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PERFORMANCE TEST OF A LIQUID TOLERANT IMPELLER AND VALIDATION OF WET COMPRESSION PREDICTIVE MODEL

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ABSTRACT

Standard centrifugal compressors are designed to process gas with limited liquid content: significant presence of liquid may be responsible for the rapid erosion and corrosion of both static and rotating components that yield to higher failure risk and shorter maintenance periods. Moreover the presence of liquid is associated to significant modifications of machine performance and machine operability. However, the possibility of compressing gas containing non negligible amounts of liquid allows to reduce size, dimensions and costs of the liquid removal stations with a consequent positive impact on the layout of subsea compression stations. With this aim, the OEM has developed a family of centrifugal impellers, the Wet Tolerant Impellers (WTI), specifically designed to process gas containing non negligible amounts of liquid, increasing the erosion resistance and promoting the liquid droplets breakup

for subsequent stages.
The validation of WTI has been carried out at the Norwegian University of Science and Technology (NTNU) in Trondheim; the test rig is a single stage air-water multiphase open-loop facility driven by a 450 kW motor capable of a maximum continuous speed equal to 11000 rpm.
In the test campaign, first a standard impeller has been tested to assess the impact of liquid phase on performance and operating range (results have been summarized in Ferrara et al. [14]), then the specific WTI design has been tested and directly compared with the standard impeller results. The comparative test campaign has been focused on three different aspects:

- Performance evaluation in dry and wet conditions;
- Operating range evaluation in dry and wet conditions;
- Comparative erosion test.

WTI test results have been compared with the performance curves obtained through the OEM internal tool specifically developed for the prediction of centrifugal compressors performance in wet conditions. Performance has been assessed varying the amount of liquid carried by the gas at two different rotating speeds: five Liquid Mass Fractions (LMF) have been considered for each speedline and a minimum of six test points have been recorded from partflow to overflow. The effect of liquid on machine operability has been assessed through a left-limit investigation by means of dynamic pressure probes readings in order to evaluate the stall/surge behavior for different values of LMF. In addition, the function between head rise to surge and LMF has been reconstructed.
Tests results have shown the effectiveness of liquid tolerant design: erosion effects are weaker on WTI while performance levels and operating range are comparable to those of standard impellers.

INTRODUCTION

Centrifugal Compressors (CC) are usually designed to operate in dry conditions; only limited amounts of liquid can be tolerated without a specific design. In presence of significant amounts of liquid carried by the gas stream, the installation of separation devices upstream of the CC are required to avoid mechanical damages and operability/performance issues. Although necessary, gas-liquid separators increase significantly the footprint, the installation complexity and the operating costs of the compression stations, thus great research effort is made in order to improve and develop new technologies to reduce the costs. One viable way is represented by the liquid tolerant compression, namely the compression of a gas containing a significant amount of liquid; through this technology the efficiency and size of the upstream gas-liquid separators can be reduced (or removed), reducing the installation and operating cost of the overall compression stations. In the recent years, great attention has been put on this new process which has revealed many advantages in particular for the subsea installations: although the content of liquid is currently limited to 5% on a volume basis [1], the WGC technology has shown promising capabilityies in the reduction of compresion stations footprint.
Even if limited to very low Liquid Volume Fractions (LVF), the presence of a liquid phase can produce severe negative effects on the CC: larger mechanical stresses due to erosion and corrosion have been observed together with significant variations of performance and machine operability. These particular issues have been documented by Bertoneri et al. [2] and Fabbrizzi et al. [3] who, despite the evident mechanical limitations, have also established suitability of centrifugal compressor for processing gas-liquid mixtures.
Recently, Ferrara and Bakken [4] - [6] have highlighted the mechanism leading to unstable phenomena (like stall and surge) in wet conditions, together with an investigation of the variations of operating range. They observed that the liquid phase has a stabilizing effect on unsteadiness coupled with a delayed instability onset. As a result, it seems that operating a compressor in wet conditions does not limit the machine working range. Similar results have been shown also by Grüner and Bakken [7].
In order to prevent erosion and mechanical damages of the stage and the downstream components, a novel impeller design has been developed specifically for WGC applications. The novelty of impeller lies in the capacity of process working fluids with significant amount of liquid with limited erosion phenomena.
The WTI has been installed on a multiphase open-loop rig, with a single-stage centrifugal compressor and tested during an experimental campaign. The main goals were the estimation of performance in dry and wet conditions, the evaluation of operating range in the presence of water and the assessment of the erosion resistance in comparison with standard CC impellers.

CRITERIA FOR LIQUID TOLERANT DESIGN

The severe erosion processes that affect the CC stages when treating multiphase flows are mainly due to the continuous impact of liquid droplets over localized areas of the impeller. Based on this evidence, an innovative aerodynamic design has been developed in order to ensure the resistance to the erosion. The resulting WTI design is based on the following concepts:

- optimized blade shape to reduce the intensity of droplet impacts;
- optimized flowpath to promote uniformity of droplet impingement;
- promotion of the droplet fragmentation along the impeller.

Since the erosion resistance has been shown to be the first concern in WGC applications, most of the development has been focused on erosion-prevention concepts. However, if not properly integrated into the aerodynamic design, the liquid tolerant features can limit the stage performance in both dry and wet conditions. For this reason, the design has been optimized in order to limit the dry performance decay, keeping unaltered the tolerance to wet conditions. The stage design has been obtained by means of multiphase CFD where turbulent flow field has been coupled with the Lagrangian particle tracking of the dispersed (liquid phase). Both fluid-fluid interactions and wall-droplet interactions have been considered to keep in account a wide range of effects, i.e. droplets breakup, wall bouncing or splashing. Similar techniques have already been adopted for the study of droplet erosion [8] – [9]. Multiphase CFD simulations have shown more uniform droplet distribution along the WTI flowpath in comparison with the standard impeller (see Fig. 1 (a) where size of droplets along flowpath is shown; scale is such that droplet size is divided by droplet size at impeller suction); droplets do not segregate in a localized region of the impeller (see Fig. 1 (b)). As a result the droplet impact regions are larger and more uniformly distributed and, ultimately, the erosion phenomena are limited. It is worth to notice that the droplet diameter and distribution at the pressure side of the trailing edge seems to be dramatically reduced, preventing the erosion in one of the most critical regions. Multiphase CFD simulations have also shown an increased droplet breakup rate through the WTI stage; depending on the flow condition and the inlet liquid volume fraction, droplet diameter reduction from 30% to 80% is expected. In particular, the stage design has been optimized to have higher breakup rates at large LVF, where larger erosions are expected (Fig. 2). As depicted in Fig. 2, in comparison with the standard impeller (STD), the WTI shows almost doubled breakup efficiency.

