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Centrifugal Compressor Rotordynamics in Wet Gas Conditions

Giuseppe Vannini
Principal Engineer
GE Oil & Gas - Nuovo Pignone
Florence, Italy

Matteo Bertoneri
Conceptual Design Engineer
GE Oil & Gas - Nuovo Pignone
Florence, Italy

Giuseppe Del Vescovo
Senior Test Engineer
GE Oil & Gas - Nuovo Pignone
Florence, Italy

Melissa Wilcox, P.E.
Senior Research Engineer
Southwest Research Institute
San Antonio, TX, USA

Giuseppe Vannini *is a Principal Engineer in the Advanced Technology Organization of GE Oil&Gas, which he joined in early 2001. He has been involved in advanced rotordynamics studies on high performance centrifugal compressors developing both analytical and experimental research activities. After leading the first subsea compressor prototype design up to the final FAT he came back to full-time rotordynamic activity and he's active especially in the field of annular seals modeling and testing, advanced gas and oil bearings validation. He holds a PhD in Mechanical Engineering at Pisa University and he's member of API684 Task Force.*

Matteo Bertoneri *is a Conceptual Design Engineer within GE Oil & Gas, in Florence, Italy. His current duties involve research, development and prototype-testing activities on mechanical and thermodynamics conceptual design of new Centrifugal Compressors. His previous tasks were in Requisition office, designing compressors for LNG and refinery jobs, and in Rotordynamic team, working on high speed rotor balancing. He has also worked as Test Engineer, responsible of Mechanical and Performance Tests of Centrifugal Compressors, Turboexpanders and Steam Turbines. Mr. Bertoneri received a B.S. and M.S. degree (Mechanical Engineering, 2010) from University of Pisa.*

Giuseppe Del Vescovo *is a Senior Engineer within GE Oil & Gas, in Florence, Italy. He joined GE in 2008 and during this period he has worked as Test Engineer on Validation& Testing activities concerning mainly R&D and new product and component introduction in the Advanced Technology Organization. Before GE he has worked for 7 years in Academic research and industrial design mainly focused in Fluid Power systems. Mr Del Vescovo holds a MS degree (2000) and a PhD (2005) from Polytechnics of Bari.*

Melissa Wilcox, P.E. *is a Senior Research Engineer in the Machinery Program at Southwest Research Institute. She is involved in a variety of experimental research efforts related to rotating machinery, including gas turbines, centrifugal and reciprocating compressors, and centrifugal pumps. In the past several years she has been a primary lead in experimental research of compressor performance in wet conditions. Ms. Wilcox holds a B.S. degree in Mechanical Engineering from Texas A&M University (2007) and a M.S. degree in Mechanical Engineering from the Georgia Institute of Technology (2012).*

ABSTRACT

A new technology challenge in centrifugal compressor design and operation is the condensate phase management. End users (especially in offshore and subsea operations) are more and more interested to have a Wet Gas Compression system which is able to tolerate liquid in the process gas. Authors' Company has initiated for several years a research program aiming to investigate the impact of the liquid phase on centrifugal compressor operability (mainly thermodynamics, rotordynamics, erosion, axial thrust).

As an introduction, the Authors' Company past experiences and more recent experimental tests [Ransom D. et al. 2011], [Bertoneri M. et al. 2012], are reviewed in order to show how the rotordynamic behaviour of a centrifugal compressor may be affected by the wet gas. However in the core, this paper is focused on the novel rotordynamic experimental outcomes of a wet gas single stage compressor test campaign. The machine was equipped with the following special instrumentation:

- Pressure and temperature probes along the flow path and internal seals
- Magnetic lamination installed on the shaft end to allow for stability test through a magnetic exciter
- Load cells installed in the thrust bearing
- Torquemeter installed at the compressor coupling

The explored test conditions were:

- Wet gas = Air and Water mixture up to 3% of Liquid Volume Fraction (LVF)
- Suction pressure levels = 10, 15, 20 bar-a
- Maximum Continuous Speed = 13500 rpm

The compressor went through an extensive test campaign where the following aspects were thoroughly investigated:

- Rotordynamic behaviour during steady state wet operation
- Rotordynamic stability (through magnetic exciter)
- Transient phenomena: response to liquid load variations (LVF up to 8%), start-up/shutdown from wet conditions, start-up with stratified flow into suction pipe

The compressor dynamic behaviour was monitored both from lateral viewpoint (using no contact probes located close to bearing locations) and axial/torsional viewpoint (through the special instrumentation described above).

Overall the compressor was able to withstand a huge amount of liquid phase, with an increased vibration level with respect to dry conditions but still in the safe area, both in steady and transient tests. Finally, major differences were found only at high flow – high liquid/gas density ratio conditions where an unexpected subsynchronous vibration (SSV) was showing up. The nature of this SSV was deeply investigated and finally it was fixed through a balance piston seal geometry change.

INTRODUCTION

The depletion of oil & gas fields is forcing the related Turbomachinery to work in more severe conditions. For centrifugal compressors the tolerance to certain amount of liquid phase in the main process gas is one of the most demanding requirements. The design of a centrifugal compressor able to process condensates is affecting different aspects, such as erosion and corrosion resistance, aerodynamic performances, mechanical design and (last but not least) rotordynamic behavior. One of the main goals of a wet tolerant centrifugal compressor is to increase the mechanical reliability due to liquid carryover.

The rotordynamic design of centrifugal compressor is related to reliability mainly through the following two aspects:

- Lateral critical speeds to be sufficiently separated from operating speed range
- Rotordynamic stability to be guaranteed in all operating conditions

Thus, the conceptual design of a centrifugal compressor able to handle a significant amount of condensates led to the following questions:

- **How much are the critical speeds affected by the liquid presence inside the process gas?**
- **What is the impact of liquid onto rotordynamic stability?**
- **Are there any other possible implications on rotordynamic behaviour when liquid phase is present?**

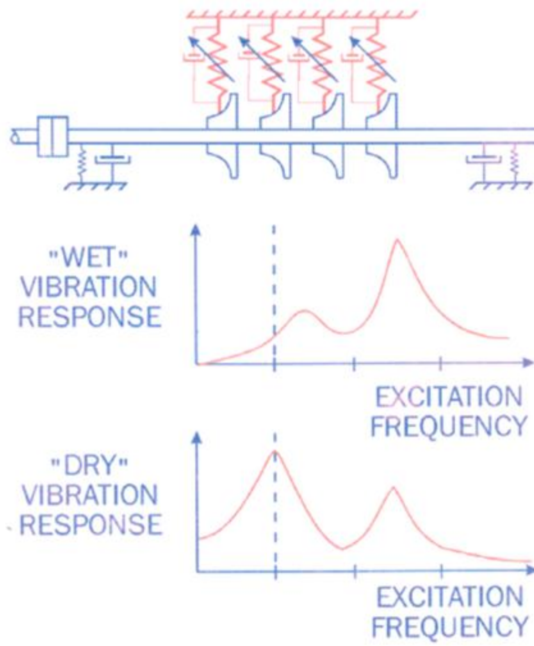

Figure 1 – Template rotordynamic model.

