UNDERSTANDING WET GAS IN A SUPERCRITICAL CARBON DIOXIDE CYCLE

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Biographies

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Abstract

Supercritical carbon dioxide (sCO_2) power cycles are most efficient when the compressor inlet in the cycle runs at the carbon dioxide (CO_2) critical point. Operation at the critical point presents many design challenges including the potential to shift in the two-phase (gas/liquid) region during operation and the dynamic changes in thermophysical properties. This paper explores the use of a wet gas compressor, which allows operation at the critical point without the concern of running in the two-phase region. Wet gas compressors are being developed for subsea compression applications and have potential for use in a sCO_2 power cycle. There are many design challenges that exist for wet gas compressors. One of these challenges is gaining an understanding of how the liquid interacts with the aerodynamic surfaces in the compressor. Flow visualization work is being conducted to better characterize the two-phase flow for subsea compression applications. This paper explores the differences in the two-phase fluids in both the subsea compression application and sCO_2 cycle and discusses what is needed in terms of flow visualization to characterize the two-phase flow in a sCO_2 compressor.

Introduction

Cycle efficiency is one of the critical parameters linked to the successful operation of a Concentrating Solar Power (CSP) plant application. This is true for a CSP plant operating on a Supercritical Carbon Dioxide (sCO_2) power cycle. Ambient conditions often change rapidly during operation, making it imperative that the efficiency of the plant cycle be optimized to obtain the maximum power production when sunlight is available. For a sCO_2 power cycle past analyses have shown that operating the cycle at the critical point provides the optimum efficiency for dry operation. However, operation at this point is challenging due to the dramatic changes in thermophysical properties of CO_2 near the critical point and the risk of the fluid having a two-phase, gas-liquid state. As a result, there is a high likelihood that liquid can form upstream of the primary compressor in the sCO_2 power cycle.

This paper looks at the possibility of using a wet gas compressor for a sCO_2 power cycle to allow operation in the dry and the two-phase regions. The existing design challenges with wet gas compressors are discussed for the subsea application. The sCO_2 cycle is compared to subsea compression application in terms of Liquid Volume Fraction (LVF) and Liquid Mass Fraction (LMF). The level and importance of LVF and LMF in the different applications is explored. Lastly, the paper looks at what work is being done in the area of flow visualization of wet gas flow in a compressor. This includes looking at existing research and discussing what is needed for a sCO_2 wet gas compressor.

Two Phase Operation in sCO_2 Cycle

In a sCO_2 cycle, the conditions at the compressor inlet are near the critical point. Figure 1 shows the critical point for CO_2 on its vapor dome. Because the properties of CO_2 vary quickly near this point, its compressibility, and therefore the work required to pressurize the gas, are low. The sCO_2 cycle uses this reduced compressive work requirement to its advantage by operating near the critical point, and thus increasing the overall efficiency.

Operating near the critical point increases the risk of slipping into two-phase (gas and liquid) flow due to slight variations in temperature or pressure. To avoid this, designs subject to variations in operating point conditions (such as a CSP plant) typically operate away from the critical point, as shown in Figure 2. The operating point shown in this figure is at 76.9 bara and 40 °C whereas the critical point is 73.8 bara and 31 °C. Figure 2 clearly shows that avoiding two-phase flow comes at a high cost to efficiency.

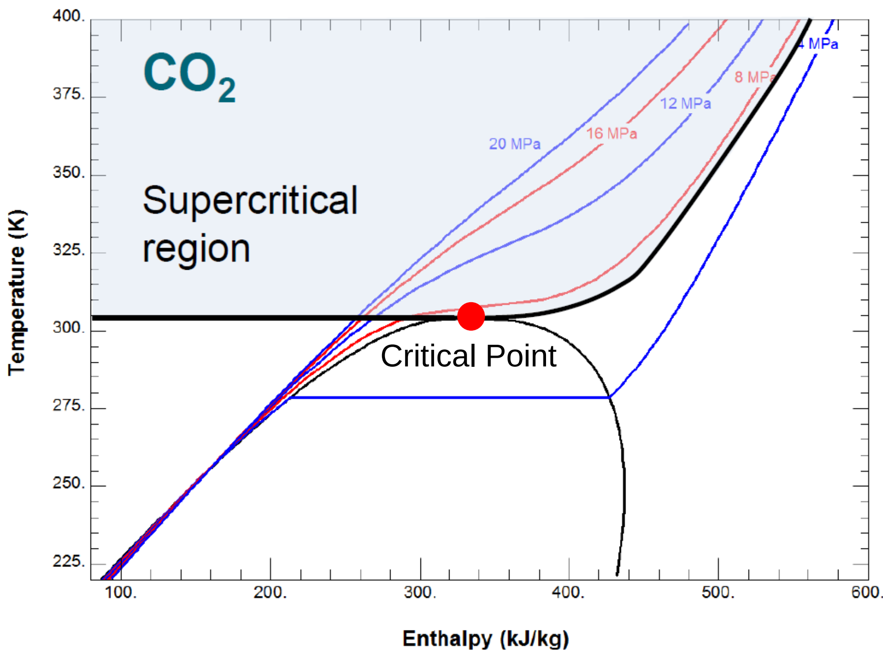

Figure 1. **Supercritical region and critical point in CO_2 phase diagram [1]**

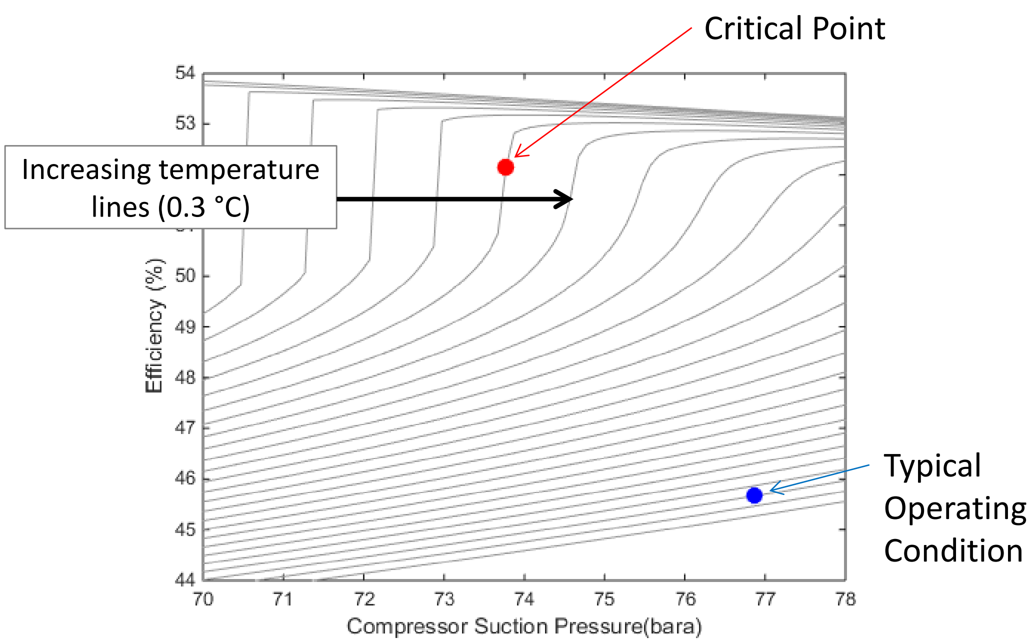

Figure 2. **Efficiency of sCO_2 cycle with operation at and near the critical point**

Because operation near the critical point is ideal, methods of mitigating the effects of liquid that could form in the flow stream are highly desired. A potential solution to the presence of two-phase flow is the use of a wet gas compressor.

Wet Gas Compressor to Improve Cycle Efficiency

A wet gas compressor is a type of compressor that is capable of handling some liquid in its flow (40% to 60% LMF). This type of compressor would allow operation near the critical point, as it could handle intermittent two-phase flow as cycle operating points change. This technology is relatively new (first implemented in industry in 2015), and progress is being made to address its various design challenges.

While no data is available on wet gas compression of sCO_2, it is believed that compressor performance may follow similar trends to the data available for air/water or gas/liquid hydrocarbon mixture flows in some regards. In general, wet gas compressor efficiency drops significantly during two-phase flow, while power required increases.