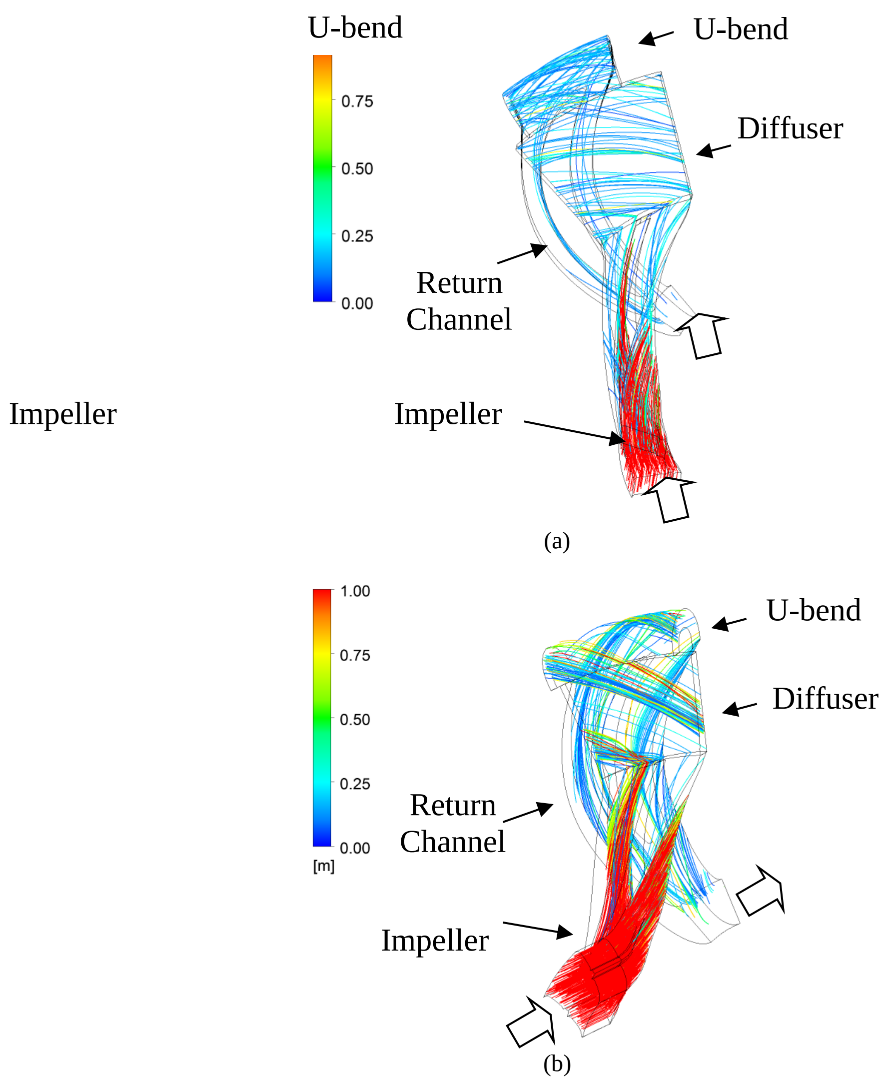
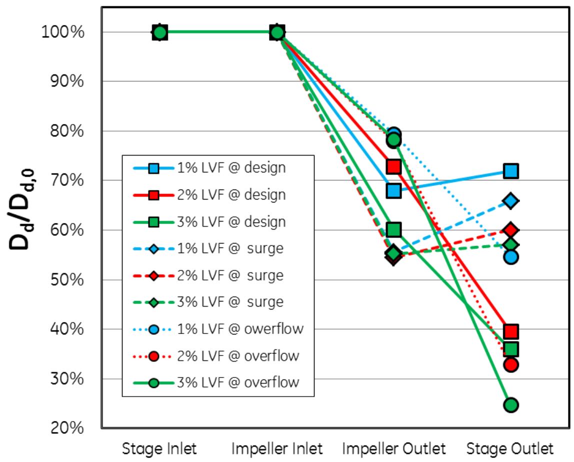
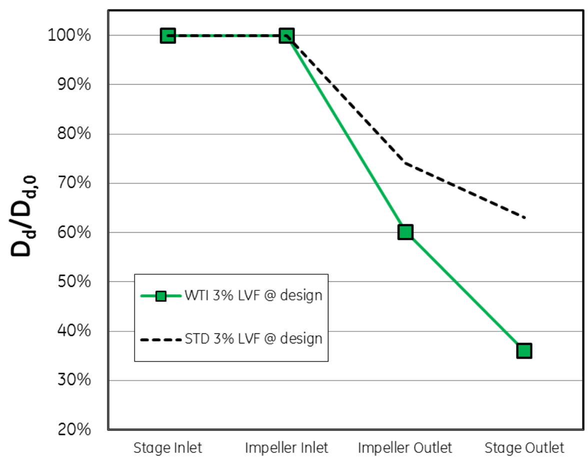


Fig. 1: Droplets streamlines size along the stage for WTI impeller (a) and in comparison with the standard impeller (b). Streamlines are colored with the local diameter D_d that is normalized with the inlet average diameter $D_{d,0}$.



(a)



(b)

Fig. 2: Droplets average size along the stage for WTI impeller (a) and in comparison with the standard impeller (b). Average diameter D_d is normalized with the inlet average diameter $D_{d,0}$. Three different flow conditions (surge to overflow) and liquid volume fractions (1% - 3%) are considered.

EROSION TESTING

The mechanism of erosion of components due to liquid presence has been described in detail by Khan et al. [8] and Li. et al. [9].
The main parameters that affect the erosion behaviour are:

- Material properties on the surfaces wet by the operating fluid;
- Size, density, velocity and impact angle of droplets.

The OEM has characterized different materials through dedicated tests, which main outcome is wear history as a function of particles' size and impingement velocity. These tests thus allow the manufacturer to properly select the materials and coatings which best suit the expected operating conditions.

The purpose of the liquid tolerant design is to modify the actual operating condition of impeller's material: this design is intended to reduce particles' size and impingement velocity through a dedicated aerodynamic design, thus allowing the material of impeller to operate with a less demanding condition if compared to a standard design. It is thus necessary a different test to understand if the liquid tolerant design really allows to operate with lower impact energy for liquid particles, as it is suggested by CFD analysis shown in previous paragraph.

Thus, in order to compare erosion behavior with the presence of liquid, the impellers (STD and WTI) were painted with three layers of soft painting: the inner layer was red, the middle layer was white, while the external layer, as presented in Fig. 3, was green. Periodic optical and boroscopic inspections were scheduled to assess the areas more exposed to erosion due to water and roughly estimate the erosion rate. After 1 hour of continuous running inlet conditions, the standard impeller has shown significant erosion of the first layer of paint, while almost no signs of erosion have been detected on the WTI (Fig. 4 and Fig. 5) confirming that, thanks to specific wet tolerant design, the WTI erosion resistance is higher than that of a standard impeller. Fig. 4 shows the status of the trailing edges of the standard impeller and of the WTI impeller after 1 hour of running; the paint layers on the pressure side of standard 3D impeller have been completely worn away, while WTI is showing a small eroded area on the blade edge. Fig. 55 shows the leading edge and the flowpath channels after 24 hours of continuous wet running. It is evident that inner layers of painting are visible for the standard impeller, as a consequence of the erosion of the green layer, while for the WTI the outer layer is still visible and only minor signs of erosion are detectable. It is worth to notice that the results from this test can only be considered as qualitative information; however it seems evident that the liquid tolerant design has met the target of improving the erosion resistance reducing the intensity of local erosion.