From theoretical standpoint, the model in Figure 1, given by a centrifugal rotor supported by journal bearings, has a first natural frequency which is roughly given by:

$$m_{eq}\omega^2 x + k_{eq} x = 0 \quad \rightarrow \quad \omega_n = \sqrt{\frac{k_{eq}}{m_{eq}}} \tag{1}$$

In centrifugal compressor analysis, the effect of the operating fluid in the rotor inertia terms m_{eq} is usually negligible. On the other hand, the operating fluid might have an impact on the equivalent stiffness k_{eq}. In the case of heavy/dense gas and damper seal presence, it is proven the effect of the gas load can change the position of the first critical speed. However, in most of the cases this gas-load effect on the critical speeds is

negligible. Thus it is fair to say for centrifugal compressors "dry" critical speeds perfectly match with "wet" critical speeds.
On the contrary, this is not true for centrifugal pumps. In this case, the "wet" critical speeds (e.g. the ones calculated taking into account the liquid-load effect on k_{eq} and m_{eq}) are significantly different (and usually higher) than the "dry" critical speeds (e.g. the ones that comes from the rotor simply modeled as a flexible shaft supported by journal bearing stiffness).
In the field of centrifugal pumps the fluid dynamic effects are mainly taking place in the annular seals and they are historically modeled by the Lomakin effect [Lomakin A., 1958]. The main outcome of Lomakin study was that when the rotor is spinning inside an annular seal and it is displaced from its centered position, the seal reacts with a strong radial restoring force (direct positive stiffness).

$$m_{eq}\omega^2 x + k_{eq}x = f_{lom} \tag{2}$$

$$f_{lom} = -k_{lom}x = -m_{lom}\omega^2 x \tag{3}$$

The magnitude of this restoring force is proportional to the Δp across the seal, which is linear with the pump head, which is finally roughly proportional to the rotational speed squared. Thus, the Lomakin effect was sometimes also modeled as a negative mass term (m_{lom}) which anyway led to the same exact conclusions: the system first natural frequency was increased due to the liquid annular seal effect.

$$\omega_n = \sqrt{\frac{k_{eq} + k_{lom}}{m_{eq}}} = \sqrt{\frac{k_{eq}}{m_{eq} - m_{lom}}} \tag{4}$$

Nowadays, this approach is overcome with the introduction of dynamic coefficients for each single seal of the rotor, creating a rotordynamic model representative of full-load conditions.
Each seal is modeled as an equivalent spring-damper element, having speed-dependent rotordynamic coefficients. While the prediction of rotordynamic coefficients of labyrinth seals operating with a single phase flow is a quite consolidated practice in the industry, the estimation of rotordynamic coefficients of labyrinth seals with a multi-phase flow is the objective of an ongoing research activity which is capturing the attention of both Academy and Industry leading to dedicated Joint Industry Projects.
As far as the Authors knowledge, all these efforts are still in the initial phase (the test rig is under construction) and few experimental data are available so far. [San Andres L., 2011] published a paper on a bulk flow code applied to damper seals which was tailored in order to take into account the liquid phase. The predictions showed not obvious results in terms of trend with LVF: a not monotonic trend of leakage and dynamic coefficients versus gas volume fractions and a stiffness hardening effect versus frequency for some gas volume fraction cases was found.
On the opposite side of the LVF operating range, that is to say on the side of the annular seals for gas tolerant centrifugal pumps application, there are two main reference works.
[Iwatsubo T. et al. 1991] published an interesting paper dealing with both the theoretical and experimental estimation of annular seals force coefficients. The subject of his research was a small plain annular seal (70mm diameter, L/D=1) which was tested in a multiphase rotordynamic test rig (air/water mixture) from full liquid up to 70% GVF (e.g. 30-100% LVF). Main outcome of the test was a general reduction of the force coefficients amplitude with the GVF and interestingly enough the liquid effect on the stability prediction of a simple rotor was also evaluated showing that the gas content increase is causing the system stability decrease (so from the perspective of a wet gas compressor the stability is increasing with the addition of liquid phase).
[Arghir M., et al. 2011] published an analytical paper dealing with the impact of a multiphase flow on the rotordynamic coefficients of textured annular seals (e.g. damper seals). While liquid plain annular seals are minimally affected by the gas presence, the textured seals are much more affected due to the increased fluid volume of interest (the pockets or cavities of the textured surface). A novel bulk flow model was developed which took into account the multiphase flow through the gas volume fraction parameter. The main outcome of the analytical study is the fact the multiphase annular seals show frequency dependent coefficients: for GVF values higher than 1%, all the dynamic coefficients (direct, cross coupled stiffness, damping and inertia) depend on the excitation frequency. This is in fact the typical behaviour of a gas damper seal like a honeycomb or hole pattern seal.

Besides the critical speed aspects, the seal dynamic coefficient modelling is important also to assess rotordynamic stability. Centrifugal compressor rotordynamic stability is normally demonstrated using two approaches:

- Technology reference maps
- Analytical calculations

While the analytical calculation is a more precise and detailed analysis, the use of referenced machines maps gives the designers and operators some practical indications of main parameters that influence rotordynamic stability. One of the most common technology reference diagram used with this purpose is the diagram originally proposed by John Fulton [Fulton J.W., 1984], where in the horizontal axis the fluid average density is represented and in vertical axis the Flexibility Ratio (MCS/NC1∞) is represented.
In wet gas compression, it is a common practice to consider homogeneous density as a first indicator of the processed fluid average density [Bertoneri M. et al., 2012].

$$\rho_h = \rho_g(1 - LVF) + \rho_l LVF \tag{5}$$

A significant increase of homogeneous density with respect to dry gas density could be possible.
Therefore the presence of liquid phase implies an increased averaged fluid density, creating a shift towards the critical area of the reference map, as showed in Figure 2.

About multiphase effects on stability [Bibet P. J. et al. 2013] presented an interesting paper about a novel multiphase pump equipped with a balance piston designed for a very high differential pressure (150bara). The relevant annular seal required a very careful optimization in order to reduce the destabilizing effects. A smooth seal design with three segments separated by swirl brakes rows was finally selected following a an extensive CFD analysis (focused both to swirl brakes aerodynamic design and to rotordynamic coefficients estimation in multiphase condition). A final test on a full scale prototype was performed with satisfactory results from rotordynamic viewpoint: only small subsynchronous vibrations in the high flow (low differential pressure) region. This is interesting if compared to the Authors' experimental findings described in details in the Operability paragraph.

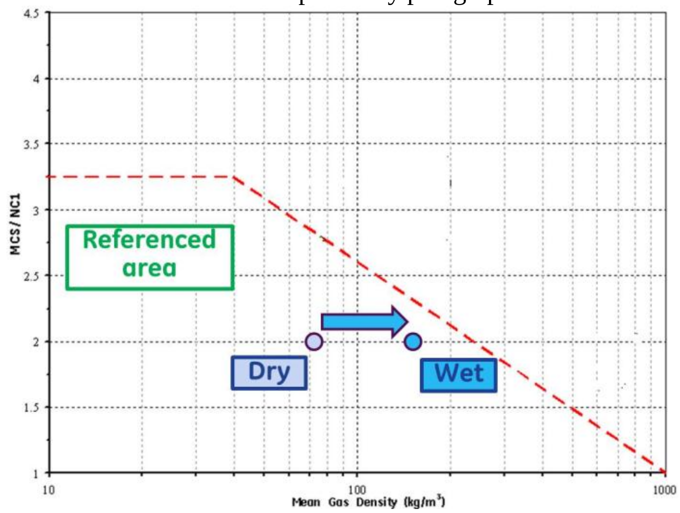

Figure 2 - Reference map for stability.

