tilt-pad journal bearings, each of which had five pads, in the load-on pad configuration. The first natural frequency of the rotorbearing system is near 9800 cpm. This mode is well damped and is not a critical speed. The first critical speed of the compressor was determined (analytically) to be between 20,300 to 21,130 cpm. This value is larger than the maximum speed at which the compressor would be tested (10,723 rpm), so smooth operation was expected within the operating envelope that would be used for the tests (9651 to 10,723 rpm).

Figure 16 presents the frequency spectra (FFT) of the shaft horizontal vibration component, measured at the driven end of the machine, while it was operating at 9651 rpm, with a suction pressure of 70 bar (1015.3 psi) and handling a volumetric flow of 2200 Am^3/hr (77,692.27 ft^3/hr). The bottom spectrum corresponds to operation with a GVF of 1.0 (dry gas); while the top spectrum shows the behavior of the machine while handling a two-phase gas mixture, with a gas-volume fraction of 0.97, which represents a gas quality (x) of 0.738. For this case, the liquid was being injected into the gas with a uniform droplet pattern. As may be seen in the figure, the vibration spectra for both the dry gas and the wet gas compression are virtually the same. This suggests that the rotordynamics of the machine remain unaffected by the liquid injection for the case of the condensate being injected with the uniform droplet pattern. This is due to the fact that if the liquids are uniformly distributed throughout the gas, they do not produce any significant source of rotor excitation as they pass through the impeller, nor do they affect the rotor unbalance levels (1× vibration component). The above similarity was also encountered when comparing the vibration spectra corresponding to GVF values of 0.98 and 0.99. The behavior of the machine while it was operating at 10,723 rpm, under the same suction pressure and liquid injection mechanism, presented similar characteristics and hence will not be shown.

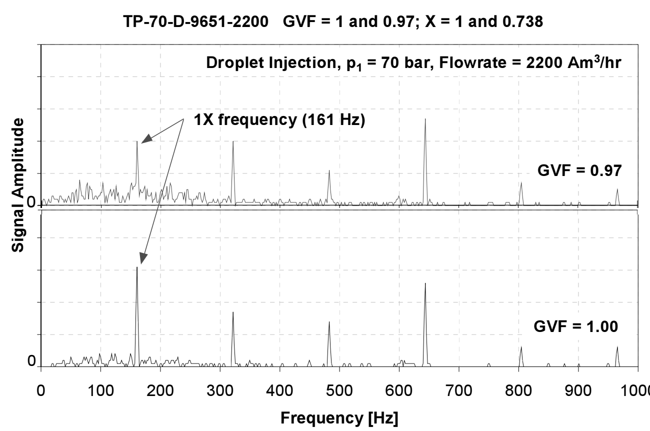

Figure 16. Effects of Two-Phase Flow over the Vibration Response of the Machine, Measured in the Horizontal Direction at the Driven End (p_1 = 70 bar, 9651 rpm, Droplet Injection Pattern).

Figure 17 presents the frequency spectra of the shaft horizontal vibration component, measured at the driven end of the machine, while it was operating at 9651 rpm, with a suction pressure of 70 bar (1015.3 psi) and handling a volumetric flow of 2200 Am^3/hr (77,692.27 ft^3/hr). The bottom spectrum corresponds to operation with a single-phase (dry) gas, while the top spectrum represents the behavior of the machine while handling a two-phase gas mixture, with a gas-volume fraction of 0.98 (gas quality of 0.810). For this case, the condensate was being injected into the gas with a liquid film pattern, which was uniformly distributed around the wetted surface of the inlet pipe. As may be seen in the figure, the vibration spectra for the dry and the wet gas cases are also very similar. This supports the belief that when the liquids are ingested in a uniform manner by the compressor, they do not provide a sufficiently strong source of excitation or unbalance to disturb the rotordynamic behavior of the machine. The above similarity was also encountered when comparing the vibration spectra corresponding to GVF values of 0.97 and 0.99.

