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SURGE EXPLORATION TESTS AND SECOND QUADRANT CHARACTERISTIC DYNAMIC MODELING ON HIGH PRESSURE RATIO COMPRESSOR (HPRC) PROTOTYPE

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Mr. Pelella graduated with honor in Mechanical Engineering at University of Naples, Italy in 1997, he has authored and coauthored 8 papers in compressor field and he presently holds 10+ patents.

ABSTRACT

High Pressure Ratio Compressor (HPRC) technology is based on an innovative architecture that combines unshrouded and shrouded impellers on a single high-speed shaft to achieve pressure ratios and efficiency levels higher than other available technologies. In the final step of the product development and validation, a test campaign has been carried out with the aim to test the performance and explore the operability of the new machine.
Surge exploration tests have been also performed allowing an evaluation of the transient behavior and the mechanical robustness of the compressor even during a critical event such as Surge.
Compressor Surge has been analyzed under different conditions, forcing the operating point to move beyond the performance envelope in full speed, at low and high pressure levels, and during emergency shut-downs. Thanks to the complex arrangement of the gas loops and several valves used to recycle back the compressed gas, different levels of Surge intensity were induced upon emergency shut-down.
The explorations of a fast event like Surge called for special instrumentations, configured to acquire process data both in direct and reverse flow conditions in the most reliable way, with a high frequency sampling for an oil and gas environment.
The result of this work is a breakthrough for the tuning of a centrifugal compressor model – not only HPRC ones – to be used for dynamic simulations and prediction of compressor dynamics during Surge events (like Surge cycle frequency and absorbed torque during Second Quadrant operation) in a more reliable and robust way.
Surge exploration tests results analysis, in terms of vibrations and thrust loads, together with development of a compressor enhanced dynamic model, allowed a change from a Surge acceptance criterion - based on the time spent on the left of the Surge Limit Line during an Emergency Shutdown event - to a more physics based criterion - based on the acceptable number of Surge cycles, thus increasing selection optimization of additional protections, such as hot/cold gas bypass valves.

1 INTRODUCTION

HPRC compressor prototype has been developed as a combination of stacked unshrouded and shrouded impellers on a single high-speed shaft to reduce the number of stages (up to a factor 2) compared to a traditional compressor. HPRC allows pressure ratios and efficiency levels higher than other available technologies, reducing compressor string footprint and weight.
A test campaign on the HPRC prototype has been carried out to evaluate compressor performance, aeromechanical and rotodynamic behavior both in steady state and upon the most critical scenarios, such as compressor Emergency Shut Down (ESD) and Start-Up [1]. Figure 1 is a depiction of the test bed set for test campaign activity.

Figure 1. HPRC Test bed arrangement at the BHGE Facility in Florence (Italy)

The HPRC prototype is a 3 sections compressor, arranged in line with two, three and two impellers respectively. The first two section impellers are of the unshrouded type.
As the final step of the product development and validation, a test campaign has been carried out to test the behavior of the compressor prototype beyond the First Quadrant envelope, focusing on the behavior at Surge. This activity allowed evaluating the mechanical robustness of the compressor upon a critical event such as deep Surge conditions (Second Quadrant Operation).
The purpose of Surge Exploration Tests was also to collect thermodynamic parameters beyond the First Quadrant compressor

envelope. In particular, Second Quadrant operation and periodic Surge cycles in which the compressor experiences a series of flow reversal and recovery, was deeply investigated in order to develop a reliable model - capable of predicting compressor behavior upon Surge.
The paper is organized as follows: in section 2, the HPRC test bench gas loop and the instrumentation used are presented; section 3 describes the HPRC test campaign; in section 4 vibrations, axial displacements and thrust loads recorded during two runs of the HPRC test campaign are analyzed; section 5.1 describes the results of a Model Test performed on a single stage compressor, used to characterize the second quadrant map branch and how it has been adapted to the HPRC machine; in section 5.2 an overall gas loop dynamic simulation model is presented; section 5.3 deals with data validation between simulation results and test data; and finally, conclusions are given in section 6.

2 GAS LOOP ARRANGEMENT AND INSTRUMENTATION

The test loop arrangement consisted of three independent closed gas loops laying on each HPRC section, equipped with recycle solenoid valves (ASVs) and a cooler at the suction. Bypass valves (PCV12 and PCV23, ball valve type) were installed to reproduce serial operation. Such a configuration (Figure 2) provided the maximum flexibility to test the prototype in several operational arrangements, allowing also for the decoupling of the three phases, as well as to study their mutual interactions during a Surge event. The compressor prototype was coupled with a variable speed electric motor driver with 10 MW maximum available power.

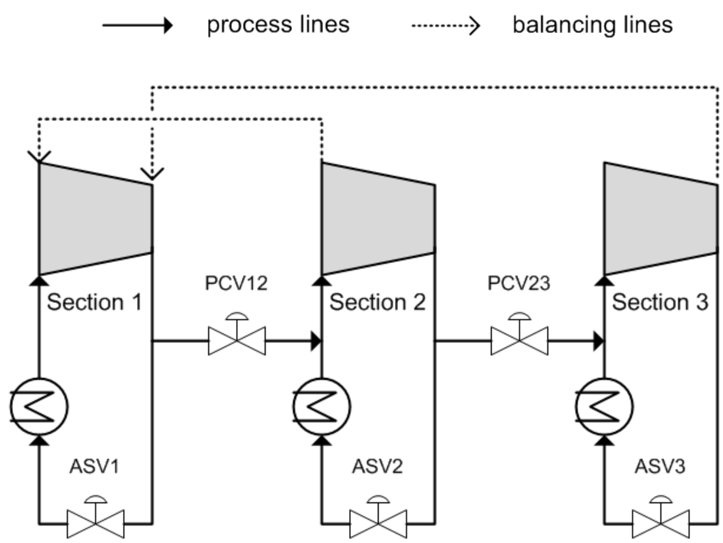

Figure 2. Testing loop arrangement

To evaluate Surge both from the thermodynamic and mechanical standpoint, the test bed was equipped with dedicated instrumentation aimed to properly collect data in direct and reverse flow operation conditions, also with suitable acquisition frequency for Surge phenomena (10Hz for flows and pressures and 1Hz for temperatures). Four dynamic pressure transducers and four thermocouples were installed on the suction and discharge of each compressor section; flow rate measurement was performed by differential pressure transducers installed both on suction and discharge, with 2% of accuracy on design values. The ASME orifice plates on each discharge was installed reversed to acquire reverse flow across the HPRC sections. Radial vibrations were measured through proximity probes located on both shaft ends with a 2000 Hz frequency response bandwidth. Axial displacements and axial thrust acting on the rotor were respectively measured through proximity probes and load cells installed on the thrust bearing.