PREVIOUS FIELD EXPERIENCES AND TESTS

The first authors' Company field experience on wet gas testing was in 1972 with a back to back medium pressure multistage compressor for urea service, equipped with labyrinth shaft end seals. The reason for field testing was that the compressor had shown a performance decrease. Some inspections revealed that this performance decrease was due to labyrinth shaft-end-seals deterioration due to high vibrations. Therefore, some liquid injection tests were made in order to understand if the presence of liquid in shaft end seals could generate such high vibrations. Two kinds of tests were made:

Test A) Liquid injected in suction flange.

Test B) Liquid injected directly in shaft end seals.

Additionally, each test was repeated using both elliptical bearings and tilting pads bearings.

The results of these field tests were:

Test A). When the liquid was injected in suction flanges, no effect on vibration was noticed using both kinds of bearings. After the test, the compressor inspection showed no damage.

Test B). When the liquid was injected in labyrinth seals, a significant effect on vibration was noted. Specifically, the test using the elliptical bearings showed a sub-synchronous lateral vibration (~0.5Xrev) on both sides of the rotor. On the other hand, when tilting pad bearings were mounted, the system was stable although a sub-synchronous shock was noticed.

Therefore, the main finding of these tests was that the presence of liquid in the labyrinth shaft-end-seals can generate sub-synchronous vibrations: probably, when the seals are flooded, vibration phenomena similar to oil-whirl may happen. For this case, the sub-synchronous vibrations were evident with elliptical bearings, while they were smoothed with tilting pad bearings, due to their higher damping effect.

Other relevant wet gas field experiences are related to centrifugal compressors equipped with washing systems or cooling systems. When the gas composition and the operating conditions induce the formation (through chemical reactions) of deposits inside the flow channels, a washing system shall be provided. As shown in Figure 3, the system is composed by a series of injectors that spray the washing liquid in the top of the diffuser flow path.

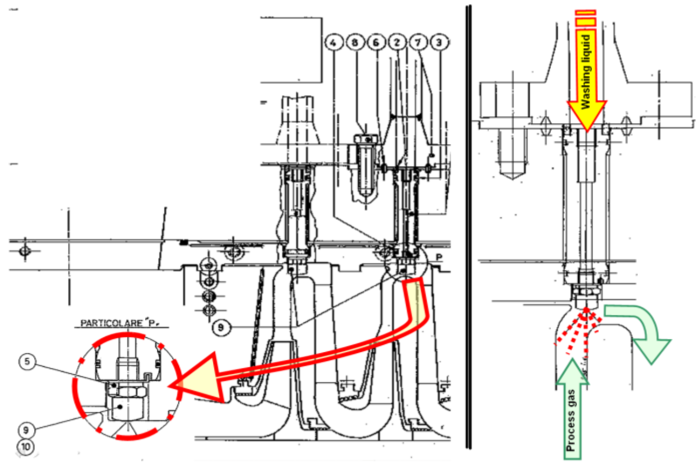

Figure 3 - Liquid injection for washing purposes.

Process gas cooling is another reason why liquid injectors may be used, like in the example of Figure 4 which shows a low pressure multistage compressor for a steam service.

In both washing and cooling injectors the maximum amount of liquid is limited with respect to the gas phase. However, it is worth to say that these machines have never generated field problems related to excessive vibrations due to the liquid injection.

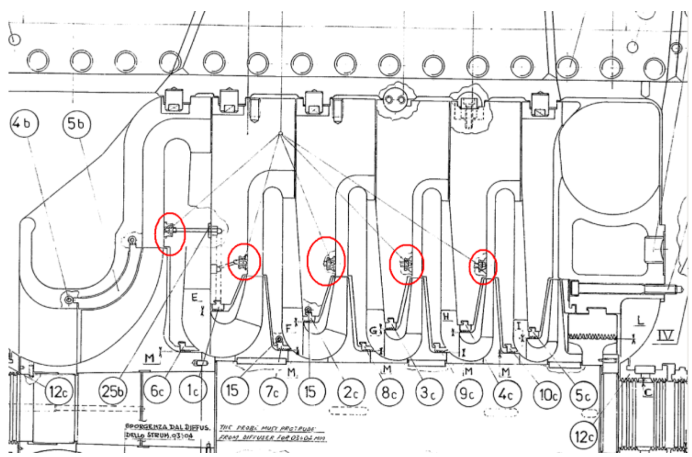

Figure 4 - Liquid injection for cooling purposes.

The first wet gas test campaign goes back to 1993, when the authors' Company first prototype of subsea motor-compressor was tested in a dedicated lab in Trondheim, Norway by a major EPC.

This compressor is a vertical multistage barrel compressor (8 stages) driven by an electric motor through an epicyclical gear and equipped with a lube oil system. The compressor is placed on the top part of a module which is composed of an inlet scrubber and a condensate pump in the bottom, as shown in Figure 5.

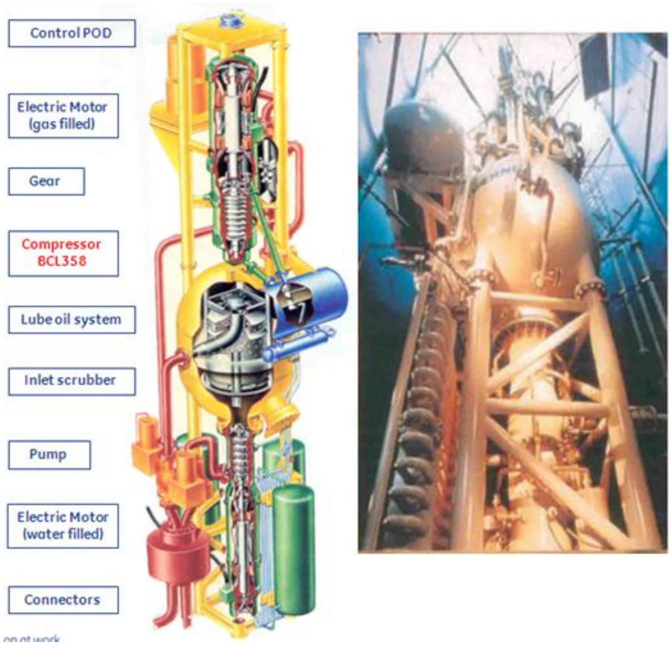

Figure 5 - Subsea compression station prototype.

The wet gas testing was made exposing the inlet of the compressor to nitrogen and heavy oil mixture bypassing the inlet scrubber. The purpose of the test campaign was to record machinery reactions to liquid content especially in terms of vibrations and speed reduction in order to assess the reliability of the design.
Figure 6 and Table 1 both show the tested points on compressor map and relevant conditions especially in terms of LMF/LVF.

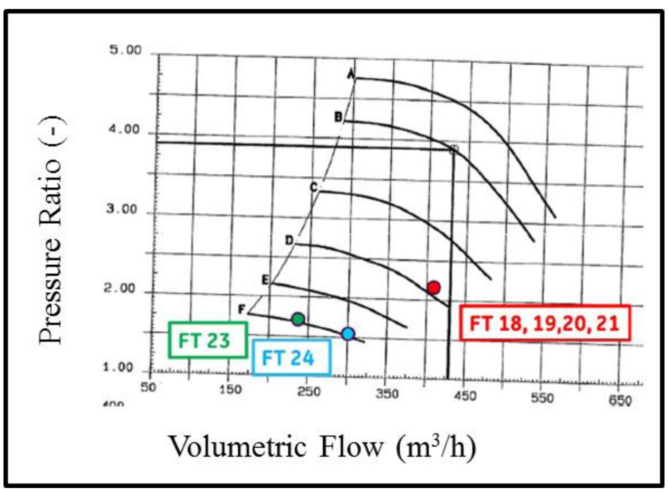

Figure 6 – Subsea compressor wet gas operating map.