An analysis was performed to investigate whether the gains of operating near the critical point outweigh the potential costs of intermittent losses in compressor efficiency for a full sCO_2 cycle. The analysis used yearly weather data for Las Vegas, NV to predict, conservatively, when "wet" flow conditions might be expected at the compressor inlet in a CSP application operating near the critical point. Typical component efficiencies and standard operating conditions were assumed for a dry cycle (operating away from the critical point, as typically done). Two wet cycles were simulated, both operating at the CO_2 critical point. One cycle used the current state-of-the-art wet gas compressors (no gas ejection). The efficiency of this compressor decreases from 80% in dry operation to 40% when liquid is present. The second compressor is assumed to have technology improvements (gas ejection) that increase the wet operation efficiency to 60%. Table 1 summarizes these assumptions.

Table 1. Summary of parameters for cycle efficiency analysis

Parameter	Value
Turbine power output (MW)	20
Turbine efficiency (%)	85
Compressor efficiency – dry (%)	80
Compressor efficiency – wet, no gas ejection (%)	40
Compressor efficiency – wet, gas ejection (%)	60
Regenerator efficiency (%)	85
Mass flow rate (kg/s)	100
Turbine inlet temperature (°C)	720
Compressor inlet – at critical point	
Pressure (bara)	73.8
Temperature (°C)	31
Compressor inlet – conventional operating point	
Pressure (bara)	76.9
Temperature (°C)	31
Compressor discharge pressure (bara)	250

The wet cycles outperformed the dry cycle year round in both power consumed and efficiency. Figure 3 shows the calculated cycle efficiency over time for the three cycles. The wet cycle gains were lessened during the winter months, but still showed significant improvement. Yearly average efficiency improvement using a wet gas compressor was 5 to 6%.

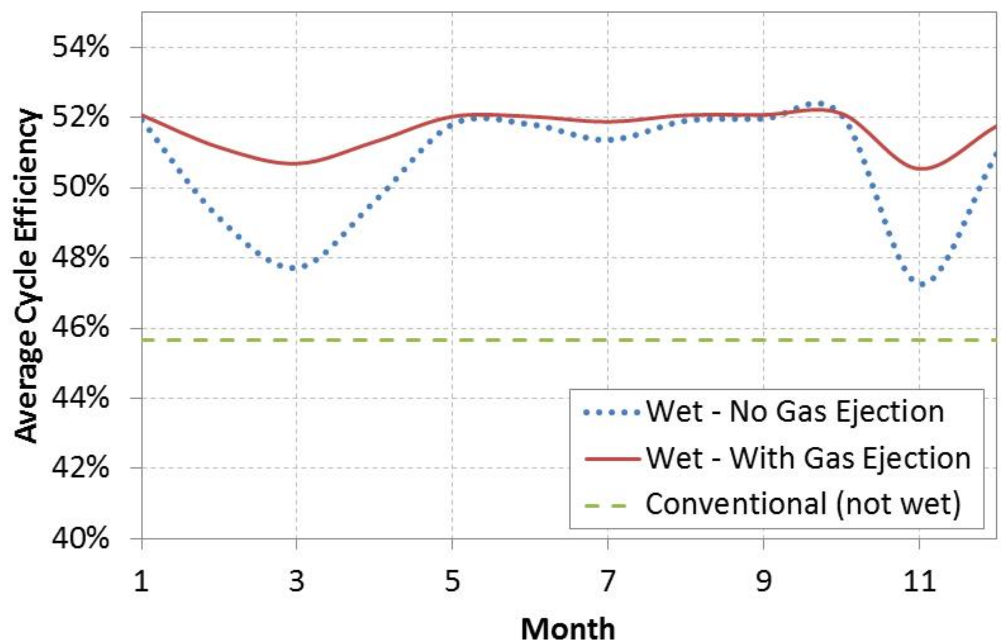

Figure 3. Average sCO_2 power cycle efficiency for one year (Month 1 is January)

While wet gas compressors show promise in improving sCO_2 cycle efficiency, many areas of the technology need improvement before they are ready for implementation in such cycles. These challenges are summarized in Table 2.

Table 2. Summary of challenges in the design of a wet gas compressor

Component	Design Challenge
Aerodynamic Design	• Reduced efficiency and increased power load when operating with liquid • Erosion of aerodynamic surfaces
Mechanical Design	• Change of spring and damping coefficients due to presence of liquid. • High sub-synchronous radial and axial vibrations due to liquid in the compressor. • Increased thrust load due to liquid
Shaft Sealing	• Seal gas and liquid against high pressure • Protect sealing elements from damage due to liquid • Prevent dry ice formation in seals

Wet Gas Compressor Technology Readiness

The performance impact of liquid in wet gas compressors is one of the chief areas needing research. Figure 4 shows the variation of a wet gas compressor's performance with increasing LVF for an air/water flow. More liquid in the flow restricts operation at higher flow rates, and actually pushes the surge limit to the left. However, surge occurs more suddenly and with less warning than in dry applications. Increasing the LVF of the flow stream has been shown to significantly influence the aerodynamic performance of the compressor. As LVF increases, fluid build-up on the aerodynamic surfaces, such as compressor blades, increases. This can alter the effective blade profile.

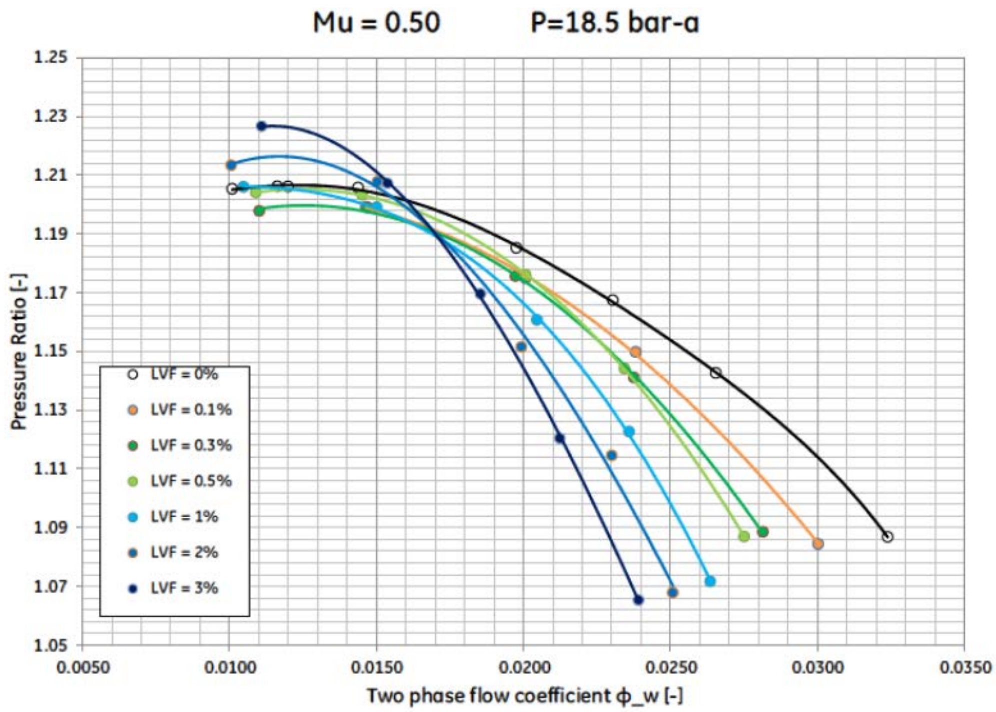

Figure 4. **Compressor performance map with dry and wet operation [2]**

The introduction of liquid also affects the mechanical performance of the wet gas compressor. Sub-synchronous vibration is generally increased, but most of the time does not restrict operation of the compressor. Ransom et al. showed a low frequency axial vibration, as seen in Figure 5, which was within the clearance of the bearings, but in general, would not meet industrial standards. Ransom et al. also showed that the presence of liquid caused axial thrust to initially decrease, then steadily increase with increasing LVF.