3 SURGE EXPLORATION TESTS CAMPAIGN

Compressor Surge has been analyzed under different operating conditions, forcing the operating point to move beyond the performance envelope at full speed and during emergency shut-downs, at low and medium gas loop pressures. Compressor control algorithm was modified to induce different levels of Surge intensity upon Emergency Shut Down through an instantaneous (as per standard procedure), delayed and frozen recycle valves opening action.
Compressor Surge at full speed was induced on a specific compressor section, by slowly closing the relevant ASV to increase the pressure ratio and reducing the flow until the operating point moves beyond compressor envelope Surge limit. In order to prevent any mutual effects upon Surge, the two other compressor sections were kept operating far from Surge limit, in the middle of the compressor envelope, close to the design point. In this way, Surge cycles on each compressor section were evaluated.
Before and after each Surge test, performance at a reference point condition was evaluated in order to capture any potential

degradation due to Surge phenomena detrimental effects on the machine. Compressor performance at reference point was always met, again demonstrating that no detrimental effect due to Surge on the compressor mechanical integrity was experienced.
The present work refers to a partially pressurized compressor loop, having compressor suction pressure at 5 barA and discharge at 60 barA.
A compressor mechanical analysis upon Surge event has been performed leveraging on data relevant to test performed on section 1 at full speed and on data relevant to ESD test with all recycle valves kept at their initial opening position.
Data validation with the dynamic model results has been then executed on the former two tests data and on data relevant to tests performed on section 2 and 3 at full speed.

4 MECHANICAL TEST DATA ANALYSIS

Mechanical behavior has been analyzed in terms of radial vibrations, axial displacements and thrust load variations. Figure 3 reports compressor speed and radial vibrations evaluated through proximity probes located on Non-Driver End compressor side. The red line depicts the direct radial vibration, including all frequency spectrum contributions; instead, the green one is the radial vibration at the revolution frequency (hereafter called in this study "1xRev"). Both trends have been normalized with respect to the value of the direct radial vibration at the initial steady state condition. It should be noted that direct radial vibration is reported as Peak-Peak, while 1xRev is 0-Peak. In the initial steady state, these contributions were identical, an indication that radial vibrations were mainly due to rotor residual imbalance. Alternatively, during Surge, non-synchronous vibrations were excited so that direct radial exceeded 1xRev vibrations: according to Figure 3, the former reached about 2.2 and the latterly 1.8 (having both values normalized and reported according to Peak to Peak scale). Therefore, Surge cycles were able to excite sub-synchronous vibrations for an amount close to 0.4. It should be noted that the maximum allowable threshold (1.7) was overcome without triggering trip. In fact, in order to disregard spurious data, a signal continuously above the threshold for at least 0.3 sec was required to initiate the ESD.
To detect the main contribution frequency to vibrations spectrum during Surge, Waterfall and spectrum plots were also developed for radial vibrations probes. As done for the other variables, vibration frequency has been normalized in the following figures with respect to the value at initial steady state condition. Figure 4 highlights that rotor Surge response was evident in the sub-synchronous range of 1xRev frequency and only during the flow inversion (from positive to negative) timeframe. A forced response on first rotor mode is present within this range (at about 0.25) corresponding to the maximum spectrum contribution (red box in Figure 4).
Also, the compressor speed variation was deemed a key parameter for compressor behavior upon Surge as it is an indication of the absorbed torque variation during the transient state. As the operating point moved in the Second Quadrant operation, the speed increased as an effect of a lower absorbed torque, reaching a maximum value while approaching the zero flow condition. Then, when operating point stepped back into First Quadrant, the absorbed torque started increasing, leading to a speed drop till its minimum value. It should be noted that speed variations were also affected by the speed controller, which was acting on electric motor torque trying to keep the speed at its set point. However, this effect has to be considered negligible on speed variation with respect to the compressor absorbed torque fluctuations due to Surge.

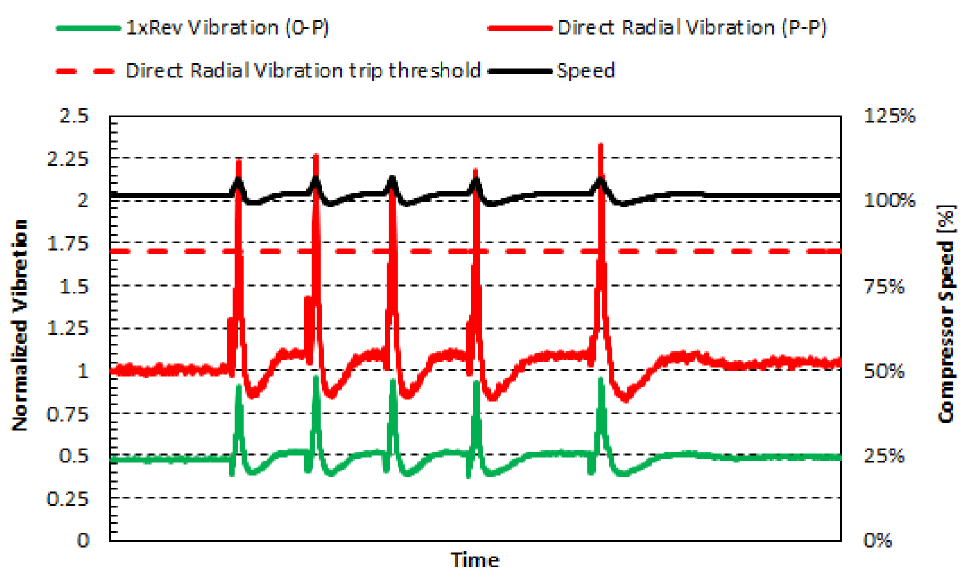

Figure 3. First Section Surge Cycles at full speed – Radial Direct and 1xRev vibration on NDE trends and compressor speed