Test No	Q1 gas [m3/h]	P1 [Bar-a]	T1 [C]	LVF [%]	LMF [%]	DR Liquid/Gas	P2 [Bar-a]	T2 [C]	PR
FT18	404	14.97	27	0.81	27.32	46.59	34.27	72	2.29
FT19	420	14.88	31	1.18	35.92	47.41	34.30	74	2.31
FT20	414	14.92	31	0.61	22.20	47.16	34.20	72	2.29
FT22	421	14.96	30	2.03	49.16	47.19	31.40	69	2.10
FT23	238	31.11	30	1.09	19.54	22.58	45.15	46	1.45
FT24	297	29.99	31	1.84	29.77	23.49	45.60	58	1.52

Table 1 – Subsea compressor wet gas test conditions.

The results were very satisfactory, since the machine was able to handle liquid loads without any rotordynamic problem.

Finally in 2010 a two stage centrifugal compressor was tested at a major Research Institute with air/water mixture up to 5% LVF. Mechanical results are described in [Ransom D. et al. 2011] however here below the main outcomes are reported:

- Wet gas operation had a negligible effect on radial vibrations.
- A low frequency axial vibration was noticed.
- An increased axial thrust was measured.

Another OEM did similar tests in the past whose results might be useful for our discussion. In 2005 a single stage compressor was tested at K-lab with natural gas and hydrocarbon mixture [Brenne L. et al. 2005]. The main findings from rotordynamic standpoint are that:

- Machine vibrations were not significantly affected by the liquid hydrocarbon condensates.
- A 0.5Xrev subsynchronous vibrations was appearing when the gas quality was reduced, suggesting that liquids could have been entrained in the seals

Moreover there is another relevant publication from the same OEM [Griffin T. et al. 2011] where stability tests results are shown at various LVF levels. Although the compressor is not a wet gas compressor but an integrated separator-compressor, it is shown a slight increase of logarithmic decrement with an increased LVF.

TEST SETUP

The authors' Company designed and tested a wet gas loop at a major Research Company in order to assess rotordynamic effect of wet gas compression. The test object is a single stage centrifugal compressor that was not optimized for wet gas operation (standard dry design).

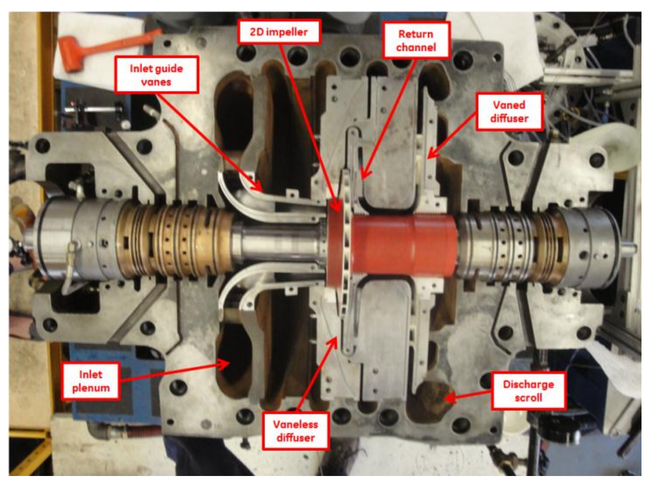

Figure 7 - Single stage compressor section.

Figure 7 shows a picture of the compressor that underwent wet gas compression tests, while a detailed description on test setup and loop operation can be found in [Bertoneri M. et al., 2014].
Shaft rotordynamic behavior experimental assessment was performed from both lateral and torsional standpoint. From torsional analysis viewpoint the test rig performed a static/dynamic measurement using a strain-gage based torquemeter located between the gearbox and the compressor, able to catch torsional phenomena from DC component up to 3kHz.

Lateral behavior monitoring was made through standard no contact probes located at both bearing sides for both gear and compressor.
Moreover the shaft behavior in terms of stability was investigated through a magnetic exciter installed on the NDE side of the compressor shaft. This test has been the first time a magnetic exciter has been mounted in wet conditions which required a careful study of exciter mechanical arrangement preventing water and oil from entering into the exciter housing.
The magnetic exciter is a flux controlled magnetic bearing developed by a major magnetic bearings supplier able to excite the shaft through a rotating force. This force can be controlled in amplitude (up to 320 N) and in frequency (from 5Hz up to 300Hz) so allowing a full investigation of the rotor harmonic response. Frequency sweep windows and time slopes can be adjusted in real time during the test running by injecting phase lagged sine waves generated by proprietary software. Real time and post processing order tracking can be performed as well in order to allow the analysis of shaft vibration filtered on the exciting frequency.
Moreover an instrumented thrust bearing with on-board strain gage load cells allowed the static/dynamic monitoring of axial thrust during the test.

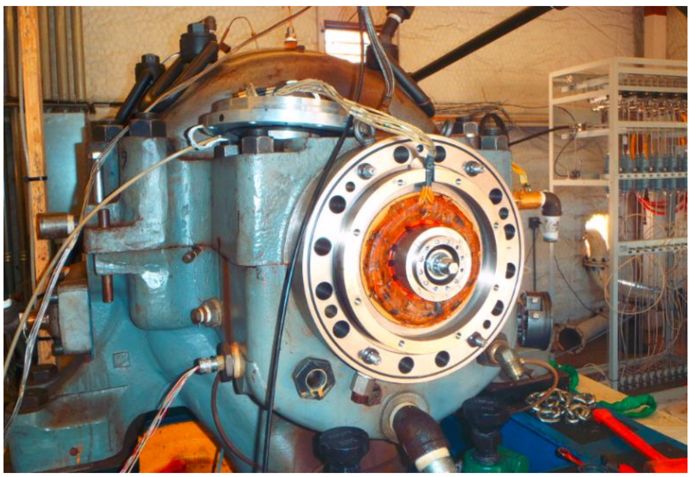

Figure 8 - Magnetic exciter assembled at NDE.

ROTORDYNAMIC MODELIZATION

During the conceptual design phase of the test, the rotordynamic analysis was instrumental to design the rotor, journal bearings and seals in order to reach the main goals previously mentioned (e.g. to detect the potential effect of liquid phase on both critical speeds position and stability).
The rotor modelization (see Figure 9) was performed using a dedicated finite element code; the model was later on fine-tuned based on the available rotor ping test data. The rotor modelization did not present critical points since it was mainly a slender shaft with one shrunk fitted impeller at the midspan. The rotor region upstream the impeller was made of thinner section on purpose in order to make the rotor more flexible and potentially more sensitive to the seals dynamic effects.
Moreover the presence of the overhung magnetic exciter laminations on thrust side was of importance for a correct modelization since their impact on the first mode shape is quite high. The model tuning was especially focused on the overhung mass amount. Finally the tuned model was able to fully capture the first free-free natural frequency (190Hz). The agreement was a bit worse on the higher order mode shapes but this was not considered of importance since those modes were much higher than the running speed (2^{nd} mode was measured @ 440Hz while 225Hz was the applicable MCS).
Since it was a single stage rotor, the rotordynamic behaviour was not fully representative of a real centrifugal compressor: see the relevant UCS map showed in Figure 10. Anyway the design of the rotor-bearing system was tailored in order not to have such a high damping on first rotor mode and so to maximize the potential sensitivity to the liquid presence when crossing the first critical speed. The second mode was moreover very well damped due to the specific shape and it was not evident at all in the rotor response.