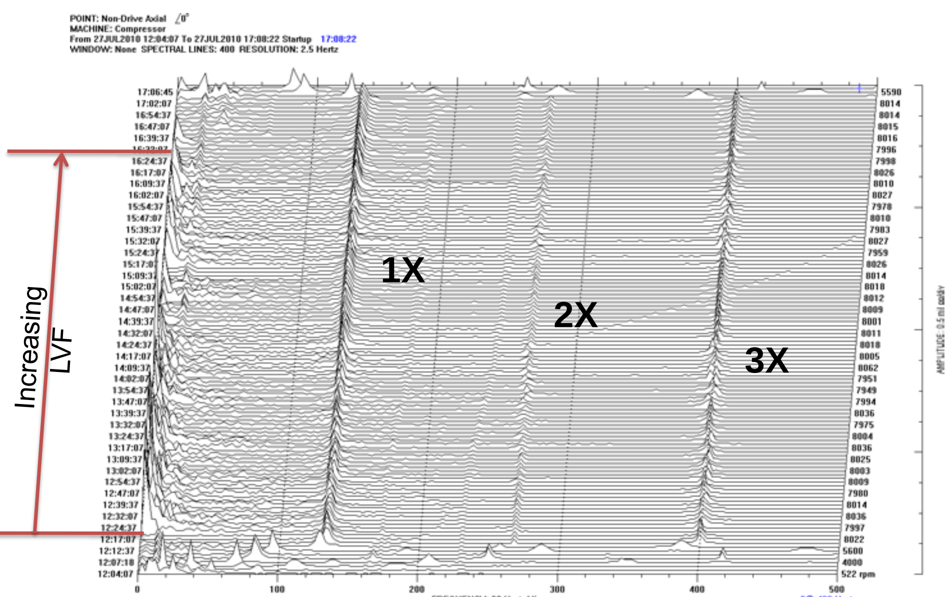

Figure 5. **Waterfall plot of axial vibrations from wet gas compressor testing of a two stage compressor [3]**

With wet gas compressors, special care must be taken to ensure that liquid does not enter the dry gas seals, as they have low tolerance for liquid and solid particles. Features such as gas buffers, labyrinth seals, slinger rings, and drains are recommended. In general, it seems that two-phase flows perform differently than dry flows when it comes to sealing. Continuous liquid leakage from labyrinth type seals has been observed even with gas buffer pressures acting against the leak direction. The cause of this phenomenon is not well understood, but further emphasizes the need for careful consideration of seal design in a wet gas compressor.

Unique Aspects of Wet sCO_2 Gas

Wet gas in a sCO_2 application is expected to be quite different than in the subsea compression applications in some regards. The compositional makeup of the gas and liquid streams differ significantly and the relationship between the LVF and LMF are different.

In the subsea compression application, the gas is comprised primarily of light to heavy hydrocarbons and the liquid includes hydrocarbon condensates and water. Table 3 lists a gas composition used by Brenne et. al [4] in a test campaign of a single stage compressor in wet gas conditions similar to a subsea environment. This gas composition only contains hydrocarbons and no water. The phase diagram for the gas composition is shown in Figure 6. A red dot is used to highlight the expected compressor suction condition of 70 bara and 35 °C. The calculated LMF and LVF for this location is 24.5% and 3.1% respectively. Note that the LVF is lower than the LMF.

Table 3. Gas composition from single stage compressor test for subsea compression [4]

Component	Composition (mol%) at 70 bara and 35 °C		
	Gas Phase	Liquid Phase	Total
N_2	0.854	0.089	0.531
CO_2	1.524	0.956	1.284
C1	90.933	25.920	63.474
C2	4.103	4.489	4.266
C3	0.341	0.955	0.600
I-C4	0.124	0.651	0.347
N-C4	0.654	4.632	2.334
I-C5	0.481	6.662	3.091
N-C5	0.454	7.856	3.581
C6	0.348	13.897	6.071
C7	0.137	12.932	5.541
C8	0.035	6.637	2.824
C9	0.008	3.084	1.307
C10+	0.005	11.239	4.750
Molecular Weight	18.483	73.52	41.728

In a sCO_2 system, the gas and liquid are assumed to be pure CO_2. The sCO_2 system starting point (or compressor suction condition) is near the critical point. The critical point pressure and temperature are 73.8 bara and 31 °C, respectively. In order for wet gas to occur, the pressure must decrease and the temperature may change. An operating point of 73.7 bara, 30.9 °C, with an LMF of 24.5% was used to evaluate the relationship between the LVF and LMF for CO_2. Figure 7 shows the phase diagram for CO_2 with the wet operating point. Notice that the point is very near the critical point. The LMF and LVF at this location are 24.5% and 21.2% respectively. The LVF and LMF in this case are nearly equal.

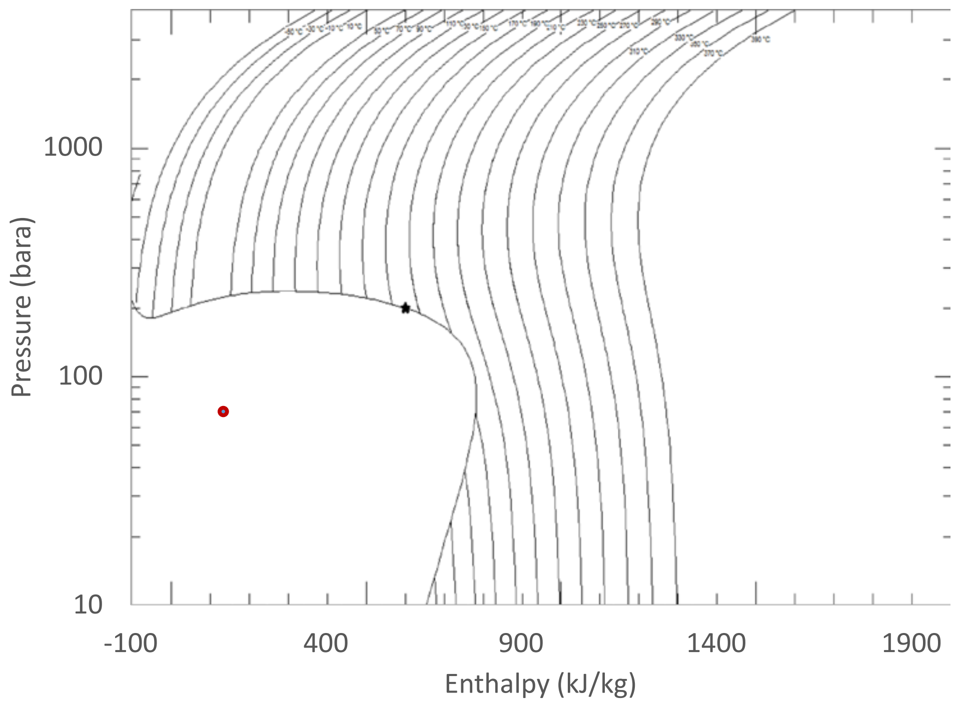

Figure 6. Phase diagram for Brenne [4] composition (subsea compression) showing typical compressor suction condition (red dot, 70 bara, 35 °C). The LMF is 24.5% and the LVF is 3.1%.

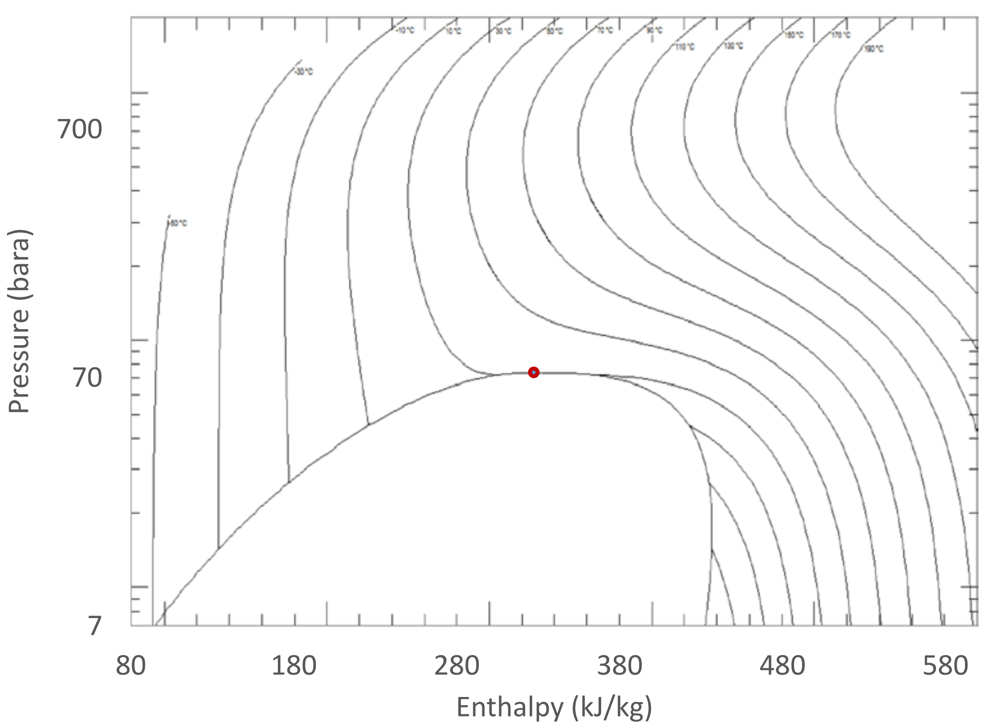

Figure 7. Phase diagram for pure CO_2 showing possible sCO_2 wet gas compressor suction condition (red dot, 73.70 bara, 30.9 °C). The LMF is 24.5% and the LVF is 21.2%.