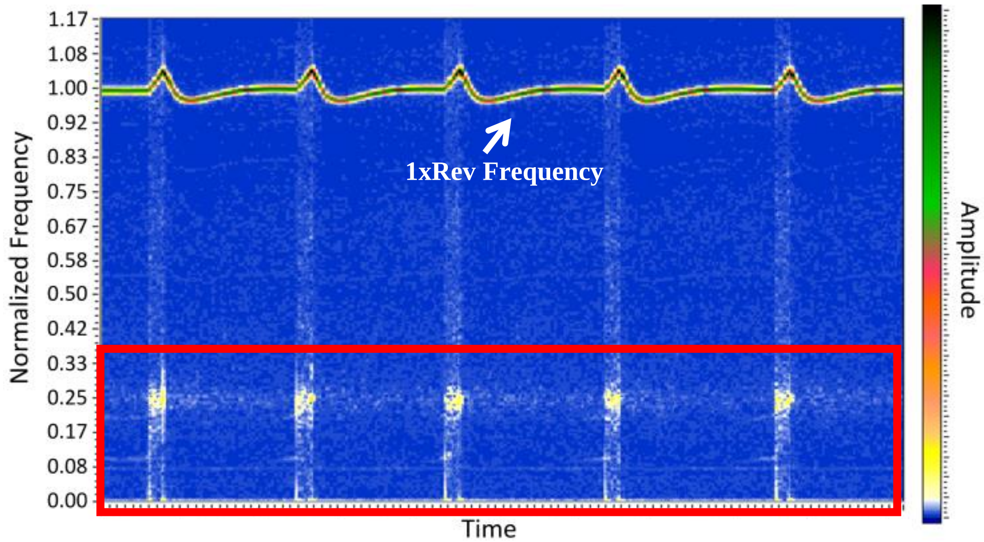

Figure 4. First Section Surge Cycles at full speed – Waterfall for Radial Direct Vibration on NDE

The compressor Surge behavior was also tested upon ESD, forcing all three sections to moves in reverse flow condition (Second Quadrant) by freezing the ASVs opening position so as to hinder the suction and discharge pressure equalization during coast down. Before initiating ESD, the three compressor Sections were brought close to the Surge Limit Line (SLL) by closing ASVs. As soon as the ESD was triggered, speed started decreasing and reverse flows were detected in all three sections. As soon as the reverse flow condition across the compressor occurred, direct radial vibrations showed a peak amplitude of 1.5 (126% of initial operation) as shown in Figure 5. The same increment was not present on the 1xRev component trend, an indication of a wider range of frequencies that was contributing to the overall vibration upon Surge.

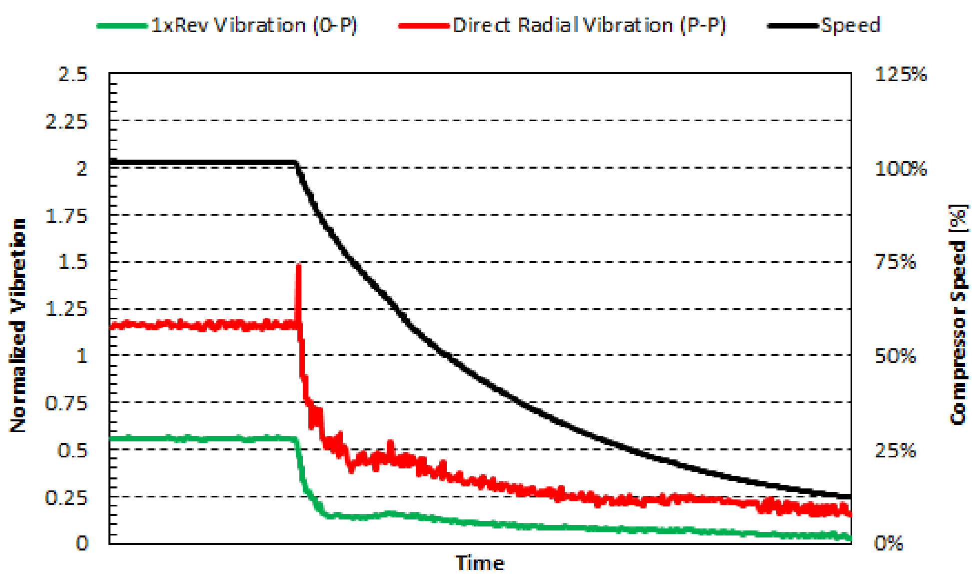

Figure 5. Surge upon ESD – Radial Direct and 1xRev vibration trends on NDE and compressor speed