Fig. 3: Coated impeller before erosion test

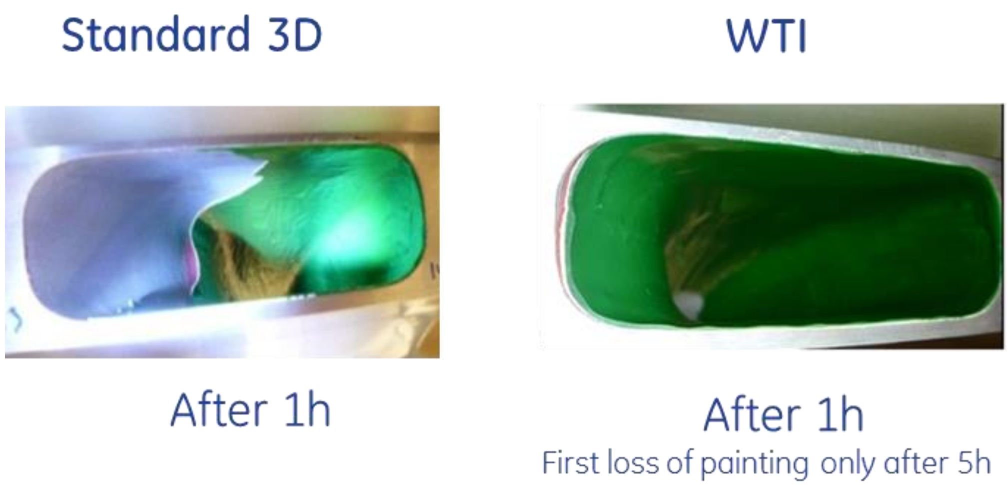


Fig. 4: Trailing edge erosion pattern

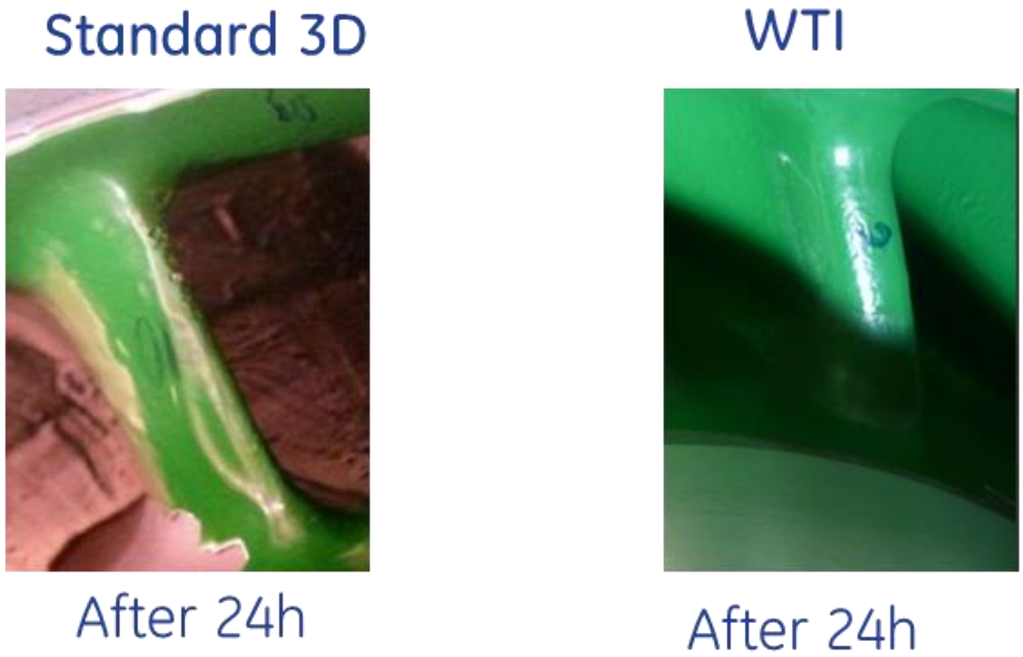


Fig. 5: Leading edge erosion pattern

TEST FACILITY

An advanced test facility has been developed in the last years by NTNU to identify the fundamental mechanisms that underpin the compression of gas with significant liquid content, highlighting also the behavior and capability of a total subsea compression system [10]. The rig has been upgraded with new features, such as Plexyglas windows on diffuser wall, aspresented in Fig. 6 (Ferrara et al. [14]). It is a single-stage centrifugal compressor, composed by the axial inlet impeller, vaneless diffuser and variable section discharge scroll. The test rig is designed to operate air and air-water mixtures in an open-loop configuration, with a maximum discharge pressure of 1.5bar, thus only large liquid/gas density ratios can be tested. Grüner et al. [15] have compared air/water performance data from the rig with corresponding results from a high-pressure hydrocarbon compressor test; the results have shown that low pressure air/water tests provide the same performance trends as high-pressure hydrocarbon tests, thus no generality is lost considering high liquid/gas density ratios. Moreover, previous experience from Hundseid et al. [1] has shown that a comparison of performance between different fluids and inlet conditions must be based on the corresponding GMF.

The compressor and the test rig have been instrumented following the schematics of Fig. 6, moreover, in order to characterize also the single components, the frontal diffuser diaphragm has been equipped with the instrumentation reported in Fig. 7. It is worth to notice that the measuring sections for stage performance evaluation (flange-to-flange) are located upstream the discharge control valve and upstream the impeller axial inlet.

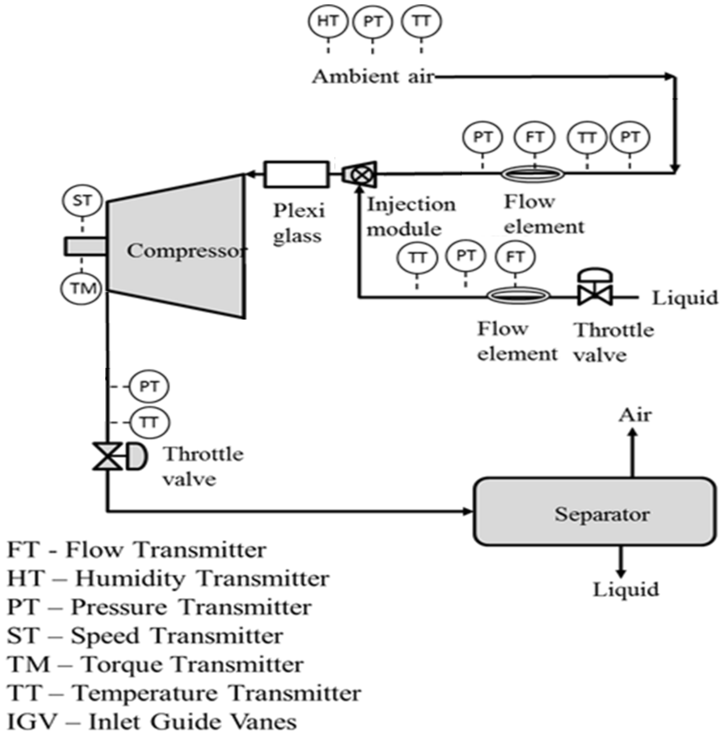


Fig. 6: Piping and instrumentation diagram of the test rig

Since multiphase flow measurements inside the compressor channels have proved to be extremely difficult, the visual observation of flow paths plays an important role in understanding wet compression mechanisms and performance. In the development of the present test rig, great attention has been paid in to increasing visual access to the flow path through the compressor, thus Plexiglas observation slots have been installed at the impeller inlet in the frontal diaphragm of the diffuser and at the volute section. However, due to the presence of those windows, the maximum discharge pressure, as stated previously, has to be restricted to 1.5 bar.