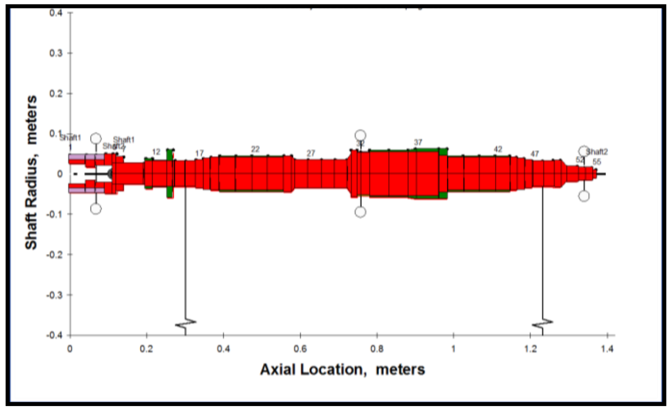

Figure 9 – Rotor model.

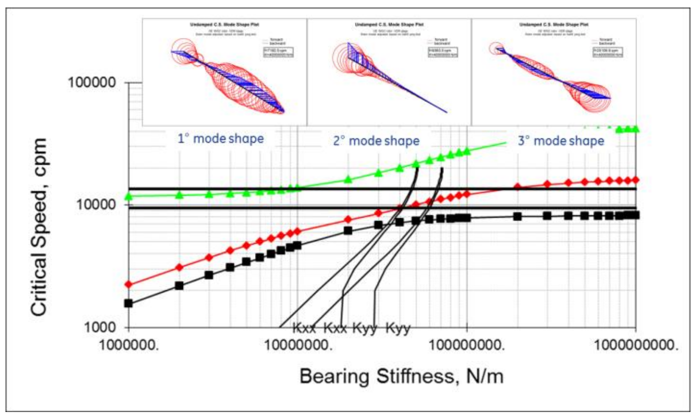

Figure 10 – UCS map and rotor mode shapes.

Additionally since the pressure level was relatively low and the pressure drop very limited (one stage only) the stability features were well within the manufacturer experience envelope as already showed in Figure 2. The same considerations regarding the damping of the first critical speed applies also to stability: the design was aimed to have a low Log Dec in order to maximize the potential effect of the liquid.

TEST RESULTS

The explored test conditions are listed below:

- Suction pressure levels = 10, 15, 20 bar-a
- Rotational speed levels = 9500rpm-11500rpm-13500rpm
- LVF = Air and Water mixture up to 3% steady state

In order to address the main aspects mentioned in the introduction paragraph, the compressor went through an extensive test campaign where the following aspects were thoroughly investigated:

- Overall rotordynamic behaviour during steady state wet operation
- Impact of wet phase on critical speed location (Wet startup/shutdown)
- Rotordynamic stability
- Compressor Operability:
  - Slug testing
  - Water injection in shaft end seals/impeller eye seal/balance piston seal
  - SSV investigation @ high flow

All these aspects will be fully described in the following paragraphs.

Overall rotordynamic behaviour during steady state

Process steady state behaviour was reached mainly during the compressor thermodynamic test. Vibrations were measured by no contact probes located close to the journal bearings. Overall the 1XREV vibration was not heavily affected by the presence of the liquid phase while the direct vibration was in general slightly increased due to a sort of widespread noise (both sub and super-synchronous).

Figure 11 is showing the direct vibration trend for both radial probes along one selected direction versus the liquid mass fraction. There is never a steep rising trend and, more important, vibrations do not change significantly from the dry case being always below the alarm level (70µm peak to peak).

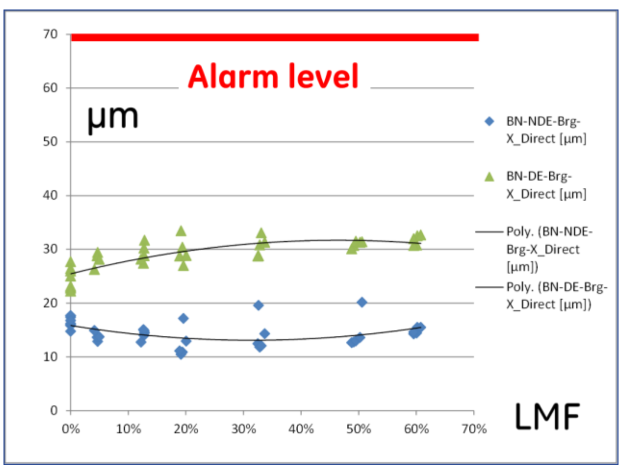

Figure 11 – Direct Vibration Trend vs. LMF.

Lateral critical speeds in wet gas conditions

Shutdown and startup tests in wet conditions have been performed in order to detect wet critical speeds.

The shutdown test was quite challenging because in a closed loop the liquid volume fraction is increasing while the compressor is coasting down, as shown in Figure 12.

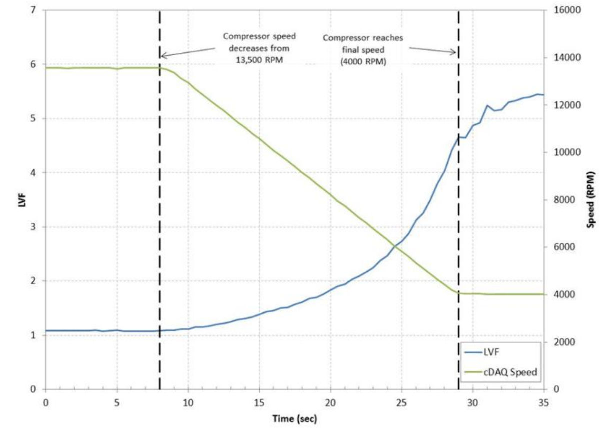

Figure 12 - Wet shutdown test: LVF trend.

The test was anyway successfully completed. The comparison of different shutdown Bode plots (see Figure 13) both wet and dry demonstrate that the critical speed position is not significantly affected by the liquid carryover.

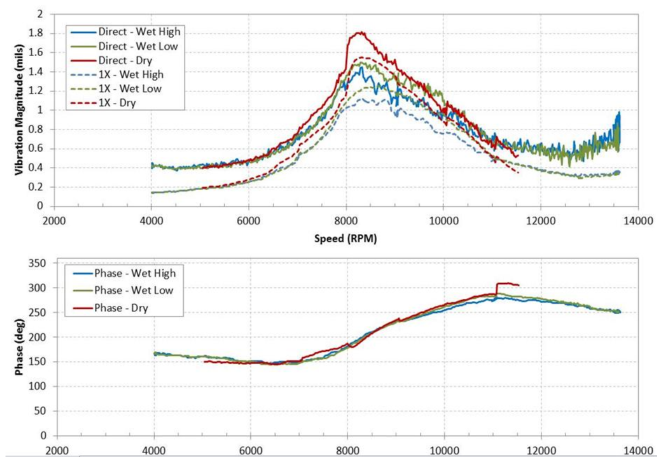

Figure 13 - Comparison between dry and wet shutdown.