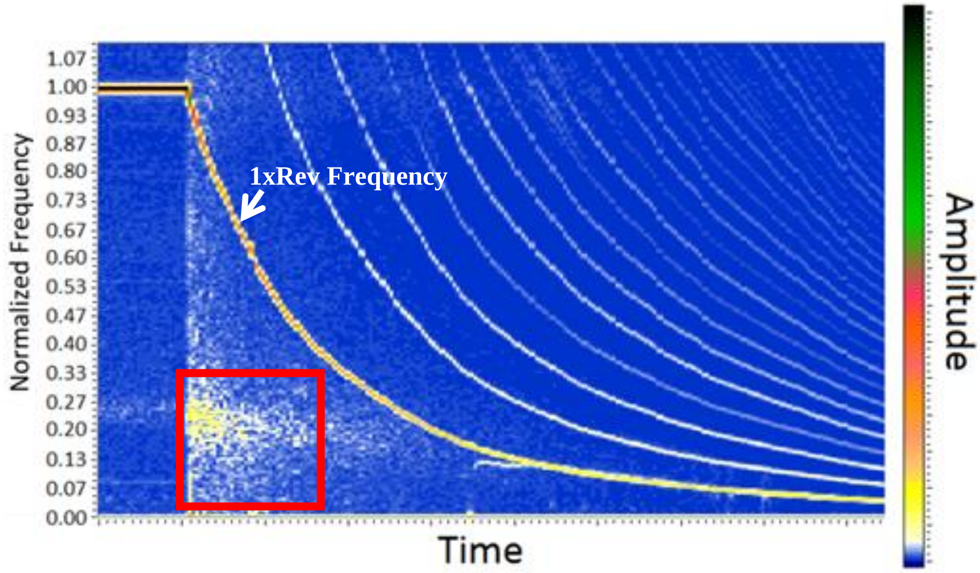

Figure 6. Surge upon ESD – Waterfall for Radial Direct Vibration on NDE

The waterfall plot in Figure 6 provides a further confirmation that Surge is a “broadband” transient phenomenon. Vibration amplitudes decayed without any further peaks during the compressor coast down, an indication that reverse flow condition is an acceptable operation from a mechanical standpoint. With regards to the entire vibrations spectrum, the same conclusions relevant to the Surge tests at full speed can be drawn since sub-synchronous vibrations of a minor magnitude were observed (red box in Figure 6).
For the sake of completeness, axial displacements and axial thrust load were also evaluated, as done in [6]. Figure 7 and Figure 8 show these variables normalized with respect to the value at their own initial steady state condition. In case of Surge under operating conditions, axial displacements and thrust load had a peak to 1.24 and 2.5 respectively (Figure 7), without exceeding the thrust bearing rated capacity.

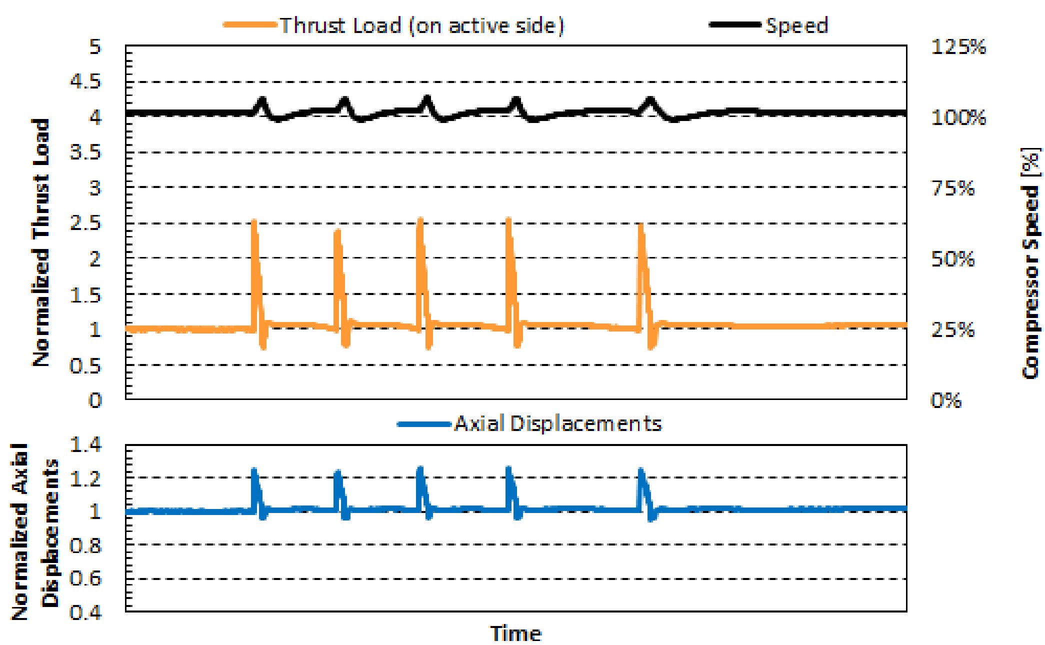

Figure 7. Axial Displacements and Thrust Load on active side – Surge Cycles at full speed