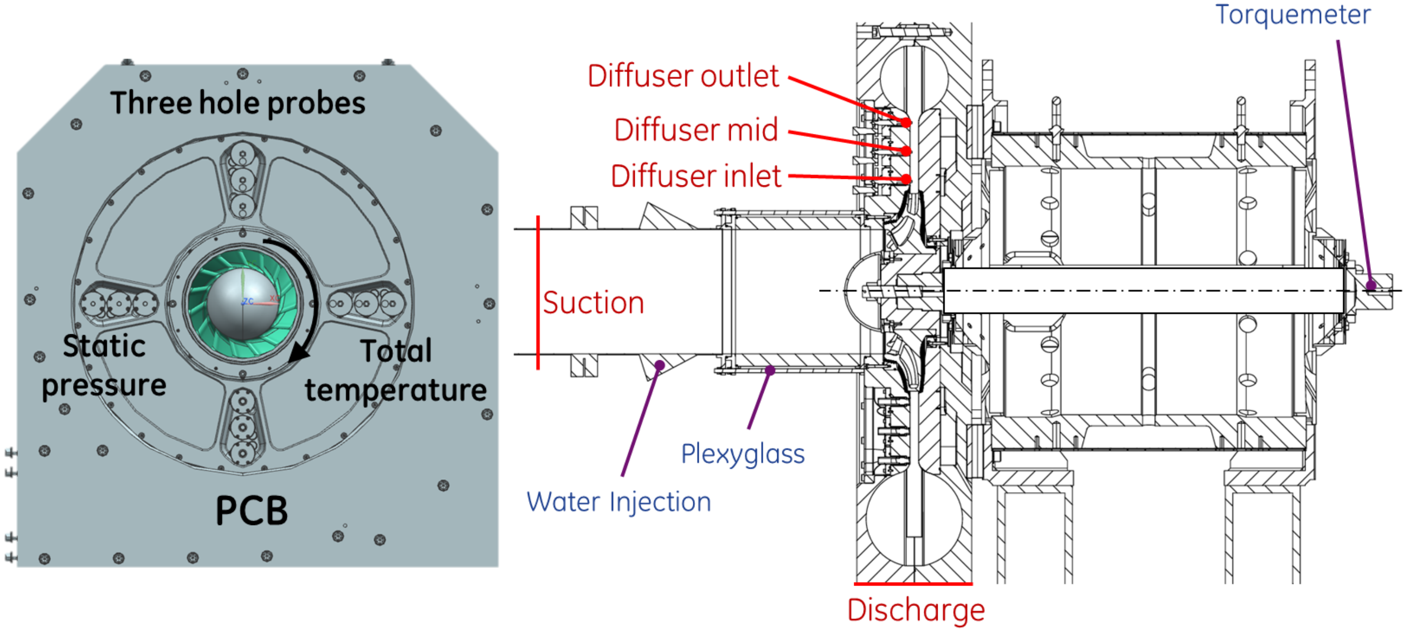


Fig. 7: Frontal diffuser section and instrumentation detail

The compressor has been operated through fast-acting, high-precision air inlet and discharge valves that made possible to test also transient flow regimes by throttling the flow. The compressor is driven by a high-speed electric motor that allows a maximum continuous speed equal to 11000 rpm.

The ambient air flows through an inlet pipe (internal diameter 250 mm) and mass flow is measured using an orifice plate. Water is injected upstream the impeller inlet through a water injection module consisting of 16 nozzles uniformly distributed around the pipe wall. Water flow is controlled by regulating supply pressure and the number of active nozzles.

The impeller is connected to the shaft of a bearing pedestal, which is used to prevent water penetrating the electric motor. Bearing pedestal and electric motor are connected through a torque meter, permitting accurate measurement of static and dynamic compressor power. Tab. 1 summarizes basic geometrical data for the impeller arrangement.

Impeller outlet diameter (D_2)	400mm
Diffuser width (b)	22mm
Diffuser ratio (D_3/D_2)	1.7
Inlet hub diameter	250mm
Outlet pipe diameter	200mm

Tab. 1: Main compressor dimensions

The gas flow at the diffuser exit is directly collected in a variable section scroll that discharges trough a 200 mm outlet pipe. When testing in multiphase conditions, a dedicated separation unit is located at the discharge pipe outlet. Test rig allowable operating range conditions are presented in Tab. 2.

Suction conditions	Atmospheric
Test fluids	Air/water
Air-flow range	0-3 kg/s
Water-flow range	0-5 kg/s
GVF range	99.93-100%
GMF range	40-100%

Tab. 2: Test rig operational range

The acquisition system for the test rig measurements is a PC-based platform, which permits synchronous sampling up to 20 kHZ. Performance testing is in accordance with ASME PTC-10. Tab. 3 presents measurements acquired during testing with related accuracy.

Instrument section	***Accuracy***	***Unit***
Ambient temperature	± 0.2	ºC
Ambient pressure	± 0.15	hPa
Relative humidity	± 1	%
Temperature flow element	± 0.15	ºC
Pressure diff. flow element	± 0.04	%
Dynamic pressure diffuser	0.14	mbar
Static pressure diffuser	± 0.002	bar
Total temperature diffuser	± 0.009	ºC
Three hole probe diffuser	0.11	%
Inlet pressure compressor	± 0.3	%
Inlet temperature compressor	> ± 0.1	ºC
Outlet pressure compressor	± 0.3	%
Outlet temperature compressor	> ± 0.1	ºC
Water flow meter	± 0.5	%
Shaft speed	± 5	rpm
Shaft torque	± 0.48	%

Tab. 3: Test rig instrumentation

PERFORMANCE TEST RESULTS

The performances of the liquid tolerant impeller have been assessed through a dedicated test campaign. Fig. 8 shows the experimental test rig: the Plexiglas window sections are visible on the dissembled diffuser and the water injection module can be seen in the foreground.

The performance curves in dry condition have been measured at two different rotational speeds corresponding to two different peripheral Mach numbers:

- 9000 rpm (Mu = 0.55)
- 11000 rpm (Mu = 0.66)

Where Mu is defined considering the speed of sound for the gas inlet conditions:

$$Mu = \frac{u_2}{a_{in}}$$

Five different values for liquid mass fraction have been selected for to characterize the WGC behavior:

- LMF = 0% (dry)
- LMF = 5%
- LMF = 10%
- LMF = 20%
- LMF = 30%

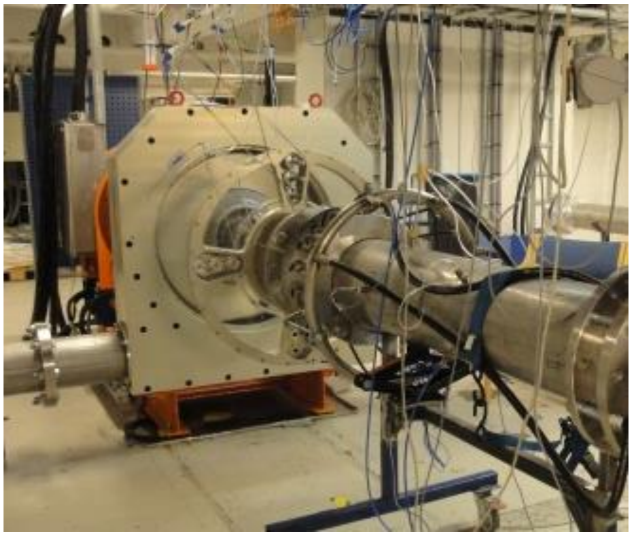


Fig. 8: Frontal view of test rig

Each speedline has been characterized through measurements for 6 different flowrates. The flow has been adjusted by progressively closing a throttle valve on the discharge pipe. Test points have been considered valid only if a steady state criterion based on the signal standard deviation was reached. Valid test points have been measured by averaging each signal on a time window equal to 60 seconds. To avoid measurements being affected by liquid inside the sensor pipe, a purge system has been set. Results shown in the present paper refer to the 11000 RPM speedline, since no significant variation has been detected modifying the rotational speed.

In Fig. 9: Non-dimensional stage pressure ratio at 11000 rpm and different values of LMF, the normalized flange to flange pressure ratio (PR/PR_{design}) is reported. The actual pressure ratio has been normalized by the dry pressure ratio at the design flow coefficient ($\phi/\phi^* = 1$). It is evident that the water injection has the direct effect of reducing the PR: increasing the LMF, the pressure ratio reduces accordingly. For low LMF conditions, the increased frictional losses due to liquid presence are probably mitigated by the cooling effect due to the presence of the liquid, as a result lower efficiency detriment is observed. The pressure ratio drop is thus almost negligible for LMF=5%, while it significantly increases moving towards LMF=30%.