Figure 8 shows the relationship between the LVF and LMF for a large collection of data on oil and gas subsea compression systems and curves representing the locations where a sCO_2 system might operate with wet gas. The sCO_2 curves were generated at conditions very near the critical point. Notice that the oil and gas data tend to have high LMF and low LVF and the sCO_2 tends to have nearly equal LVF and LMF. This Figure and the two above all clearly show that there is a distinct difference in the relationship between LVF and LMF for subsea compression systems and for a sCO_2 system.

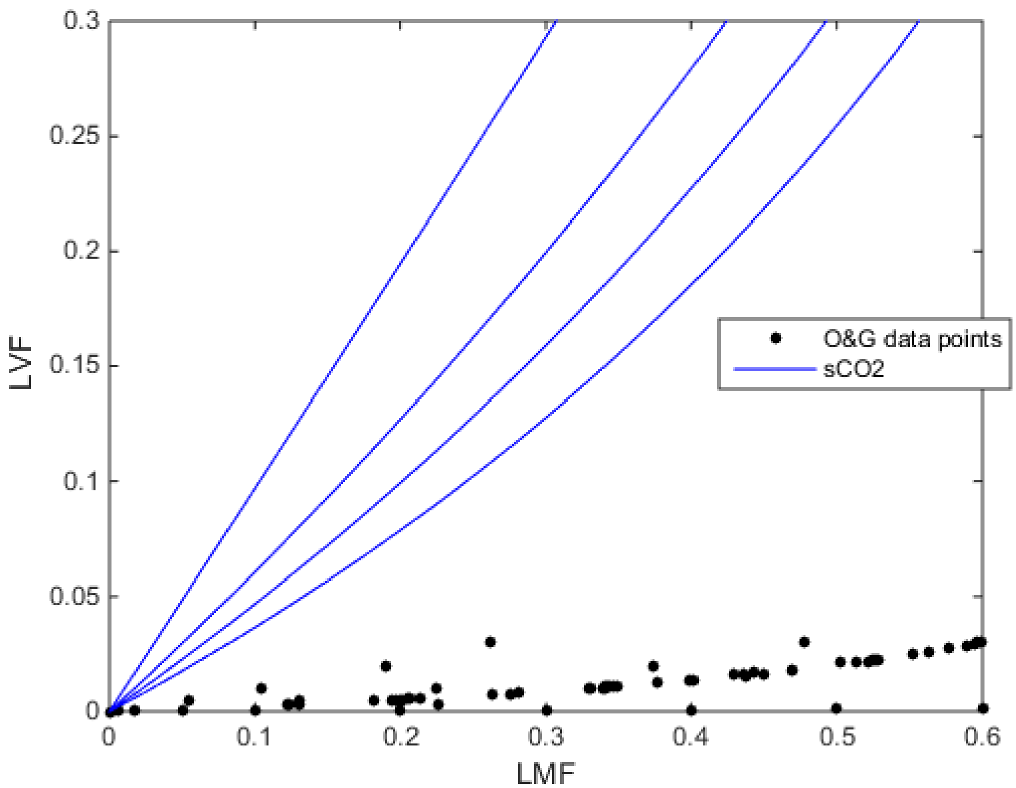

Figure 8. Comparison of LVF and LMF for oil and gas applications and the sCO_2 power cycle

Impact of LVF and LMF on Compressor Performance

The next question that needs to be answered, is which value is important; the LMF, LVF, or both? And does the difference seen in the two data sets make a difference? These questions are answered by reviewing the results of a regression analysis performed by Musgrove et. al. [5]. Musgrove et. al. analyzed data from a wet gas testing campaign of a two stage compressor that was conducted with air and water. Air and water are different than hydrocarbon based fluids, but the LVF and LMF relationships with the air and water closely represented that of the hydrocarbon based fluid. In these tests, the LMF values tended to be higher (40 to 60%) and the LVF values tended to be low (less than 3%).

The analysis was conducted using a multiple regression model to evaluate the impact of various operating parameters on the performance of a wet gas compressor. The operating parameters that were studied included the compressor speed, suction pressure, suction temperature, difference between the gas and liquid inlet temperatures, the gas volumetric flow rate, the liquid volumetric flow rate, the gas/liquid density ratio, LVF, and LMF. The output variables considered for the analysis were the compressor pressure ratio, compressor torque, axial thrust, and the isentropic efficiency.

Figure 9 shows the sensitivity analysis on the various input and output parameters. The figure demonstrates the progression of the analysis. On the left hand side is the first analysis set where all input parameters were considered and the right hand side shows the final selection with a reduced number of inputs. In this progression, the input parameters of the regression model were decreased to see if a good regression fit could be obtained with a minimal number of input parameters. Note that the on the right hand side the two parameters that remain are the LVF and LMF. This provides an initial indication that both of these parameters are important for compressor performance.

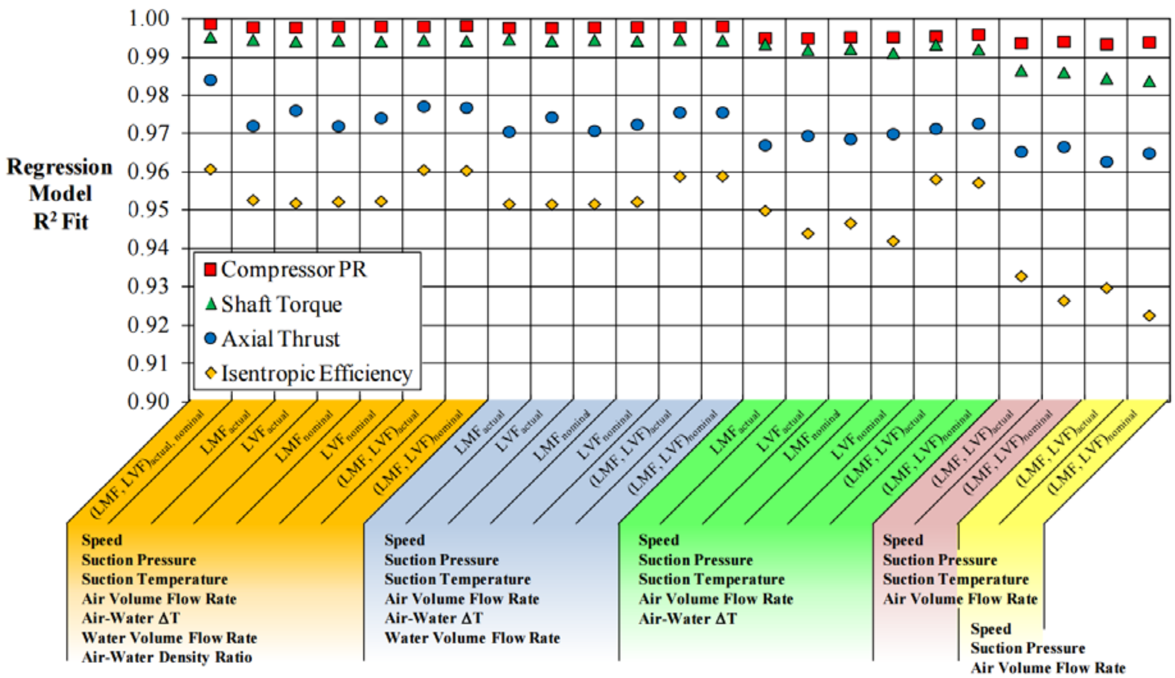

Figure 9. Sensitivity of 2nd order regression to selected input variables [5].

The last step in the analysis was to compare the regression coefficients for each set of parameters to evaluate the relative impact of each parameter. Figure 10 shows the comparison of regression coefficients for the second order model. A review of this data shows that the speed, air volumetric flow rate, LMF and LVF are the dominant parameters. Speed and airflow rate were also found to be dominant in an analysis or dry performance data (no liquid). Therefore, the two parameters that have an impact with wet flow are the LMF and LVF. Both of these parameters contribute to the compressor performance. The LMF influences all the compressor performance metrics and the LVF has a significant influence on the compressor pressure ratio and the isentropic efficiency.