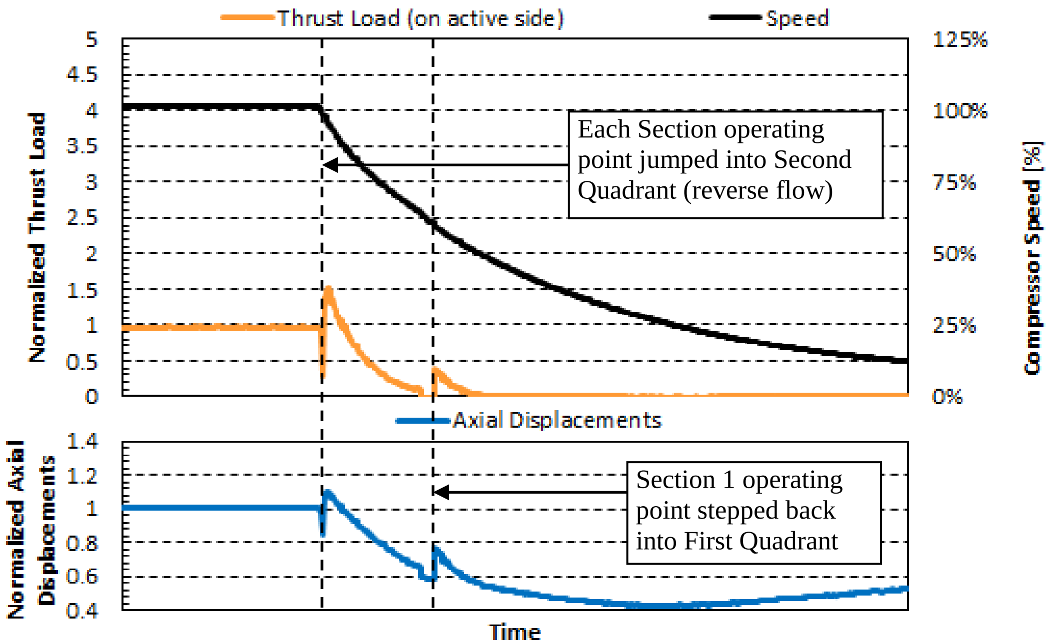

Figure 8. Axial Displacements and Thrust Load on active side – Surge upon ESD

On Surge during the ESD test, both axial displacements and thrust load decayed during the compressor coast down apart from two minor increments experienced as soon as the operating point moved in the Second Quadrant region and when first section compressor stepped back from Second to First Quadrant (see Figure 8).
An important outcome of the Surge exploration tests is that a reverse flow condition (aka Second Quadrant region) can be withstood by the compressor, even for a prolonged period of operation (such as the entire compressor coastdown duration), since vibrations and axial thrust trends decayed smoothly. However, the Surge phenomenon associated with forward and backward flow, not only across the compressor but even to compression system equipment (like scrubbers, coolers, valves, piping and relevant anchors, etc.), requires a threshold on the number of Surge cycles to avoid undesirable upsets due to pressure waves and vibrations.
A new Surge acceptance criterion has been developed accounting for a maximum of three Surge cycles instead of operating time beyond the compressor Surge Limit Line (SLL) as per the current commonly used design practice.
The new approach to evaluate compressor Surge is still conservative, but at the same time it is expected to reduce or prevent undue complexity such as installation of additional Surge protection devices (like Hot Gas Bypass Valve or Cold Gas Bypass Valves) or hardware modifications.

5 SURGE MODEL AND DATA MATCH

The purpose of Surge Exploration Tests was also to collect thermodynamic data relevant to the Second Quadrant operation to be compared with simulation results.
For this scope, compressor behavior has been simulated with a custom model based on a modified Moore-Greitzer [5] compressor dimensionless characteristic curve (cubic shape). The Second Quadrant map branch has been characterized with an ad-hoc Model Test carried out on a single stage compressor [3]. The model has been then adapted on the HPRC based on Surge Exploration Tests, the subject of this paper. Compressor map development in the second quadrant and HPRC Surge tests data match are described in the following sections.

5.1 Second Quadrant compressor model

In the Model Test arrangement (which is more deeply investigated in [3] and [4]) a booster compressor was connected in series with the tested stage forcing the flow to be stable in reverse flow condition (Figure 9).

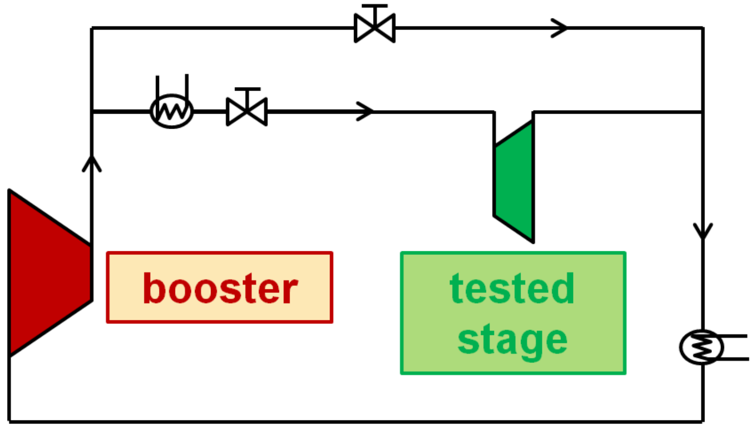

Figure 9. Model Test Arrangement

The trends of pressure ratio measured in the second quadrant are reported as a function of flow coefficient in Figure 10.

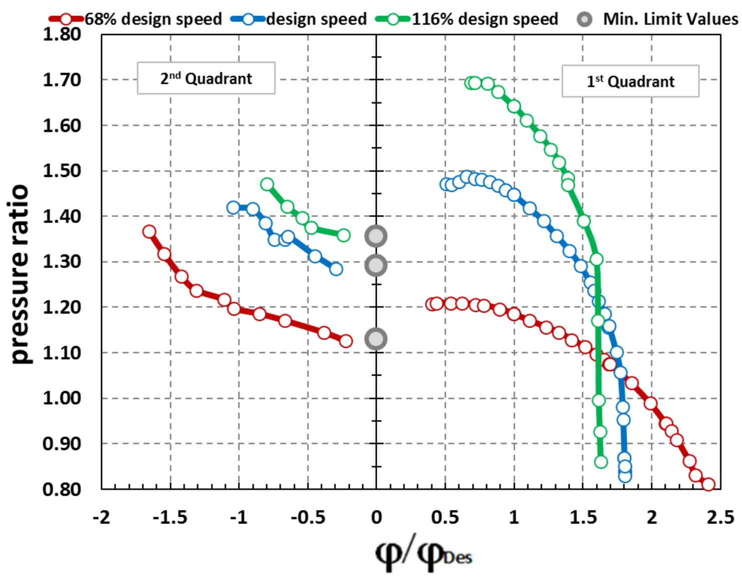

Figure 10. Measured Performance in 1st and 2nd Quadrant

$$\varphi = \frac{4\dot{m}}{\rho_{in}\pi D_2^2 u_2} \qquad \textbf{Eq. 1}$$

In the first quadrant (positive flow), compressor curves have the well-known shape: decreasing the flow, the pressure ratio that the compressor is able to deliver increases until it reaches the surge limit. In the second quadrant (negative flow), reducing the amount of reverse flow to zero, pressure ratio tends to a minimum limit value which corresponds to centrifugal forces at the imposed rotational speed.

Compressor curve trends observed are in accordance with the cubic shaped compressor dimensionless characteristic curve of Moore-Greitzer model (Figure 11) which has been taken, together with the equation of motion (Equation (2)) as reference for the development of the compressor dynamic model.