The PR drop increases moving from minimum flow to the overflow: considering LMF=30% with respect to dry conditions, a PR drop of 2% and 6.5% is observed at partflow and overflow, respectively. These results are in apparent contrast with previous tests [2] where an increased slope of PR curve was observed with PR gain at partflow and PR drop at overflow. Test [2] has been executed considering liquid/gas density ratios typical of industrial applications, while this campaign has been performed with higher liquid/gas density ratio. As a result, only the effects connected to the increment of losses in multiphase flows can be detected.
Polytropic efficiency has been evaluated in the following way:

$$\eta_p = \frac{\Delta h_{pol}}{\Delta h_{eff\_mech}}$$

with

$$\Delta h_{eff\_mech} = \frac{W_{eff\_mech}}{\dot{m}_g + \dot{m}_l}$$

and

$$\Delta h_{pol} = LMF\ \frac{p_{out} - p_{in}}{\rho_l} + (1 - LMF)\ \Delta h_{pol\_g}$$

As stated above, the polytropic efficiency in wet conditions is based on a mechanical measurement (energy supplied to the compressor shaft; the subscript "eff_mech" stands for effective mechanical, to clarify that the value used for this evaluation is the power from torquemeter minus the mechanical losses due to shaft's bearings); using a mechanical measurement allows to rule out the inaccuracies of temperature measurement due to the presence of a liquid phase. It is worth to notice that the leading parameters that control the wet performance of a compressor are the liquid mass fraction and the liquid volume fraction [11].
Fig. 10: Non-dimensional stage polytropic efficiency at 11000 rpm and different values of LMF
reports the normalized polytropic efficiency in dry and wet conditions varying the LMF. Polytropic efficiency, $\eta_{pol,}$ is normalized with the polytropic efficiency, $\eta_{pol,design,}$ measured at the design flow coefficient ($\phi/\phi^* = 1$). As expected the main effect of the presence of water is a reduction of the compressor efficiency: the higher the LMF, the higher the efficiency drop. In this case the efficiency reduction dependence with flow rate is less evident and, from this first analysis, a clear trend is not detected. However, considering the highest LMF, efficiency drops from 45% to 57% are observed at partflow and overflow, respectively.
The normalized work coefficient, τ/τ_{design}, as a function of the liquid mass fraction is presented in Fig. 11. Similarly to the wet polytropic efficiency, to rule out the uncertainties on the temperature measurement in wet conditions, also the work coefficient calculation is based on a mechanical measurement:

$$\tau = \frac{\Delta h_{eff\_mech}}{u_2^2}$$

As expected the work coefficient increases with the LMF: due to the presence of the liquid an increased mechanical work is associated to higher power consumption; similar trends have been observed during tests at 9000 rpm. While for the LMF 5%-20% range the work coefficient increment is uniform and no variation of slope is observed; when larger LMFs are considered also a deviation of the slope is highlighted. One of the possible causes is that for large LMF, a significant quantity of liquid is expected to accumulate as a film on the impeller blade. As a result, it may be possible that the drops detaching from the impeller tip may tend to reduce the flow radial velocity component, yielding to a reduction of the work coefficient.

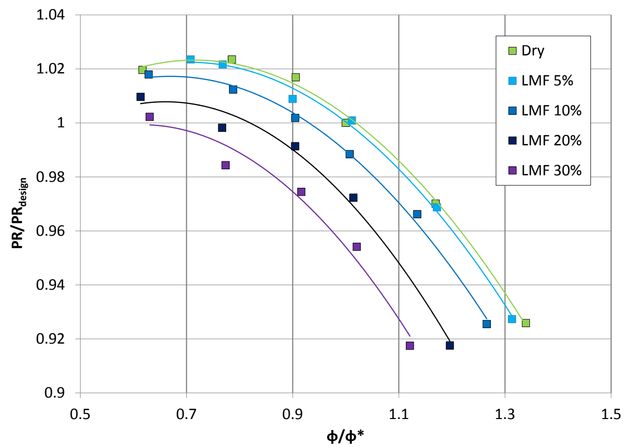


Fig. 9: Non-dimensional stage pressure ratio at 11000 rpm and different values of LMF

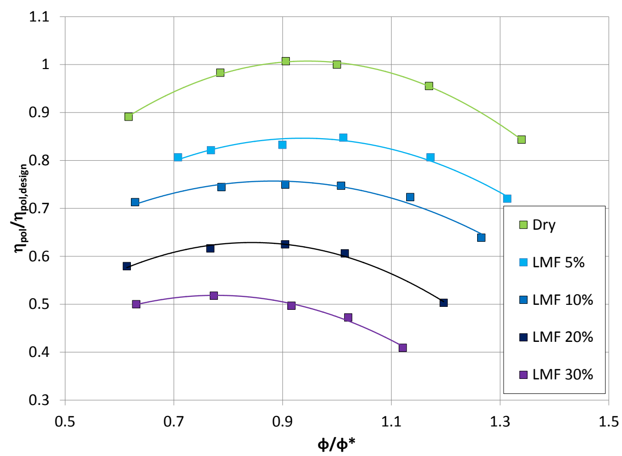


Fig. 10: Non-dimensional stage polytropic efficiency at 11000 rpm and different values of LMF

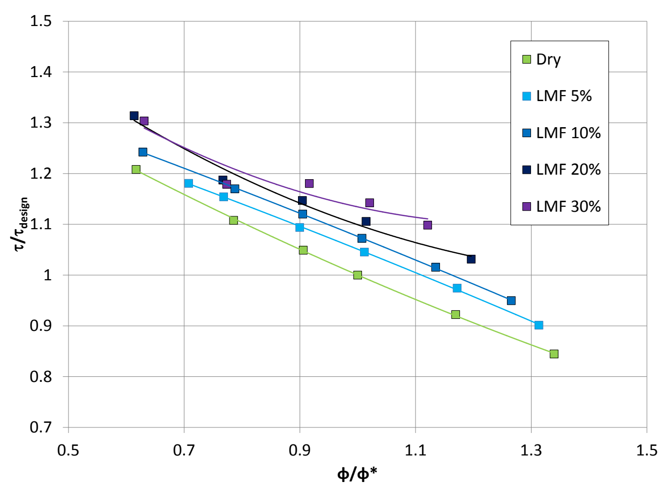


Fig. 11: Non-dimensional stage work coefficient, at 11000 rpm and different values of LMF

The increase in work coefficient with liquid mass fraction is also visible through Plexiglas windows: the following images show the wake from the three holes probe at diffuser inlet for different LMF. By comparing Fig 12 (a), which refers to a condition with low liquid mass fraction, with Fig. 12 (b), where LMF has been increased, it is possible to see that the wake of the probe (thus the trajectory of multiphase flow) becomes more tangential (the width of the wake also increases because of the shape of the probe, that results in a wider resistant area when velocity is more tangential). This is a direct evidence of the increased variation in tangential velocity component which means an increased work done on the processed fluid.

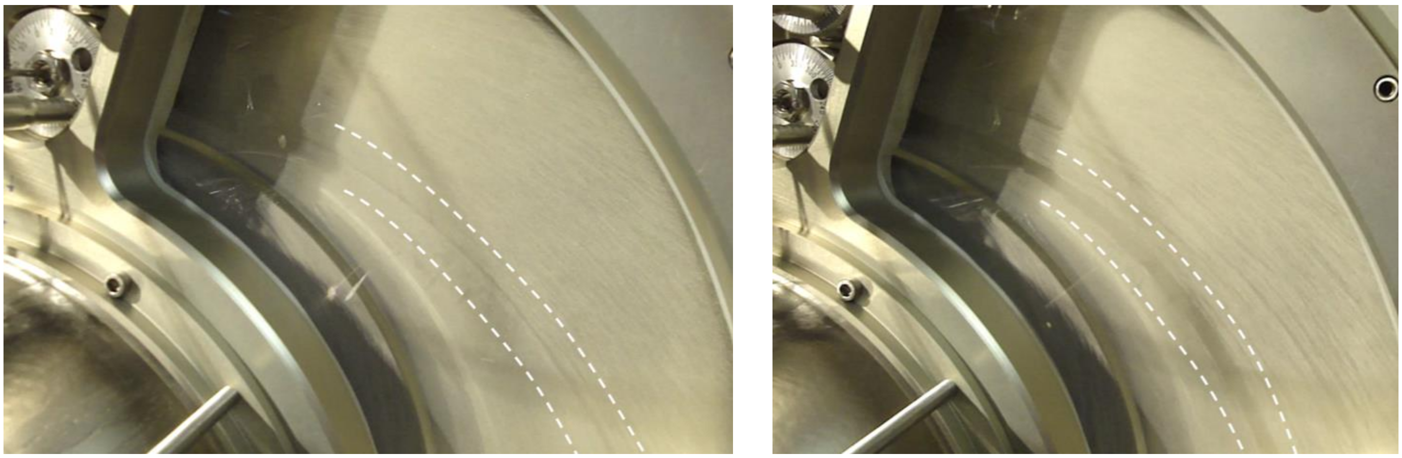


Fig. 12: Visualization of streamlines on diffuser for different values of liquid mass fraction; (a) low LMF, (b) high LMF.