Rotordynamic stability test

As mentioned in the previous paragraph the compressor was equipped with a magnetic exciter in order to perform stability testing and measure the rotor Log Dec in presence of liquid phase. The stability test and the relevant postprocessing was based on the system frequency response approach according to the same methodology also reported by other researchers see [Moore J. et al. 2002] and [Pettinato B. et al. 2010]. The magnetic exciter was used to apply a sweeping frequency forward rotating force having constant amplitude while the no-contact probes were used to measure the relevant rotor frequency response. The frequency range of interest was centered across the first rotor mode which was expected to be in the range of 130Hz.

As a first step, a test in dry conditions and with no compressor load was done to establish a baseline: only rotor and journal bearings were active here.

The first rotor mode Log Dec was measured to be 0.33 at 129Hz frequency.

Later on the load was applied still remaining in dry conditions and the Log Dec increased significantly to 0.45-0.55 level depending on the suction pressure.

Regarding the effect of the liquid phase, the outcome was finally positive since the Log Dec showed little but positive changes depending on the specific test conditions.
Typically the test was performed keeping the same rotational speed and compressor volumetric flow and increasing the LVF amount after performing the sweeping frequency excitation. The inlet oil journal temperature was controlled to be as much constant as possible. Overall two frequency sweeps were performed each time and the average between the two was considered as a final result.
Figure 14 is showing the Log Dec trend versus LVF for a specific test condition:

- Speed: 11500rpm
- Suction pressure: 15bar

The trend is clearly increasing and the same results are obtained using two different identification methods:

- SDOF: single FRF curve fitting through a function typical of a single DOF system
- MDOF: identification technique considering multiple FRFs simultaneously and many DOFs (up to 4 in this specific case)

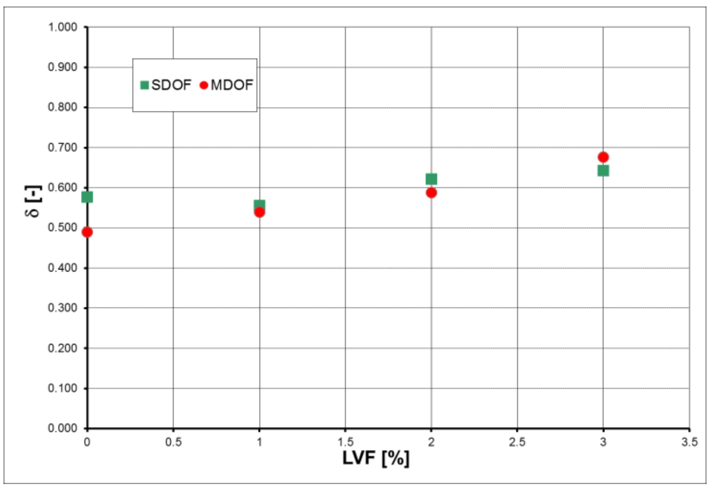

Figure 14 – Log Dec vs. LVF.

Operability: transient liquid slugs

In the wet gas compressor test rig, the liquid mass fraction was a controlled variable. Most of the tests have been conducted in a steady state condition. Anyway, in real field conditions, the liquid load could change over time with a quite fast dynamics. While the field conditions shall have very large characteristic time, the compressor suction piping system and the compressor scrubber could generate liquid load variation into the compressor inlet flange. Therefore, the compressor capability to withstand liquid load variations needs to be assessed and dedicated transient tests were performed. While the compressor was operating near the design condition, the following variations were tested:

- LVF oscillating periodically between 0.4% and 1.4%
- Impulse tests in order to simulate “slugs” according to both Figure 15 and Table 2.

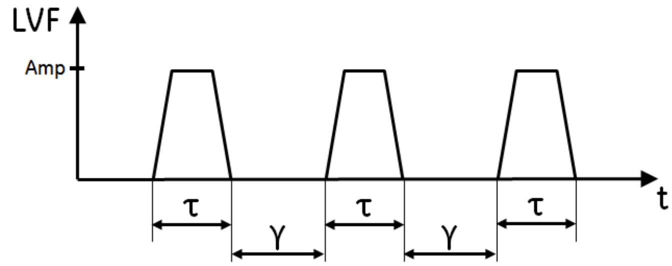

Figure 15 – Impulse test scheme.

τ	γ	LVF Amp	Corresponding LMF
30”	5’	0.5% on impulse	20%
5’	30”	0.5% on impulse	20%
30”	3’	2.0% on impulse	50%
3’	30”	2.0% on impulse	50%
30”	5’	5.5% on impulse	73%
10’’	5’	5.5% on impulse	73%

Table 2- Impulse test scheme legend.

The compressor was finally able to withstand all the transient operations. As an example of the typical results, the following plot is showed: the vibration trend of all the compressor radial probes tracks the LMF trend and vibration step changes are limited within the alarm level (70μm peak to peak).

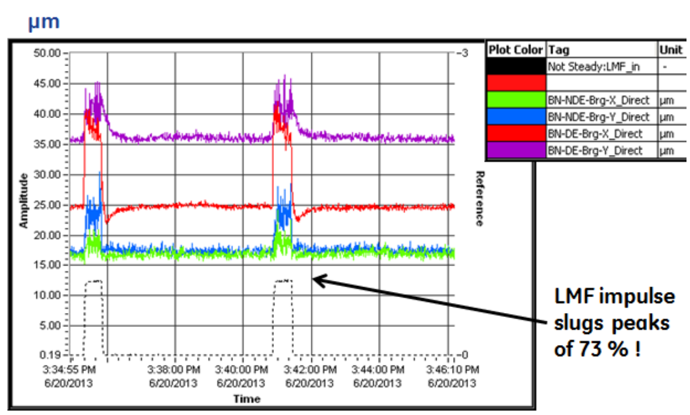

Figure 16- Transient test results.

Operability: water injection in labyrinth seals

While the compressor was operating in the low flow area and in dry condition, the water was injected within the labyrinth seals in order to isolate the effect of possible labyrinth seal flooding on compressor stability. Dedicated feeding holes in the seal stator parts and a valve system were realized to make this test possible (see Figure 17).

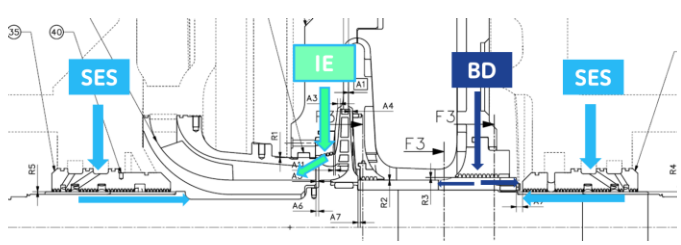

Figure 17 – Seal water injection scheme.

Stability test using the magnetic exciter was then performed in different conditions in order to assess the relevant rotordynamic stability (for such test, the rotational speed was 11500rpm and the suction pressure was 10bara).

Condition	Log. Dec. (-)	Frequency (Hz)
Dry	0.535	131.7
+ IE flooded (impeller eye seal)	0.573	132.4
+ BD flooded (balance drum)	0.593	132.3
+ SES flooded (shaft end seals)	0.489	132.5

Table 3 – Result summary for seal flooding test.

The compressor still showed to be very stable even when operating with flooded seals. It is important to remark that those stability tests have been conducted near the surge line of the compressor (at minimum flow).