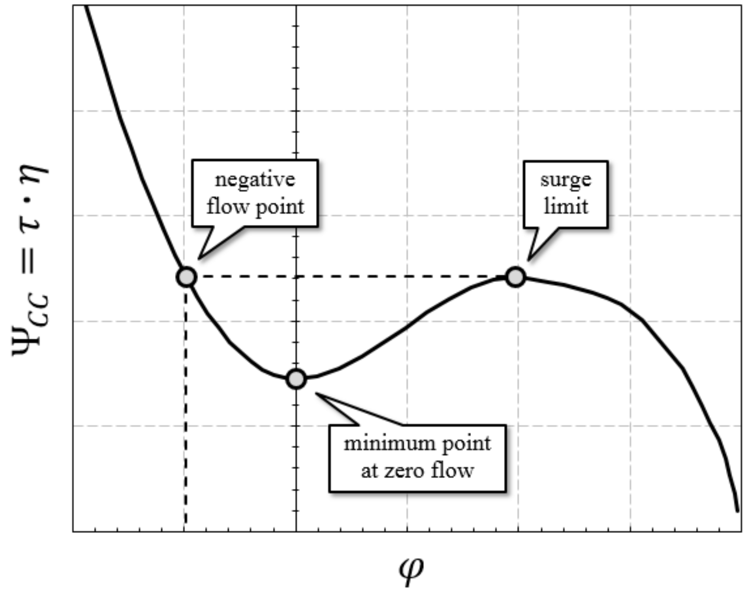

Figure 11. Dimensionless Map in 1st and 2nd Quadrant

$$\begin{cases} \eta_{pol} = f(\varphi) \\ \Psi_{CC} = \frac{H_{pol,CC}}{u_2^2} = f(\varphi) \\ \mathrm{H}_{pol,P} = f(T_{in}, P_{in}, P_{out}) \\ \frac{\partial \varphi}{\partial t} \propto \left(\mathrm{H}_{pol,CC} - H_{pol,P}\right) \end{cases} \qquad \textbf{Eq. 2}$$

In Equation (2), $H_{pol,P}$ is the polytropic head calculated from the process gas thermodynamic properties at the inlet and outlet section of the compressor, $\mathrm{H}_{pol,CC}$ is the polytropic head of the compressor given by the compressor map and η_{pol} is the compressor polytropic efficiency also given by the compressor map (for more details on equation of motion refer to [5]).
The three stage curves of the HPRC have been tuned by matching dynamic simulation results with maximum and minimum direct/reverse flows measured during the Surge tests. Fine tuning parameters used are the coordinates of minimum point of the dimensionless polytropic head Ψ_{CC} at zero flow and coordinates of negative flow point at which the dimensionless polytropic head is equal to the one at surge limit (see Figure 11).

5.2 Dynamic Simulation Model

With reference to the actual gas loop (Figure 2), a HPRC compressor loop dynamic model has been built with a detailed characterization of all the testing equipment such as valves, piping, coolers, etc.
The software used was Aspen HYSYS® configured with the Peng-Robinson equation-of-state and Lee-Kesler option for the calculation of enthalpies. The standard Aspen HYSYS® compressor element has been replaced by a custom element able to correctly simulate the behavior of the compressor in the Second Quadrant according to the model described in the previous section.
The compressor speed model used is given by a torque balance equation (Equation (3)) applied to the train, in which the inertia of all the elements composing the train (electric motor driver, gear box, couplings and compressor) and the speed controller acting on electric motor torque have been considered. Such a model also allowed simulating the speed fluctuation due to the variations of both absorbed torque by the compressor and available torque from the electric motor.

$$\frac{\partial \omega}{\partial t} = \frac{\frac{T_{EM}}{GR} - T_{CC} - f \cdot \omega}{I_{TOT}} \qquad \textbf{Eq. 3}$$

In the dynamic model, the speed controller has been tuned to have its action on the Electric Motor Torque (T_{EM}) be able to reproduce the speed variation experienced during full speed tests. However, it should be noted that the main parameter affecting the speed variation in Equation (3) is the compressor absorbed torque (T_{CC}), which is calculated by the custom compressor element using the aforementioned curves.
The HPRC compressor loop dynamic model has been first validated matching initial steady state condition of each test run in terms of

suction and discharge pressure, temperature, mass and volumetric flow of each compressor stage, and then used to replicate data obtained in the tests where Surge events were induced on compressor sections 1, 2 and 3 (separately) at full speed and upon ESD with all recycle valves kept at their initial opening position. Data validation results are presented in the next section.

5.3 Simulation data validation

Data validation study results are shown in Figure 12 to 15 for the four runs analyzed. Variables under investigation in this study are: mass flows, suction and discharge pressure in each compressor section and train speed. Compressor suction and discharge temperatures have not been included in this analysis as the temperature probe response time and thermal inertia of the system does not allow to catch properly the dynamics of a fast event like surge.

Even if all the variables mutually influence each other and so it is not possible to analyze them individually, data validation of a single variable can give an indication of the accuracy of a particular part of the dynamic model.

A good data alignment of pressures in both, full speed and emergency shut-down tests, is mainly an indication that valves and all volumes of the gas loops have been simulated correctly.

Train speed predictability mainly indicates the capability to model the compressor speed control system and accuracy of the torque balance equation. Model accuracy in predicting train speed can be better assessed in the full speed tests, where compressor speed control system is active and modulating electric motor delivered torque. In order to achieve a good speed matching, an accurate estimation of compressor absorbed torque is necessary, especially in the Second Quadrant. In the First Quadrant, the Torque can be calculated as follow:

$$T_{CC} = \frac{\frac{\mathrm{H}_{pol,CC}}{\eta_{pol}} \cdot \dot{m}}{\omega} \qquad \textbf{Eq. 4}$$

In the Second Quadrant, Equation (4) can be considered to be still valid: as soon as the operative point shifts to the Second Quadrant, the absolute value of polytropic head and efficiency does not vary significantly, while the mass flow decreases till almost 10-20% of the direct flow condition (see Figure 12, Figure 13, Figure 14 and Figure 15). Hence, in full speed tests, when the operative point moves to the Second Quadrant, the compressor absorbed torque decreases significantly leading, according to the torque balance equation (Equation (3)), to an increase in the compressor speed. This behavior has been correctly caught by dynamic simulation, as shown in Figure 12, Figure 13 and Figure 14 (black lines).

Finally, the predictability of mass flow processed by each compressor section is an indication of the accuracy of the model of the compressor itself.