OPERATING RANGE EVALUATION

The impeller operating range has been investigated varying the LMF.

Three dynamic pressure sensors have been installed in different radial positions of diffuser (see Fig. 7 for detail). Signals have been processed by fast Fourier transform analysis. The blade passing frequency (BPF) is calculated as

$$BPF = \frac{RPM}{60} \cdot n°_{impeller\_blades}$$

at 9000 rpm, the BPF is equal to 1950 Hz.

Generally unstable phenomena are detected when the pressure fluctuations cause subsynchronous frequencies greater than the blade passing frequency [13]. Typical stall range is included from 20% to 60% of the revolution speed frequency, while lower frequencies are representative of surge phenomenon.

The tests have been conducted with same procedure considered for standard design. The starting flow coefficient was selected to have a frequency spectrum which did not present any sub-synchronous phenomenon, in general 0.04. The gas throttle valve was closed, reducing the flow coefficient and keeping a constant liquid flow rate.

Fig. 13 to Fig. 15 report results regarding the power spectrum at 9000 rpm under dry and wet conditions. In Fig. 13, it should be noted that the subsynchronous term, at about 78 Hz, overtakes the blade passing frequency at φ equal to 0.036. Comparing the waterfall diagrams under dry and wet conditions for 5% of LMF, Fig. 14, the same frequency component is revealed with the presence of a liquid phase, the instability appears at lower φ (0.027), respect to dry test (φ = 0.036), consequently the liquid presence is preventing the stall inception and does not limit the machine operating range. This behavior is consistent with what has been observed for the standard design. Similar behavior is confirmed when increasing the water amount, that is the liquid mass fraction, stall tends to disappear, as shown in Fig. 15, where a really weak component is present for a frequency approximately equal to 48 Hz. In this condition, the predominant components have very low frequency and can be better related to a surge phenomenon.

Consequently, the results from standard impeller are confirmed: liquid does not restrict the operating range.