The review of the regression analysis has shown that both LMF and LVF are important for wet gas compression. This means that the difference in the LMF and LVF relationship between the subsea compression and sCO_2 systems is relevant. The trends observed in the subsea compression data can provide some insight into the how a sCO_2 wet gas compressor might perform, but additional testing and research specific to the sCO_2 system is required.

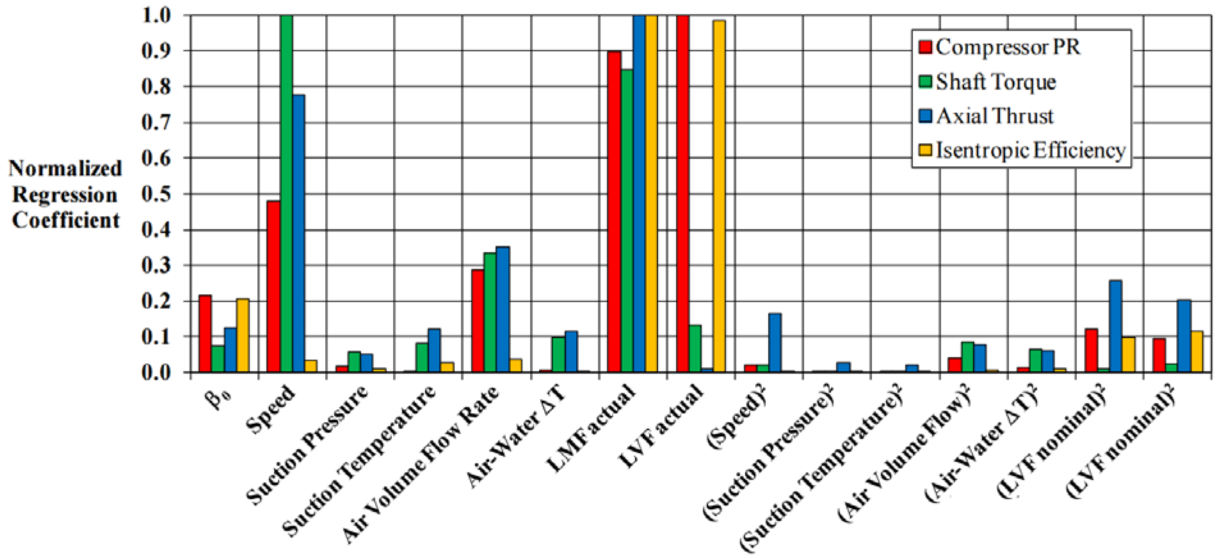

Figure 10. *Comparison of regression coefficients for second order model for the wet gas data [5]*

Flow Visualization of Wet Gas Flow

The fundamental reasons of why the liquid alters the compressor performance have been studied by various organizations. The primary focus has been on thermodynamic and Computational Fluid Dynamics (CFD) models. The thermodynamic models make a major assumption that the liquid is in the form of evenly disperse and spherical droplets that travel with the flow stream [4]. This type of approximation is suitable for certain applications, such as inlet fogging systems on gas turbines. However, it does not appear to hold true for the wet gas compression application in question. Figure 11 demonstrates this by showing the make-up of the shaft power from wet gas test data. The vertical axis is the shaft power and the horizontal axis shows the LVF in the compressor. The height of each bar is the total shaft power for that test condition. Note that the power consumption predicted by a thermodynamics two-phase model [4] and evaporation model is accurate for dry flow and wet flow with a small amount of liquid, but as the LVF increases, there is a large portion of power that is not accounted for by thermodynamics, evaporation, or mechanical losses.

In a wet gas flow stream, there is a mixture of liquid droplets, flow attached to the wall of the flow channel, and, possibly, large amounts of liquid traveling at the bottom of the flow channel. In addition, as the liquid flows through the compressor, it will attach to aerodynamic surfaces, which distorts the aerodynamic performance of the machine, and some liquid will evaporate due to the heating of the gas from the compression process. All of these influences require that multiple corrections be added to the thermodynamic models to obtain realistic results.

In an effort to improve upon the thermodynamic modeling, researchers have been attempting to use CFD to model the two-phase flow through the compressor. Both discrete-phase and multiphase models are being considered. The discrete-phase models are weakened by the fact that the fluid particles are modeled as spherical droplets and do not affect the gas flow field [6]. This means that the fluid interaction with blade surfaces is not captured. The multiphase modeling allows for the interaction of the liquid with the gas flow field and includes influences such as a liquid film building on the compressor blades, as seen in Figure 12 [7].

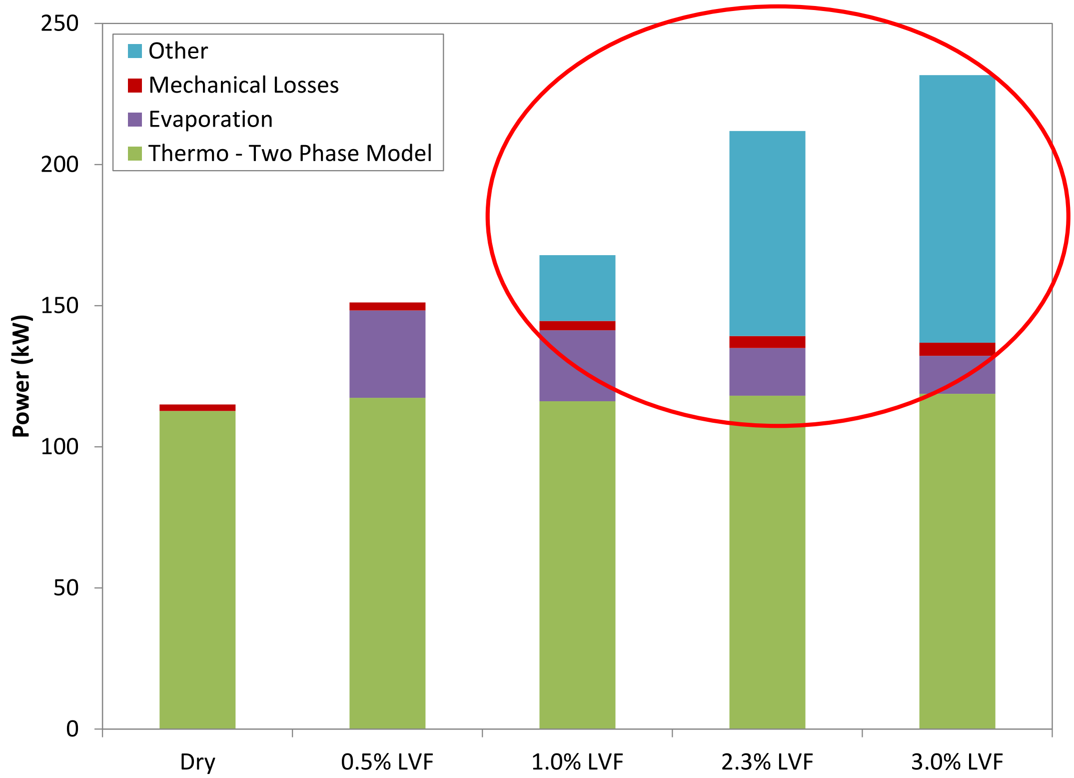

Figure 11. Components of shaft power from wet gas test data measured at on a two-stage compressor