Operability: startup from wet SOP

These tests have been conducted in order to simulate a pressurized start-up after a wet shutdown. The test procedure was:

- Compressor was operating in a stable wet gas condition.
- Compressor was tripped while the liquid pump was still injecting water into the loop.
- The water level in the horizontal suction pipe line was visually checked with a level gauge.
- When the level reached the desired value (e.g. almost half of the pipe diameter) the pump was stopped.
- The compressor was finally re-started with partially-flooded suction pipe and internals.

These tests were done successfully up to a 50% of suction pipe flooded. Starting torque was increased as well as the vibrations, but the compressor was still able to re-start with no need to open the bottom drains. The relevant torque trend is shown in the following picture, where two different levels of pipe flooding are considered.

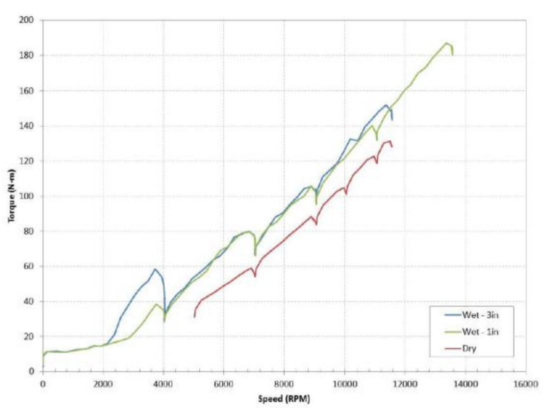

Figure 18 - Startup torque for wet SOP testing.

Operability: SSV investigation at high flow – high density ratio

One of the most interesting results from the rotordynamic viewpoint was the presence of an unexpected SSV when operating the compressor at high flow.

The SSV amplitude was quite high (e.g. comparable to the 1XREV vibration amplitude) so it was critical to investigate the relevant root cause in order to have the full control of it in the perspective of future wet gas compressors. Figure 19 is showing a typical waterfall plot of compressor lateral vibrations when it was operating in the high flow region. Synchronized with the waterfall there is the LVF trend displayed on the bottom part. In this specific test, the gas flow was kept constant and the liquid flow was reduced step by step starting from a LVF=3% up to zero. The other operating parameters were settled as follows:

- Rotational speed: 13.5krpm
- Suction pressure: 10bara

From the same figure, the following main aspects are evident:

- 1XREV frequency is constant at 225Hz (steady run).
- A 0.5Xrev is always present in the spectrum (both dry and wet) so this is something not related to the wet phase.
- The SSV is occurring at a frequency which is about 0.45Xrev and it is slightly changing with the LVF level.
- The SSV amplitude is also changing with LVF showing a not monotonic trend: maximum amplitude reached in this condition was about 18μm peak-peak @ LVF=0.7% versus a synchronous vibration which was around 10μm.
- There was a threshold LVF which was about 0.5%.

As an additional comment, the SSV occurrence was almost synchronized with the liquid injection at the threshold level: no evident time lag was detected between the liquid injectors opening/closure and the vibration occurrence/disappear.

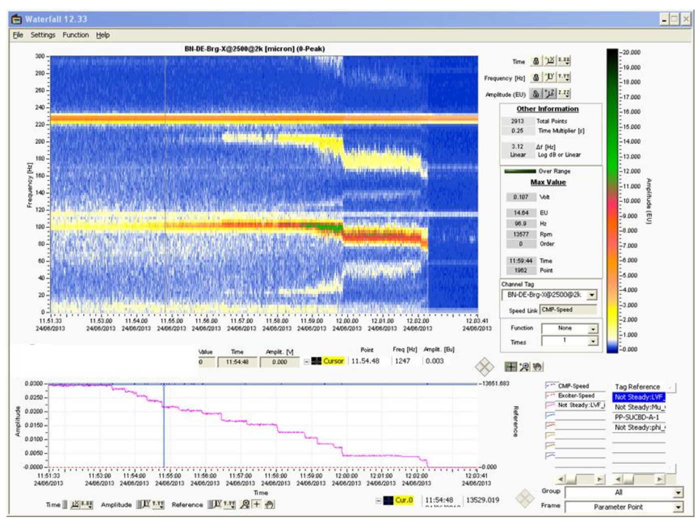

Figure 19 – Typical waterfall plot of lateral vibrations @ high flow coupled with the LVF trend.

The investigation performed during the test campaign allowed to collect the following further information:

- The SSV frequency was tracking the running speed: this was considered as a symptom of a forced excitation and not of a self-excited natural frequency.
- The SSV presence was detected at high flow only: keeping

the same LVF and increasing the compressor load led to SSV fade-off.
- The SSV amplitude was higher for lower suction pressure at given speed and flow conditions.

The two last hints together led the authors to think that the source of the phenomenon might have been located in the balance piston seal (the vibration phenomenon was in fact very strictly related with the pressure drop across the balance piston). From a simple conceptual viewpoint, the liquid in the process might have been trapped in the balance piston seal causing the fractional speed vibration as it was a sort of "liquid whirl". The liquid whirl build-up is conceptually consistent with the seal geometry (a tooth on stator labyrinth seal) which is made of annular cavities with no circumferential brake.
In order to substantiate more this Root Cause Analysis a further test was done where the pressure measurement lines in the seal cavities were pressurized in order to purge the seal (in fact the seal was instrumented with static pressure taps in a couple of intermediate cavities and these lines could be purged time by time with dry nitrogen). The results are showed in Figure 20.

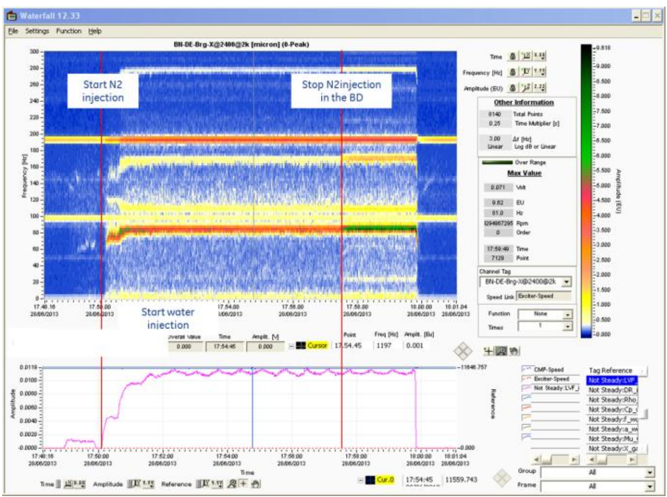

Figure 20 – Waterfall plot showing seal dry purging effect.

Interestingly enough, the SSV amplitude was decreasing during the nitrogen injection showing a clear relationship between the seal inner flow distribution and the vibration. The purge lines were only of 1.5mm diameter size so the limited effect (the SSV was reduced but it was still present) could be explained with the minimal amount of dry nitrogen injected.
On this basis it was decided to do a more significant change in the compressor replacing the original labyrinth seal with a Pocket Damper Seal: the basic concept behind this decision was the possibility to have physical brakes in the circumferential direction thanks to the partition walls presence. The two seal geometries (laby and PDS) are compared in Figure 21.
This kind of seal was also previously tested inside the authors' Company showing promising results from rotordynamic viewpoint. See [Ertas B. et al. 2011]. The seal length and clearance were unchanged with respect to the original laby seal in order to establish a fair comparison.
Figure 22 is showing the result of the re-test of the compressor in the same conditions previously mentioned. As it is evident from the waterfall plot, the SSV almost disappeared this time, being reduced from almost 20μm down to a few microns only.