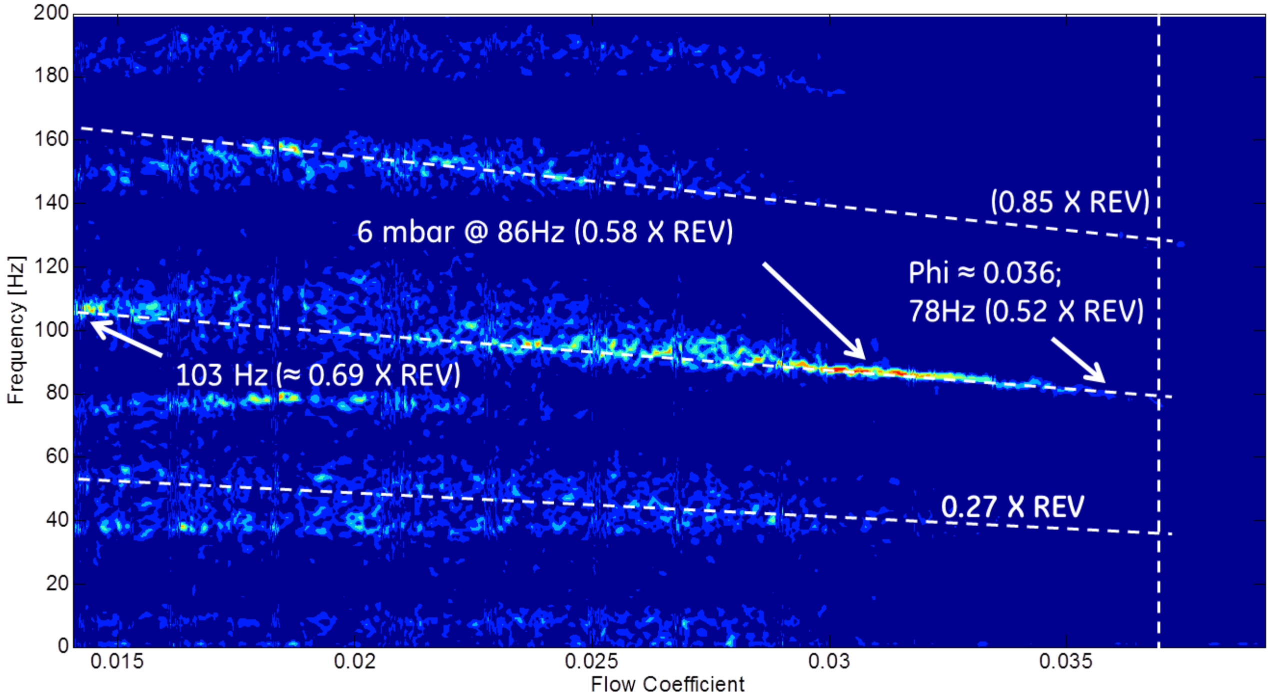


Fig. 13: Waterfall diagram, at 9000 rpm in dry conditions

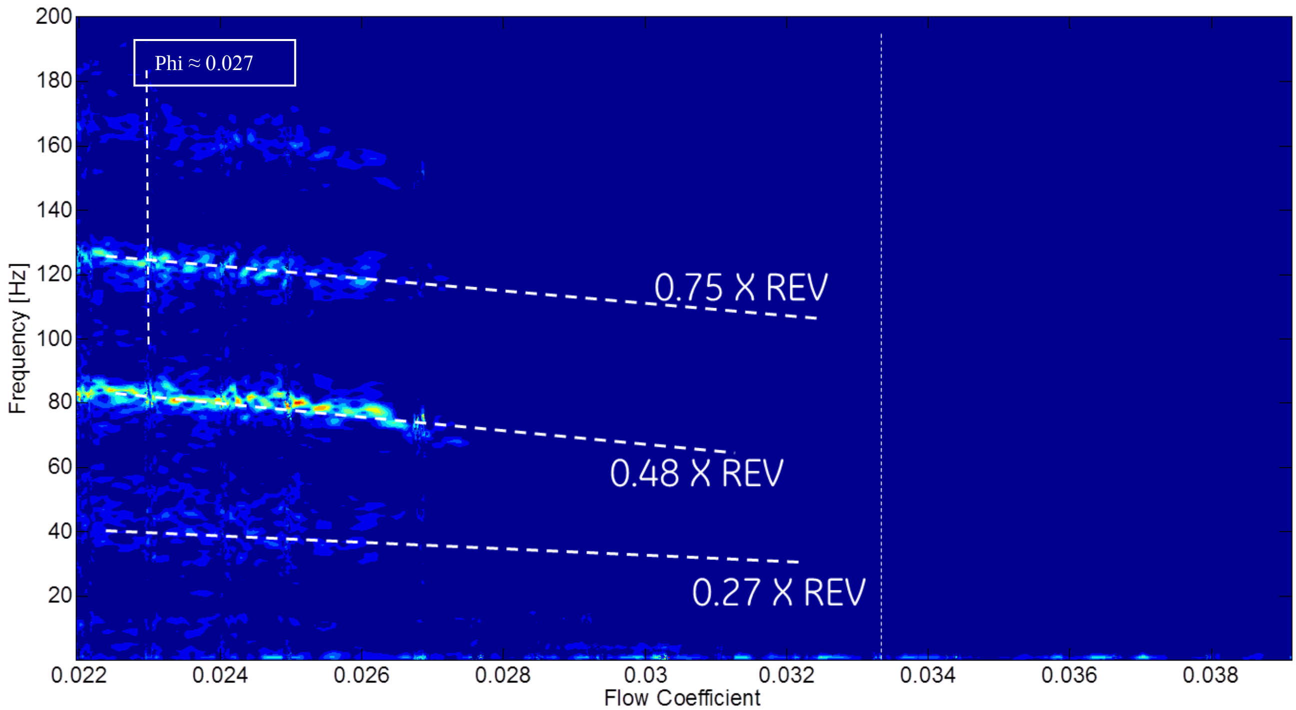


Fig. 14: Waterfall diagram, at 9000 rpm for 5% of LMF

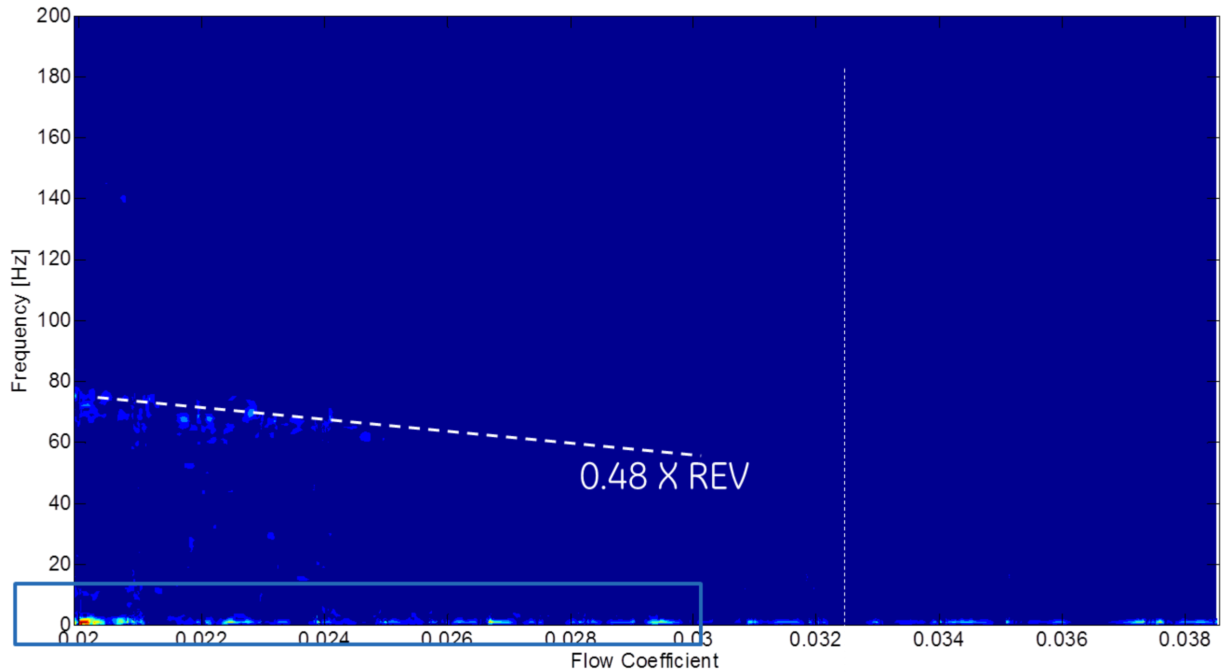


Fig. 15: Waterfall diagram, at 9000 rpm for 10% of LMF

OEM'S PREDICTIVE MODEL VALIDATION

Standard (dry) CC performance prediction models are known to be inadequate in the estimation of the main performance parameters of CC operated in wet gas conditions [2, 12] due to the significant modification of the aerodynamics and thermodynamics of the involved flow.

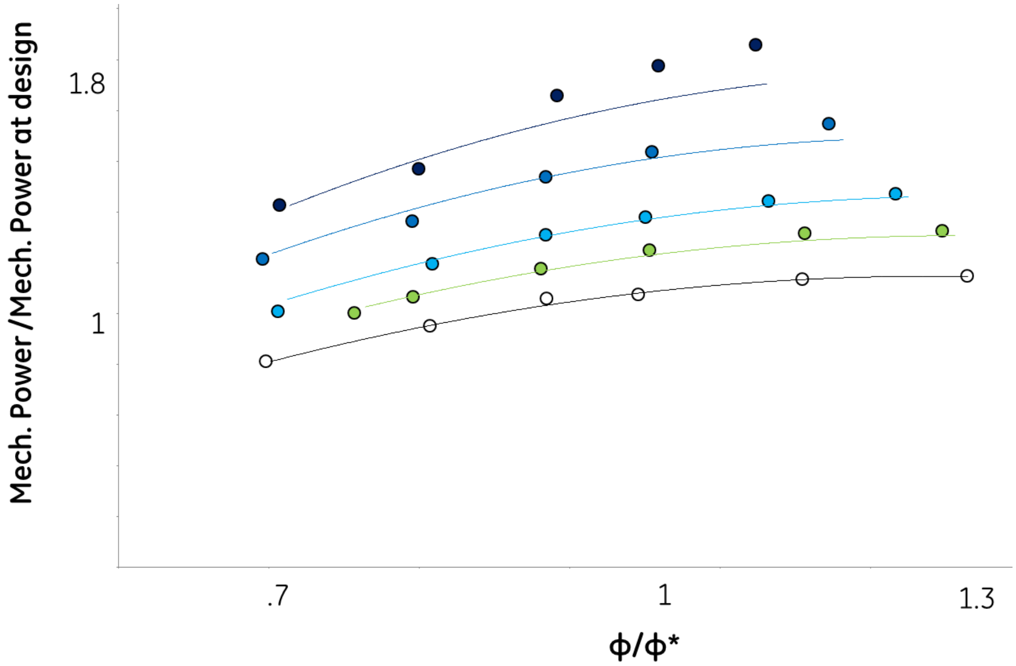
Ferrara et al. [14] classified the WGC performance prediction approaches based on the level of potential accuracy and complexity:

- multiphase CFD;
- 1D Models;
- dry-corrections model;

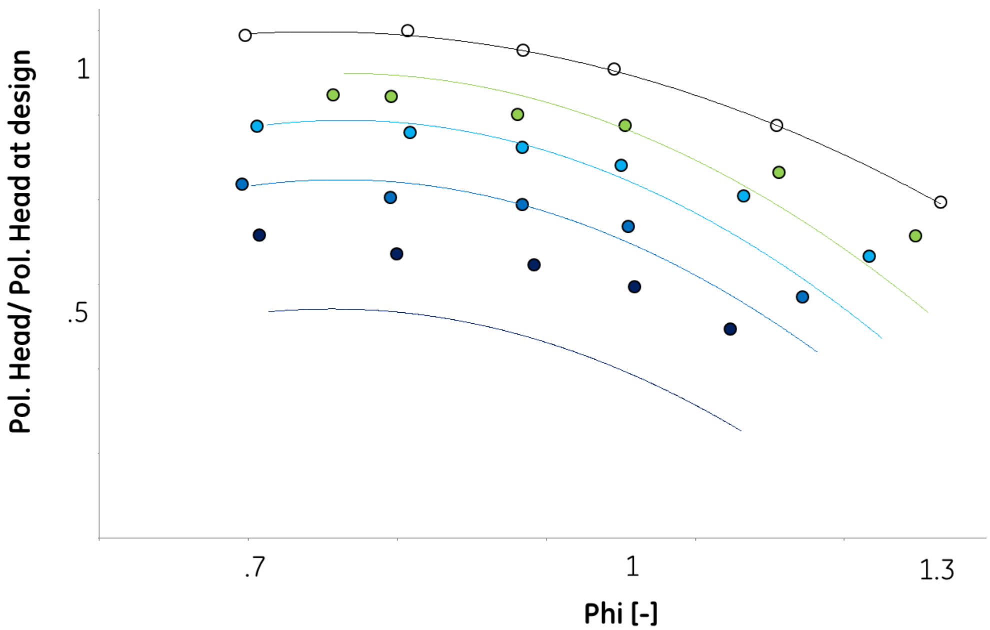
Multiphase CFD is the most detailed and complex approach: all the gas-liquid interactions can be modelled. However, due to the lack of validation with experiments, to the models complexity and to the associated computational cost, multiphase CFD predictions of WGC performance are still limited to qualitative evaluations. However, in particular areas of the multiphase analyses, this tool has shown promising capabilities for the detailed comparison among different designs and, for this reason, has been used for the design of the WTI.