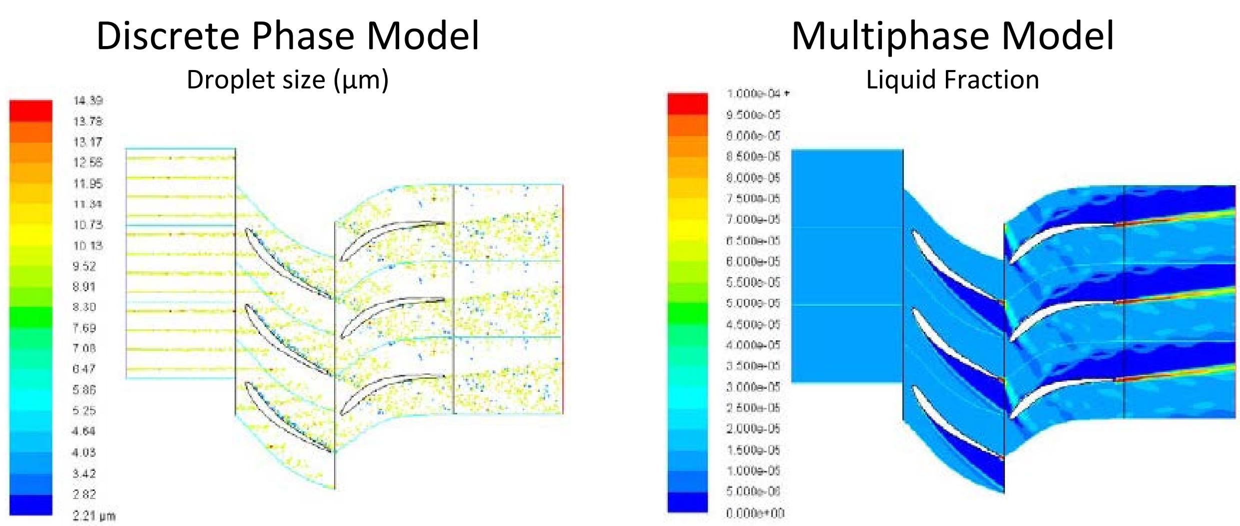

Figure 12. Comparison of Discrete Phase and Multiphase CFD Modeling for a Wet Gas Axial Compressor [7]

A large effort is being made to improve the modeling capabilities of multiphase flows in the compressor, but this work has only been validated with data captured in a wind tunnel on a stationary blade and inside an ambient pressure compressor. Figure 13 shows images from two-phase wind tunnel testing of a compressor blade. These images clearly capture the effects of the liquid attaching to the blade. However, they do not include the influences of the rotation of the blade, interaction between the flow from the rotor to stator, or a distorted flow regime at the inlet of the compressor. These effects will influence the flow regime and affect the performance of the compressor. In order to improve the performance predictions in wet gas compression beyond its current state, data that characterizes the flow regime inside the compressor is needed. For the sCO_2 system, this is more challenging. In order to replicate the relationship between LMF and LVF, the flow visualization must be conducted in a pressurized system that is near the critical point.

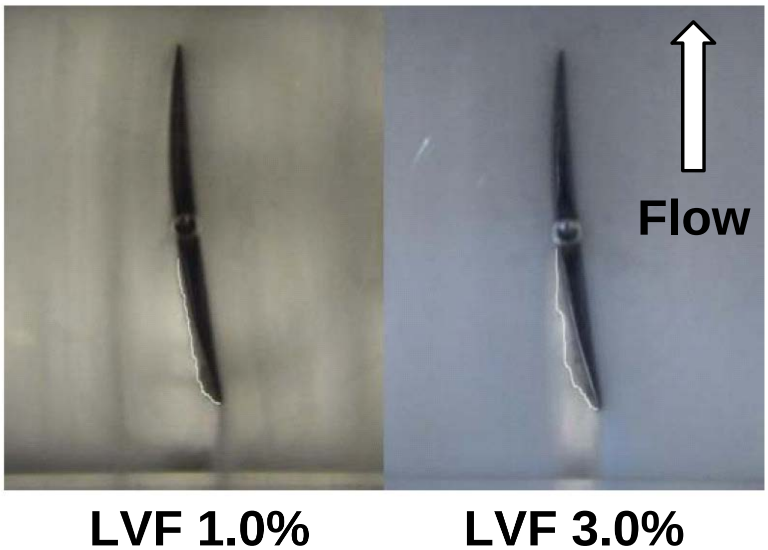

Figure 13. Visualization of Liquid attached to a Blade at Different LVF in a Wind Tunnel [8]

The following sections look at what flow visualization techniques have been used for wet gas compressors to date, what other visualization techniques are available for two phase systems, and discusses the technology gaps for visualizing flow in a sCO_2 system.

Flow Visualization Techniques for Two Phase Flow

A large portion of existing flow characterization technologies uses optical systems to capture the velocity, shape, and size of the fluid particles. Some of the available optical technologies are listed below with their applicability to visualization in a wet gas compressor.

- Surface flow visualization method: This method is not directly applicable to wet gas compression because it only provides images of the streamlines near a wall. In a compressor, the liquid will be attached to the wall and, also, in the free stream. Both locations need to be captured.
- Particle tracer methods: Methods such as Particle Image Velocimetry (PIV) could be useful for tracing the flow of liquid through the compressor. However, large liquid flow rates are often needed for testing. The amount of tracer particles needed could be cost prohibitive. In addition, in a sCO_2 test system, the liquid CO_2 may not be fully recycled (relates to control of evaporation and condensation). This will cause the tracer particles to deposit and be lost in the test system.
- Optical refractive index methods: Methods such as shadowgraph, Schlieren photography, and interferometry are ideal for the visualization of planer images. These types of techniques could be useful for flow at the inlet of the compressor or through the diffuser vanes on the exit, but would not provide the view needed inside the compressor.
- Light scattering methods: The methods that utilize light scattering could be applicable to the inlet or exit of the compressor, but would be difficult to use inside the compressor primarily because it will be challenging to have sufficient light inside the impeller. This is true for a shrouded impeller. If an unshrouded impeller was used, the light scattering methods could be

used at all locations. In most industrial applications, shrouded compressors are used for operation.

- Photographic methods: This method would be applicable to the inlet, inside the unshrouded compressor, and at the exit. It can provide the high-speed frequency needed for capturing images inside the compressor impeller. Also, this would be the most simplistic method. The primary challenge with this method is providing enough light inside the compressor for capturing the images and the camera accessing the internals of the compressor (for a shrouded machine).

One of the challenges in visualizing two-phase flow in a closed channel is keeping the optical windows clean. Liquid can quickly build up on these windows and distort the view. Active cleaning systems are necessary to ensure that the camera has an unobstructed view. Some of the techniques used to keep optical windows clean are:

1. Apply a gas purge onto the window to keep liquid off [9-11]
2. Heating windows to a temperature higher than the boiling point of the liquid [12]
3. Application of coating to window to reduce surface tension to prevent smearing of liquid [13]

The inlet and exit of the compressor could be viewed using optical techniques, but the inside of a full-scale shrouded compressor impeller is not accessible through viewing windows. Therefore, a non-optical method must be used for characterizing the flow inside the impeller. One of the primary areas that need to be quantified is the liquid film thickness on the compressor blade. This directly influences the aerodynamics of the compression process.

Electrical resistance probes have been used in the past to measure the film thickness on flat metallic surfaces. The electrical resistance probe is two electrodes placed side-by-side. A high-frequency and high-voltage signal is passed to the electrodes and the resistance between the electrodes is measured. Their use has been demonstrated in both stationary and rotating disk applications [14-15]. The primary challenge with use of these probes is carrying the electrical signal from the impeller blade to a stationary signal conditioner and data acquisition system. This has been done before for other measurements on a rotating component. A slip ring can be used to transfer the electrical signal from the rotating component to a stationary device.

Another challenge is the calibration of the probes in the test fluid. It is expected that films on the compressor surfaces would likely be very thin (approximately 10 to 100 mils). The liquid conductivity also impacts the sensor response. The sensors must be calibrated with the test fluid at various liquid film levels.

Flow Visualization Techniques Used for Wet Gas Compressors

To date, flow visualization work focused on wet gas compression has been performed by Southwest Research Institute (SwRI) and the Norwegian University of Science and Technology (NTNU). Both test programs are focused on gaining a better understanding of how the liquid interacts with the compressor surfaces and the influence this has on compressor performance. It should be noted that both test programs have been targeting LVF conditions seen in subsea compression applications where the maximum LVF is approximately 3%.

SwRI conducted an internal research program focused on the wet gas flow across an airfoil. This program studies how the liquid interacted with the airfoil and investigated methodologies to control how much liquid attached to the airfoil. The testing was conducted in a wind tunnel with air and water. Figure 14 shows one of the images obtained during this test program. The top image shows the water attached to the airfoil surface during normal flow. The second image shows the water being blown off the airfoil surface with the use of air flowing out holes in the surface of the airfoil (named gas ejection). The gas ejection scheme was found to be able to reduce the drag on the airfoil surface while maintaining the lift.