The test was extended to other operating conditions based on different suction pressure and speed levels with the same very satisfactory results.

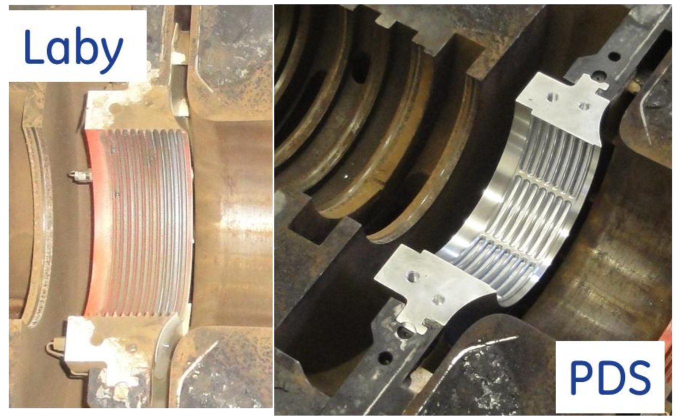

Figure 21 – Laby and PDS installed on the balance piston.

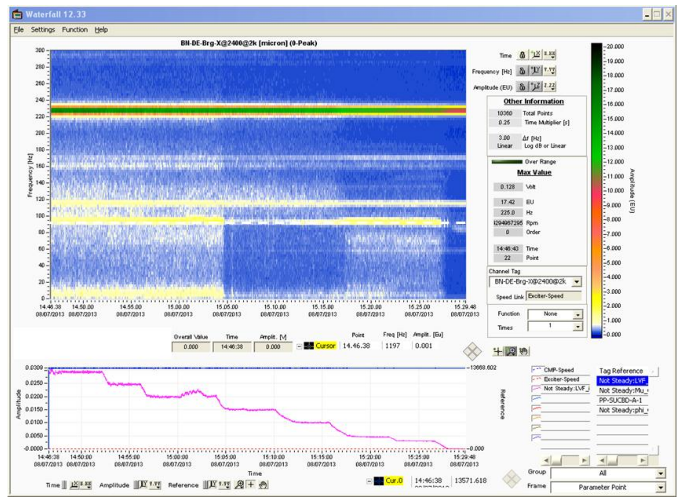
Figure 22 – Waterfall plot of re-test with PDS.

Later on, in the attempt to better capture the physics behind the test results a multiphase CFD investigation was launched together with the support of a major engineering consulting Company. Even if the study is currently ongoing at the time of this paper preparation, some interesting conceptual results may be anticipated in order to better substantiate the experimental results (details will be showed in a future dedicate paper).
The CFD run was setup in order to simulate the same test condition showed above. The model boundary conditions for the balance piston seal are well known in terms of pressure and temperature but not at all in terms of preswirl and LVF. Preswirl was assumed to be zero due to the specific upstream geometry (gas was coming into the seal from a statoric channel) while the LVF was a tuning factor (even if the compressor inlet LVF was controlled, the seal inlet LVF is not). Both Laby and PDS were subject of the simulation.
The major result from this analysis is a couple of simple but interesting concepts:
- The Tooth on Stator laby seal is susceptible to accumulate liquid on the walls due to the centrifugal force which is related to a medium-high circumferential fluid velocity in the cavities.

- The Pocket Damper Seal is less susceptible to accumulate liquid in the pockets due to the presence of the partition walls which are capable to break the fluid circumferential velocity hence reducing the centrifugal force.

Figure 23 is showing the main differences coming out from both the simulations. Both transients are based upon a 30% inlet LVF (tuning result) and came up to a steady state. While the laby is accumulating a lot of liquid in the first half of the seal length, the PDS remains almost empty.

This result is perfectly in agreement with the test outcome and it is confirming the original intuition which led to the PDS selection.

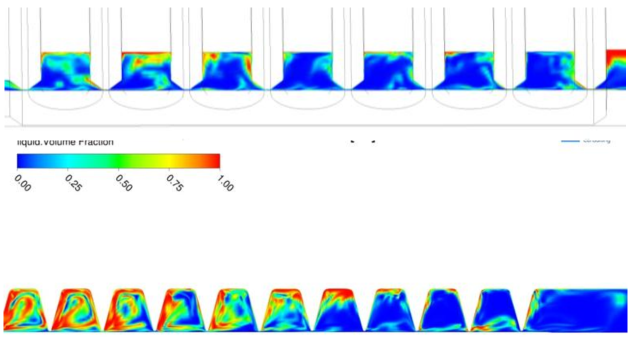

Figure 23 – CFD results: PDS (top), Laby TOS (bottom).

CONCLUSIONS

This paper is discussing how the liquid phase could affect the rotordynamic behavior of a centrifugal compressor. At first, some basic theoretical concepts highlight two main possible impact areas:

- Critical speed position
- Rotor stability

An overview of authors' Company field experience and testing is presented, from which no major issues have been found.

A dedicated single stage test campaign has been conducted by the authors' Company together with a major research institute in 2013. The main findings from rotordynamic standpoint are:

- Overall vibration level at steady state is far below the alarm level even with very high levels of liquid carryover in most of the compressor operating range
- Wet gas does not affect critical speed position
- Rotordynamic stability is slightly improved by the presence of liquid: from Figure 14 the δ increase is in average +6% for each 1% LVF step increase.
- Centrifugal compressor design is robust towards operational transients, such as liquid loads variation, shutdown and startup with wet gas
- Flooded balance piston may generate whirl phenomena at very high flows and with high liquid-gas density ratio.
- A swirl breaker device, such as a pocket damper seal, is able to suppress the liquid-whirl phenomena.

Therefore, considering both the test findings and the previous field experience, the main conclusion is that centrifugal compressor technology is a viable solution for wet gas compression due to its robustness in rotordynamic behavior with liquid carryover.

Moreover, some design adaptation is needed in order to optimize compressor reliability, such as swirl braking device in balance piston seals to avoid SSV phenomena due to possible seals flooding effect.

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NOMENCLATURE

k_{eq}	= Equivalent Stiffness	(N/m)
k_{lom}	= Lomakin Stiffness	(N/m)
f_{lom}	= Lomakin Force	(N)
GVF	= Gas Volume Fraction	(-)
Log Dec	= Logarithmic decrement	(-)
LMF	= Liquid Mass Fraction	(-)
LVF	= Liquid Volume Fraction	(-)
NC1∞	= First Critical Speed at infinite support stiffness	(rpm)
m_{eq}	= Equivalent Mass	(kg)
m_{lom}	= Lomakin Mass	(kg)
MCS	= Maximum Continuous Speed	(rpm)
ρ_g	= Gas Density	(kg/m3)
ρ_l	= Liquid Density	(kg/m3)
ρ_h	= Homogeneous Phase Density	(kg/m3)
ω_n	= Natural Frequency	(rad/sec)
ω	= Rotational Frequency	(rad/sec)
DE	= Drive End	
DOF	= Degree of Freedom	
FRF	= Frequency Response Function	
MDOF	= Multiple Degree Of Freedom	

NDE = Not Drive End

PDS = Pocket Damper Seal

SDOF = Single Degree Of Freedom

SOP = Settling Out Pressure

SSV = Subsynchronous vibration

1XREV = 1X vibration (synchronous)

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