1D models are usually physic- based: the balance equations for both phases have to be written and then integrated along the compressor flow path. Each compressor component (i.e. impeller, diffuser, etc…) is described using characteristic curves (determined from experiments or other predictive methods) using a unique coordinate that describes the movement through the compressor; an example of physical-based 1D model is described by Fabbrizzi et al. [3]. Due to the detailed description of the physical process, the inclusion of characteristic liquid/gas interactions is not trivial and thus, at the moment, they seem too complex for a practical application.

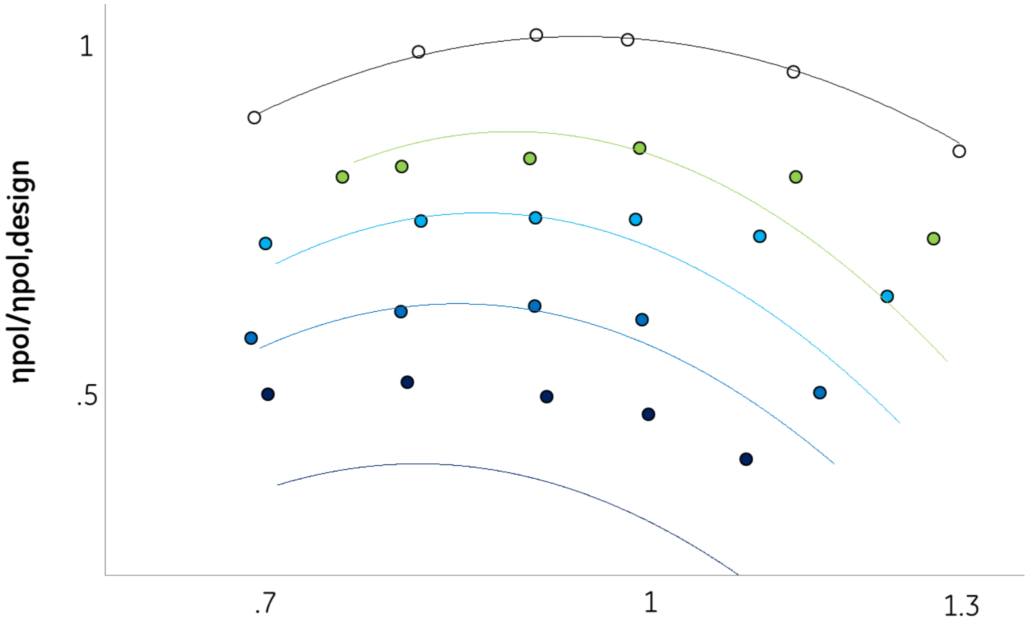
Based on the previous considerations, the OEM has chosen to develop a "black-box" model based on the corrections of the stage dry performance curves. The model has been calibrated using test results from different test arrangement: as a result, the stage dry curve and the geometric details of the stage are the only inputs for the model. The model has been presented by Ferrara et al. [14], and compared with test results from a similar performance test carried out on a standard impeller with 3D blades for natural gas applications. In the present paper, the same comparison is performed considering the test data from liquid tolerant impeller performance test; the comparison of test results with expected performance curves predicted by the tool is shown in Fig. 16.


Mech. Power /Mech. Power at design
1.8
1
.7
1
1.3
ϕ/ϕ*

(a)


Pol. Head/ Pol. Head at design
1
.5
.7
1
1.3
Phi [-]

(b)



(c)

Fig. 16: Comparison between experimental and expected results at 11000 rpm, in dry and wet conditions. (a) Absorbed power; (b) Polytropic head; (c) Polytropic Efficiency.

The prediction tool seems able to predict the increase in power related to the liquid injection: the work coefficient trend, shown in Fig. 16 (a), is thus well matched as well. The polytropic head trend is also well captured; the magnitude of variations seems accurate up to LMF=20%, then significant discrepancies are observed, as it can be seen in Fig. 16 (b). In any case the polytropic head prediction seems more accurate for flow coefficients below or equal to design flow; for higher liquid mass fraction or volume flowrate the tool tends to underestimate the polytropic head. Since the work coefficient is well captured, the possible reasons for the discrepancies have to be addressed to the polytropic efficiency trend: Fig. 16 (c) shows that the polytropic efficiency prediction shows a trend similar to that observed for the polytropic head. The main reason might be that the model has been calibrated considering the efficiency losses of standard impellers, while the performance decay for this impeller (that has been specifically designed to handle gas with liquid particles) is lower.

CONCLUSION

A test campaign has been conducted by the OEM for an innovative liquid tolerant impeller design to assess the effect of liquid in terms of erosion, performance change and impeller operating range.

The tests have demonstrated that the liquid tolerant features have proven to be effective: erosion for WTI is less evident if compared with a standard design and the erosion is distributed on hub and shroud, while for the standard impellers water concentrates on the hub region resulting in a faster erosion of this area.

The effect of liquid content on performance is evident and a clear trending has been observed: this allows for a good predictability of WTI performance in wet conditions, allowing to correctly predicting performance of multistage compressors which will include this novel design.

Performance curves become steeper with liquid content, resulting in an increased head rise to surge; considering also the absence of peculiar dynamic phenomena (dynamic behaviour is fully coherent with standard design, presented in [14]), the impeller can be operated both in dry and wet conditions in the same flow region of standard impellers.

As a general conclusion, the test results show that through the adoption of a specific liquid tolerant design, the compression of gas containing non-negligible amounts of liquid becomes a viable approach to reduce size and complexity of liquid-gas separators, thus reducing the overall cost of the compression station.

NOMENCLATURE

a	Speed of sound [m/s]
b	Diffuser width [mm]
BPF	Blade Passing Frequency [Hz]
CC	Centrifugal compressors
D_2	Impeller outlet diameter [mm]
D_3	Diffuser diameter [mm]
FFT	Fast Fourier Transform
FT	Flow Transmitter
GMF	Gas Mass Fraction [-]
GVF	Gas Volumetric Fraction [-]
h	Enthalpy [kJ/kg]
LMF	Liquid Mass Fraction [-]
LVF	Liquid Volume Fraction [-]
$\dot{m}$	Mass flow [kg/s]
Mu	Peripheral Mach Number [-]
p	Pressure [Pa]
PR	Pressure Ratio [-]
PT	Pressure Transmitter
r	Density ratio [-]
RPM	Speed [rpm]
STD	Standard Impeller
TT	Temperature Transmitter
u_2	Impeller Peripheral Speed [m/s]
W_{mech}	Mechanical power [W]
WGC	Wet Gas Compression
WTI	Wet tolerant Impeller (Liquid tolerant Impeller)
ϱ	Density [kg/m^3]
η	Polytropic Efficiency [-]
τ	Work Coefficient [-]
φ	Flow Coefficient [-]
Subscripts	
eff_mech	effective mechanical
g	gas
in	inlet
l	liquid
out	outlet
pol	polytropic

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