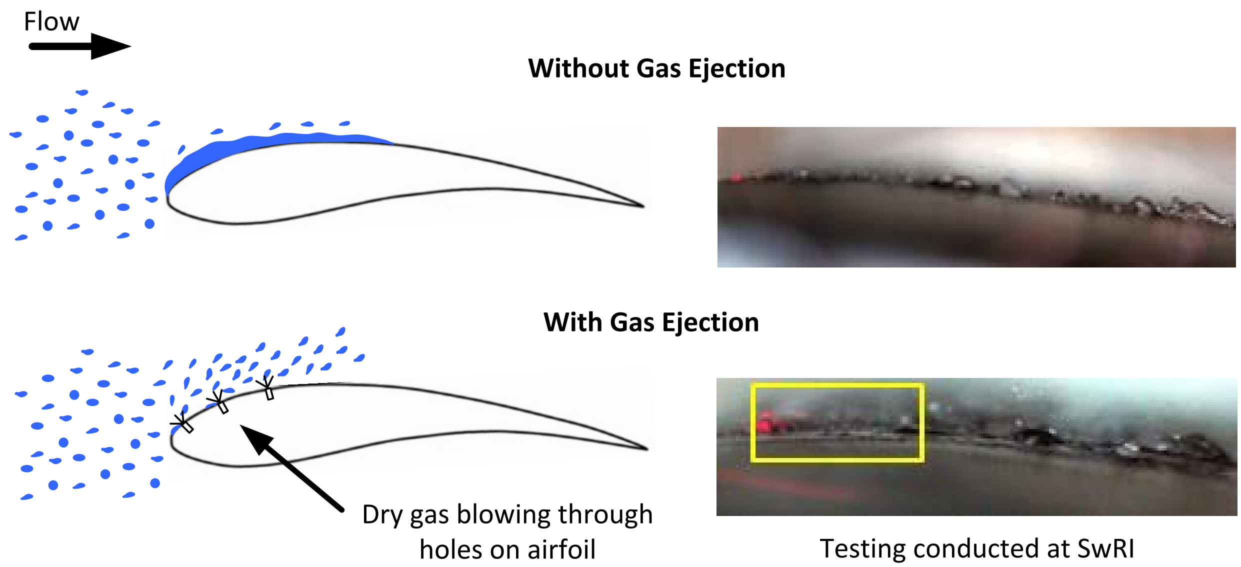

Figure 14. Water attachment and gas ejection on an airfoil in wet gas flow

Another ongoing test program at SwRI is focused on visualizing the flow inside the compressor. The intent in this program is to record images of the two-phase flow with air and water at various locations in the compressor including the compressor inlet, leading edge of the compressor blades, along the compressor blades, the trailing edge of the compressor blade, and in the diffuser section. This program is using various windows with a camera and electrical resistance sensors to characterize the flow through an unshrouded compressor. Some initial studies have been conducted on the windows to look at how to mitigate the attachment and smearing of water on the windows. As discussed above, liquid on the window can distort the view, which makes it difficult to capture clean images.

Figure 15 shows a comparison of visualizing flow through a plain glass window and a window coated with a hydrophobic coating. Note that the impeller at the top edge of the windows was rotating during the test. At first glance, it seems as if both windows perform terribly. With a closer inspection of the coated window, it can be found that there is much less water attached to the window. The majority of the water streaks seen in the image are attached to the aluminum plate that is behind the window. A close inspection of the impeller in the coated window shows that the edges of the impeller can be distinguished. Other schemes to keep the windows clean such as using a gas purge are currently being pursued.

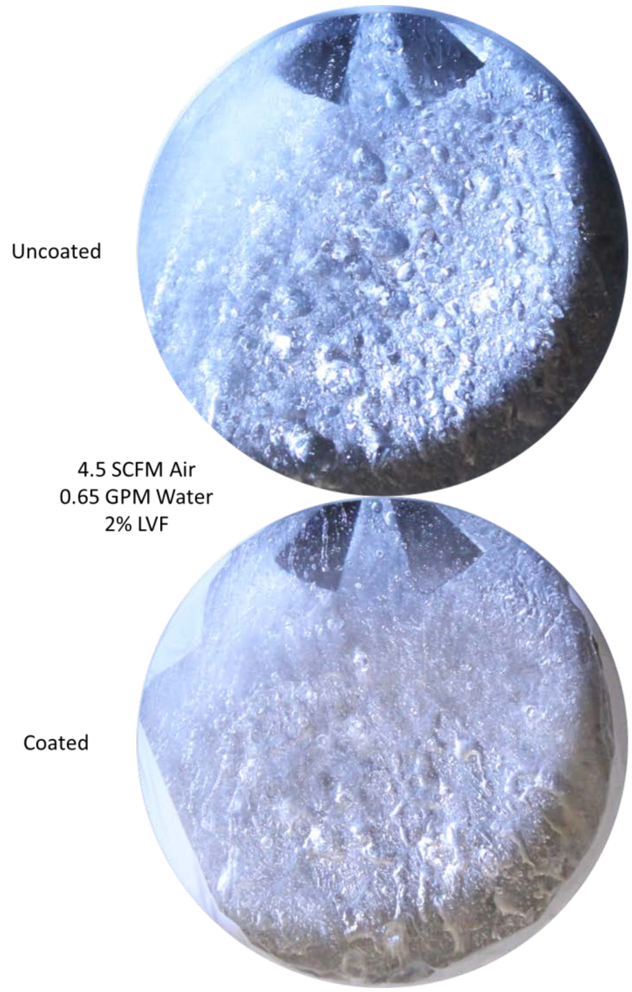

Figure 15. Comparison of coated and uncoated windows in SwRI window test

In addition to optical methods, the program at SwRI is developing sensors to measure the liquid film thickness on the compressor aerodynamic surfaces. Real machines will likely have shrouded impellers. This prevents optical methods from being used to gain an understanding of the two-phase flow inside compressor impeller. Electrical resistance film thickness sensors could be used to measure the liquid film thickness on the impeller blades. Figure 16 shows a film thickness sensor that is being tested. This sensor has two electrodes. An AC voltage signal at a specific frequency is passed through the sensor. The voltage across a load resistor is measured to determine the liquid film thickness. Currently the sensors are being calibrated for testing inside a compressor with air and water. The performance of the sensor has been found to be highly dependent on the electrical conductivity and temperature of the liquid.

NTNU has also been working on flow visualization inside a wet gas compressor. They conducted a flow visualization study on an airfoil in wet gas flow inside a small channel. The airfoil was subjected to a wet gas flow with LVF ranging from 0.2 to 3%. Figure 13 shown above is from this study. For this testing, a module was fabricated from Plexiglas that contained the airfoil and measurement section (see Figure 17) [8].

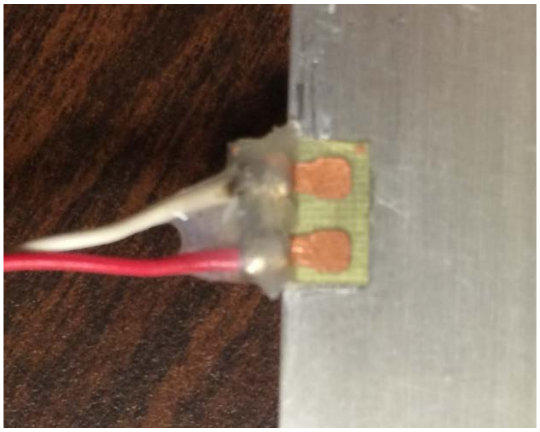

Figure 16. *Liquid film thickness sensor bonded to aluminum plate*

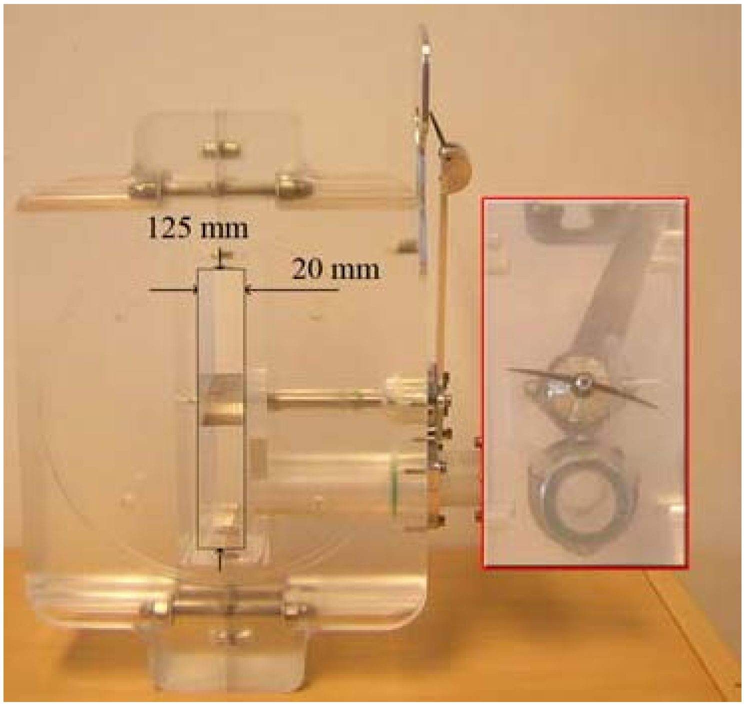

Figure 17. *Plexiglas module used in NTNU wet gas airfoil testing [8]*

NTNU has now moved to visualizing flow on a compressor. Figure 18 shows an image from this work. This image shows the compressor body and the location of various windows looking at the flow in the diffuser section of the compressor and at the inlet of the compressor. They also performed some testing looking at the flow around the inlet guide vanes [16-17].