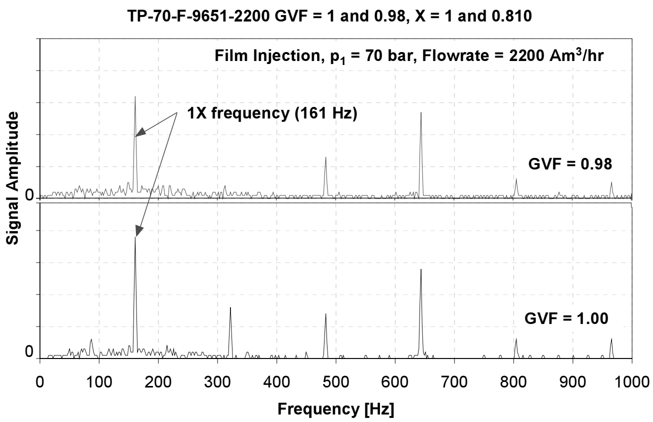

Figure 17. Effects of Two-Phase Flow over the Vibration Response of the Machine, Measured in the Horizontal Direction at the Driven End (p_1 = 70 bar, 9651 rpm, Film Injection Pattern).

Figure 18 presents the frequency spectra of the horizontal vibration component, measured at the driven end of the machine, while it was operating at 9651 rpm, with a suction pressure of 30 bar (435.1 psi) and handling a volumetric flow of 2200 Am^3/hr (77,692.27 ft^3/hr). The bottom spectrum corresponds to dry gas operation while the top spectrum represents the machine behavior while handling a two-phase gas mixture, with liquids injected with a uniform droplet pattern. As may be seen in the figure, the vibration spectra for both the dry gas and the wet gas compression show similar trends at frequencies above the machine running speed. However, the spectrum corresponding to the wet gas compression shows some peaks in the subsynchronous range. The appearance of a peak at one half the running speed suggests the presence of some type of rotor instability. The compressor manufacturer is currently conducting an investigation to determine the source of this instability. Note, however, that this behavior was observed when the machine was operating with a gas-volume fraction of 0.97, which at the suction pressure of 30 bar (435.1 psi) represents a gas quality of 0.530. In addition, it is important to state that the vibration amplitude at the running speed and its harmonics did not increase when the subsynchronous component appeared. Furthermore, this subsynchronous component disappeared when the gas-volume fraction was increased above 0.98 (quality was increased above 0.62).

The figures presented above provide a sample of the machine rotordynamic behavior that was observed during the tests. It is important to note that the behavior characteristics presented for each combination of suction pressure, machine speed, and liquid injection pattern were exhibited by the machine throughout the whole range of volume flows that were evaluated during each test period. This information is not included in this paper as it would be repetitive and would produce an excessively long document.

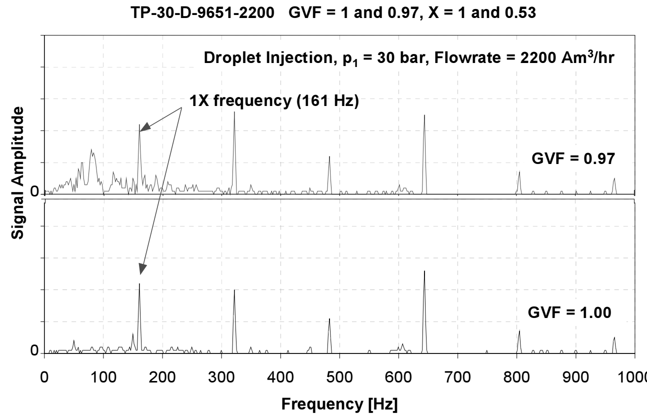

Figure 18. Effects of Two-Phase Flow over the Vibration Response of the Machine, Measured in the Horizontal Direction at the Driven End (p_1 = 30 bar, 9651 rpm, Uniform Droplet Injection Pattern).