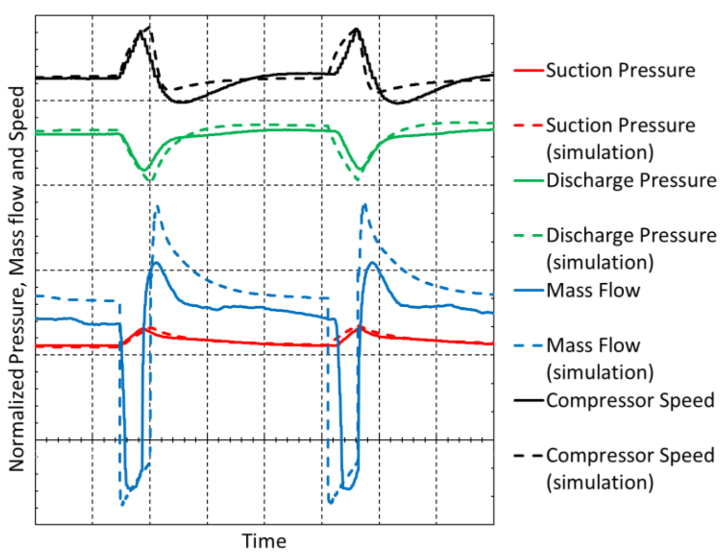

Figure 12. Surge tests data validation between dynamic simulation and test data – Section 1 Surge at full speed

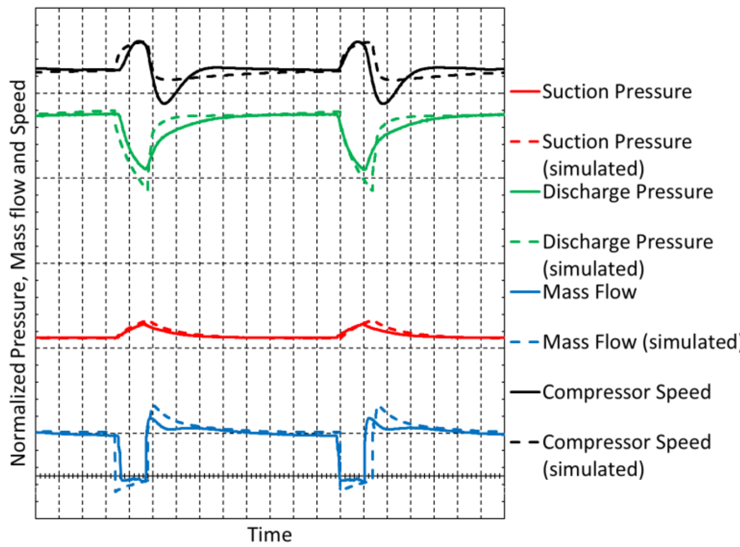

Figure 13. Surge tests data validation between dynamic simulation and test data – Section 2 Surge at full speed

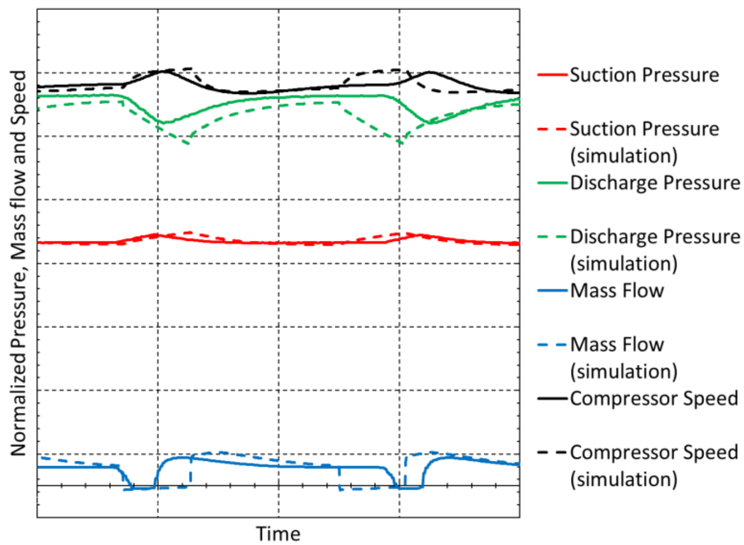

Figure 14. Surge tests data validation between dynamic simulation and test data – Section 3 Surge at full speed

As highlighted above, when the operative point is recovering from a reverse flow condition, i.e. flow processed by the compressor is increasing, the train also experienced a speed increase. Since speed has a direct influence on the compressor deliverable head (compressor deliverable head is proportional to the square of the speed), the equation of motion (Equation (2)) is also affected: the greater the compressor deliverable head, the higher the derivative of flow and so the less time is spent in the second quadrant. Derivative of flow, and so the time spent in the second quadrant, is also affected by the process polytropic head and so by the rate of suction/discharge pressures equalization which is strongly dependent on plant volumes, reverse mass flow through the compressor and mass flow passing through the recycle valves. This leads to the conclusion that all the variables of the compressor loop have an influence on Surge cycles parameters: reverse flow duration, number of cycles and maximum and minimum direct/reverse flows. Since these are the key parameters on which the Surge assessment of the machine during ESD is based, a good dynamic simulation model, capable of correctly predicting all the variables that describes the compressor loop behavior during both normal running and ESD scenarios, is essential to properly design Surge protection devices.