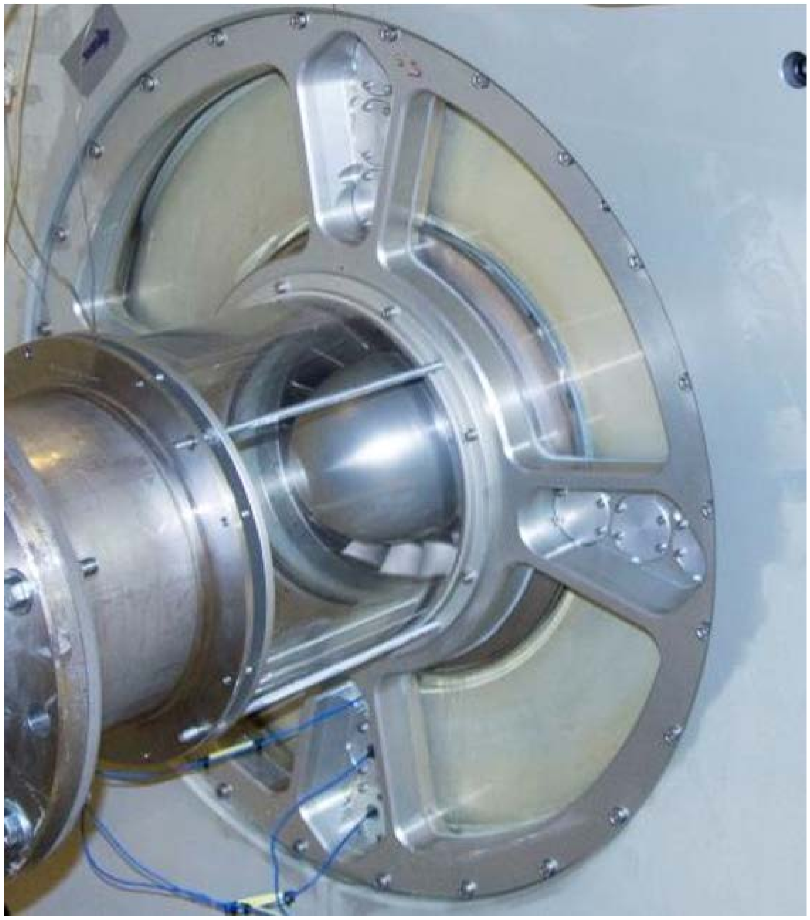

Figure 18. Window placement on test compressor at NTNU wet gas test facility [17]

Considerations for sCO_2

The flow visualization testing conducted to date has primarily been at or near ambient pressure. This is due to the pressure limitations of the windows or clear viewing sections. At these test conditions, the LVF of the subsea system process can be matched. The LMF tends to be very high when testing at ambient pressure (near 90%). Still as discussed before, the LVF in subsea applications tends to be low (3%) and the LMF tends to be higher (40 to 60%). Even with the variation in the relationship between LVF and LMF from the flow visualization tests to the subsea environment, the results of the testing provide useful information.

Flow visualization of the two phase fluid in a sCO_2 system would likely need to be at higher pressures. As discussed above, the LVF and LMF are approximately equal in sCO_2 systems. The windows or other flow visualization hardware will need to be rated at the planned operating pressure. The critical pressure of CO_2 is approximately 73.8 bara. In addition, the CO_2 test compressors are likely shrouded machines. This makes it difficult to access the internal compressor surfaces for visualizing the flow. Sensors such as the electrical resistance probes may be used measure blade film thicknesses.

Conclusions

As discussed above, a wet gas compressor can be used to operate at the critical point in a sCO_2 cycle. Wet gas compression is being developed for subsea compression applications and has potential for use in a sCO_2 system. There are several design challenges that are still being addressed in wet gas compression include: understanding the influence of the liquid on compressor aerodynamic and mechanical performance and finding appropriate methods to seal the shaft ends with a two phase flow. These are pertinent for both subsea compression and sCO_2 applications.

It was shown above that the relationship between LVF and LMF in the subsea compression and sCO_2 applications is very different. The LMF and LVF both influence how the compressor performs. It is important to understand how the liquid interacts with the compressor internal surfaces for predicting compressor performance. Some flow visualization work has been done in low-pressure compressor with air and water mixtures to gain better understanding in this area, but there is still more work to do to fully characterize the two-phase flow. Flow visualization for a sCO_2 system will be different from that required for a subsea compression system. It requires operating near actual pressures to get the correct LVF and LMF relationship. This presents many design challenges for future development of sCO_2 wet gas compressors.

References

1. Musgrove, G., et. al., "Tutorial – Heat Exchangers for Supercritical CO_2 Power Cycle Applications," Presented at ASME TurboExpo, June 2014.
2. Bertoneri, M. et al, "Development of Test Stand for Measuring Aerodynamic, Erosion, and Rotordynamic Performance of a Centrifugal Compressor under Wet Gas Conditions," Proceedings of ASME TurboExpo , GT2014-25349, June 2014.
3. Ransom, D., et. al., "Mechanical Performance of a Two Stage Centrifugal Compressor under Wet Gas Conditions," Proceedings of Turbomachinery Symposium, September 2011.
4. Brenne, L., et. al., "Performance Evaluation of a Centrifugal Compressor Operating Under Wet Gas Conditions," Proceedings of Turbomachinery Symposium, September 2005.
5. Musgrove, G., et. al., "Overview of Important Considerations in Wet Gas Compression Testing and Analysis," Proceedings of Turbomachinery Symposium, September 2015.
6. Luo, M., Zheng, Q., Sun, L., Deng, Q., Yang, J., "Numerical Simulation of an Eight-Stage Axial Subsonic Compressor with Wet Compression," Proceedings of ASME TurboExpo, 2013, GT2013-94486, June 2013.
7. Khan, J.R., "Comparison between Discrete Phase Model and Multiphase Model for Wet Compression," Proceedings of ASME TurboExpo, GT2013-96022, June 2013.
8. Grüner, T.G., Bakken, L.E., Brenne, L., Bjorge, T., "An Experimental Investigation of Airfoil Performance in Wet Gas Flow," Proceedings of ASME TurboExpo, GT2008-50483, June 2008.
9. Gomes, P., "Validation and Implementation of Optical Diagnostics for Particle Sizing in Rocket Motors," Naval Postgraduate School Thesis, December 1993, Monterey, California.
10. Bates, S., et.al, "Gas Cooled Probe Protectors," Thoughtventions Unlimited, May 2014.
11. Poppiti, J., "Practical Techniques for Laboratory Analysis," CRC Press, Inc., 1994.
12. "Gas Flow Measurements in Hazardous Environments," AZO Materials, October 19, 2013.
13. Toth, B., et. al., "Two-phase flow PIV experiments in a solid rocket motor model," Proceedings of the 14th International Symposium on Applications of Laser Techniques to Fluid Mechanics, July 2008, Lisbon, Portugal.

14. Hastings, E.C., et. al., “Preliminary Indications of Water Film Distribution and Thickness on an Airfoil in a Water Spray,” NASA Technical Memorandum 85796, July 1984.

15. Miyasaka, Y., “On the Flow of Viscous Free Boundary Jet on a Rotating Disk,” Bulletin of the JSME, Vol. 17, No. 113, November 1974.

16. Vigdal, L., et. al., “Inlet Guide Vane Performance at Dry and Wet Gas Conditions,” Proceedings of TurboExpo, GT2015-43225, June 2015.

17. Ferrara, V., et. al., “Wet Gas Compressor Surge Stability,” Proceedings of TurboExpo, GT2015-42650, June 2015.