SUMMARY AND CONCLUSIONS

This paper presented the results of tests performed on a centrifugal compressor operating under wet gas conditions, showing the effects of suction pressure, gas-volume fraction, machine speed, and liquid injection pattern over the thermodynamic and mechanical performance of the machine.

The application of centrifugal compressor technology is a viable option for single-stage, two-phase compression at gas-volume fractions at or above 0.97, corresponding to gas qualities as low as 0.522 for suction pressures of 30 bar (435.1 psi) and 0.738 for suction pressures of 70 bar (1015.3 psi). The exact level of gas-volume fraction will of course depend on the values of suction pressure and pressure ratio, as well as the distribution of the liquid phase within the gas when it enters the machine. For future applications, the relative densities and phase properties of the gas and liquids that are handled will need to be considered.

The thermodynamic evaluation of the machine showed that relative to a dry gas compressor, the compressor pressure ratio increased when the gas-volume fraction was decreased within the values that were tested (GVF between 1.0 and 0.97). The increase in pressure ratio was attributed to the larger density of the fluid that was being handled by the compressor when liquids were injected. In addition, the compressor temperature ratio showed a slight decrease when liquids were injected. This was probably caused by a transfer of energy from the gas to the liquid (heating of the liquid), and a limited condensate phase transition through the compressor.

The specific compressor power consumption was also reduced as the liquid fraction was increased. Nevertheless, when compared to separating the liquid and vapor phases and boosting them as separate streams, the specific power consumption for wet gas compression was larger.

For the data presented herein, the polytropic head for two-phase compression can be represented by a nondimensional head coefficient provided that the proper two-phase terms are included in the calculations. The two-phase polytropic efficiency of the machine decreased as the gas-volume fraction was reduced. This effect was more pronounced for the tests executed at the lower suction pressure.

No evidence of liquid erosion was detected by visual inspection of the machine internals after the test. It was noticed that the internals of the machine were cleaned by the liquid that had been ingested.

A repeatable reduction in the noise level of the machine was detected when the compressor was handling the wet gas mixture. The dynamic pressure transducer data showed that the pressure fluctuations within the flow were attenuated by the presence of liquid in the gas-stream.

In general, the test results showed that the vibration of the machine was not significantly affected by liquid hydrocarbon ingestion for both the uniform droplet as well as the fluid film injection pattern. This may not be the case if the liquids are not uniformly distributed. Also, for the case in which the quality of the gas was below 0.62, the appearance of a subsynchronous vibration suggested that liquids could have been entrained in the seal areas at the impeller eye and balance piston, causing some type of rotor instability.

The effects of liquid phase change that may occur inside the machine should be further investigated prior to embarking in the design of a multistage centrifugal compressor for wet gas applications. Recall that the phase change inside the compressor may cause a mismatch between the stages, leading to performance shortfalls. Furthermore, the effects of liquid ingestion over the machine internal loss mechanisms should be investigated.

DISCLAIMER

The information contained in this paper consists of factual data and technical interpretations and opinions, which, while believed to be accurate, are offered solely for informational purposes. No representation or warranty is made concerning the accuracy of such data, interpretations, and opinions.

NOMENCLATURE

Parameters

D = Impeller exit diameter
GVF = Gas-volume fraction
h = Enthalpy
$\dot{m}$ = Mass flow
MW = Molecular weight
n = Polytropic volume exponent
N = Machine rotational speed
p = Pressure (absolute)
P = Power
$\dot{Q}$ = Actual volumetric flow
R_o = Universal gas constant
T = Temperature
U = Tangential velocity
U_2 = Impeller tip speed
ν = Specific volume
W_p = Polytropic head
x = Gas quality
Z = Compressibility factor
μ_p = Polytropic head coefficient
η_p = Polytropic efficiency
ϕ = Flow coefficient

Subscripts

1 = Machine inlet
2 = Machine discharge
l = Liquid
g = Gas
p = Polytropic
Tot = Total
TP = Two-phase

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