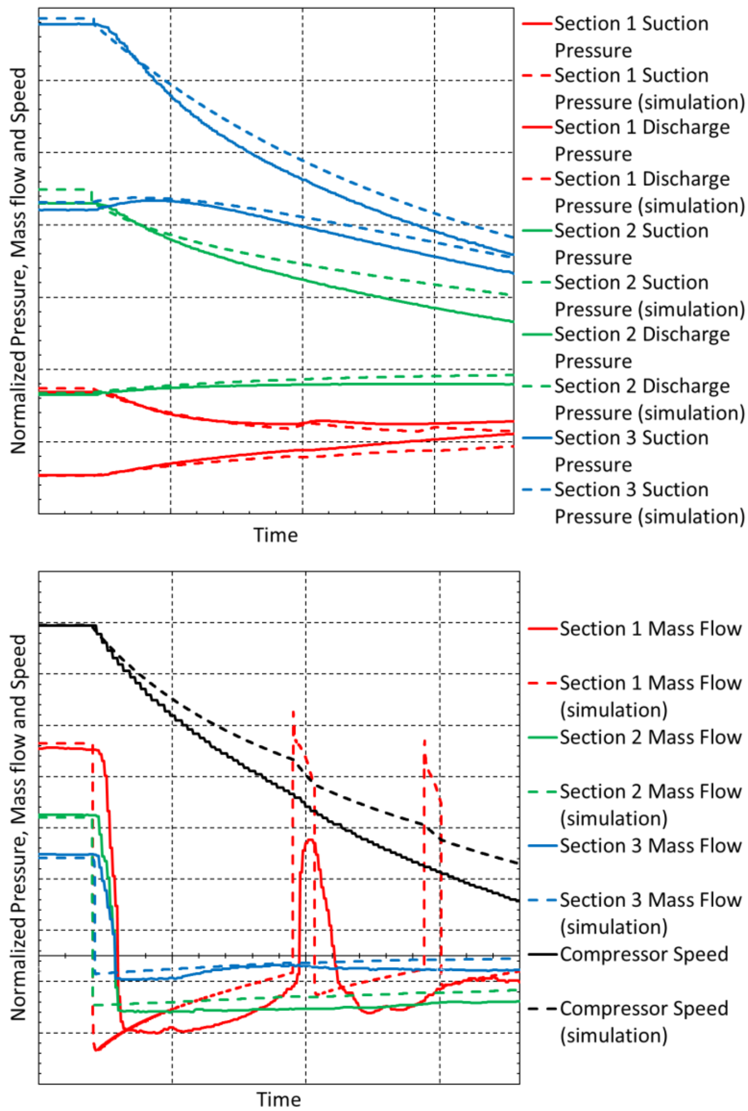

Figure 15 Surge tests data validation between dynamic simulation and test data – ESD with Recycle Valves frozen

Looking at tests results data validation from a Surge cycles parameters standpoint, it can be observed that:

- maximum and minimum direct/reverse flows were the parameters fitted in the tuning process of the three compressor stage curves. Indeed, the simulated values almost perfectly fit the field data;
- time spent in reverse flow has been fairly matched in section 1 and 2 surge at full speed and ESD tests (within 20% difference), while it has been overestimated in section 3 surge at full speed test (+70%);
- number of surge cycles predictability can be assessed only in the ESD scenario. In this test, second and third HPRC sections coast down occurred in reverse flow condition consistently with the test run; even the timeframe relevant to the first section excursion in the Second Quadrant is matched exactly as per Figure 15. Anyway, the dynamic simulation tended to overestimate the number of Surge cycles upon coast down (for section 1, simulation estimated 2 surge cycles instead of only 1), therefore leading to a more conservative analysis of the ESD event.

6 CONCLUSIONS

As the final step of the test campaign on HPRC, Surge exploration tests have been performed allowing for an assessment of the machine design robustness and data collection in reverse flow operation. The tests have been carried out on different scenarios of gas loop pressurization and control action on recycling valves, to differentiate the Surge severity.
An important outcome is that reverse flow condition can be withstood by the compressor even for any amount of time, since vibrations and axial thrust trends decayed smoothly during the compressor coastdown without any significant peaks in the amplitude and increased only during flow reversal, as shown in Figure 3, 5, 7, and 8. However, Surge phenomenon associated with the change in forward and backward gas flow, requires that a threshold on the number of Surge cycles shall be taken into account to avoid

undesirable upsets due to pressure waves and vibrations. Therefore, a new ESD Surge acceptance criterion considering three Surge cycles as threshold has been developed, instead of a maximum allowable operating time beyond the compressor SLL.
Data validation between Surge exploration tests and dynamic simulations, along with the characterization of the second quadrant branch, allowed for the enhancement and fine tuning of the capability of transient compressor modelling. A good level of consistency between the main parameters affecting the transient compressor operability has been demonstrated. With the aid of this activity, a more accurate evaluation of compressor Surge and reverse flow parameters (especially the number of Surge cycles) is achieved whenever a dynamic simulation is performed on a particular compressor application.
Surge exploration test results analysis and dynamic modeling enhancement have been a breakthrough for the introduction of a new Surge Acceptance Criterion based on number of Surge cycles instead of time spent in the Second Quadrant. The new criterion is applicable not only to HPRC but also to standard compressors, with expected benefits in reducing the complexity of Surge protection devices (such as Hot Gas Bypass Valve or Cold Gas Bypass Valves) and associated costs, whilst still maintaining the safe operation of the compressor and related compressor system equipment.

NOMENCLATURE

Variables

ω	= Compressor speed	(t^{-1})
t	= Time	(t)
T	= Temperature, Torque	(T, $M \cdot L^2 \cdot t^{-2}$)
GR	= Gear Ratio	(-)
f	= Friction Loss factor of the train	($M \cdot L^2 \cdot t^{-1}$)
I_{TOT}	= Train total Inertia	($M \cdot L^2$)
τ	= Work coefficient	(-)
u	= Peripheral speed	($L\ t^{-1}$)
D	= Impeller diameter	(L)
φ	= Flow coefficient	(-)
$\dot{m}$	= Mass flow rate	($M \cdot t^{-1}$)
M	= Mass	(M)
L	= Length	(L)
ρ	= Mass density	($M \cdot L^{-3}$)
r	= Pressure ratio	(-)
P	= Total Pressure	($M \cdot L^{-1} \cdot t^{-2}$)
H	= Head	($L^2 \cdot t^{-2}$)
η	= Efficiency	(-)
Ψ	= Dimensionless polytropic head	(-)
Mu	= Mach number	(-)

Subscripts

EM	= Electric Motor
CC	= Centrifugal Compressor
P	= Process
Des	= Design
pol	= Polytropic
2	= Section at the external diameter of the impeller

Acronyms

BHGE	= Baker Hudges a GE Company
HSS	= High Speed Shaft
HPRC	= High Pressure Ratio Compressor
DE	= Driver End
NDE	= Non-Driver End
ASV	= Anti Surge Valve
ESD	= Emergency Shut-Down
SLL	= Surge Limit Line
HGBV	= Hot gas Bypass Valve
CGBV	= Cold gas Bypass Valve
1xRev	= Revolution Frequency

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