45TH TURBOMACHINERY & 32ND PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 12 – 15, 2016
GEORGE R. BROWN CONVENTION CENTER

FULL SCALE VALIDATION OF A HIGH PRESSURE RATIO CENTRIFUGAL COMPRESSOR

Stefano Falomi
stefano.falomi@ge.com
Lead R&D Engineer
GE Oil & Gas

Giuseppe Iurisci
giuseppe.iurisci@ge.com
Engineering Manager
GE Oil & Gas

Stefano Fattori
stefano.fattori@ge.com
Lead Instrumentation Engineer
GE Oil & Gas

Angelo Grimaldi
angelo.grimaldi@ge.com
Senior NPI Engineer
GE Oil & Gas

Carlo Aringhieri
carlo.aringhieri@ge.com
Systems Engineer
GE Oil & Gas

Giuseppe Sassanelli
giuseppe.sassanelli@ge.com
Senior Section Manager
GE Oil & Gas

Gianni Iannuzzi
gianni.iannuzzi@ge.com
Lead Test Cell Engineer
GE Oil & Gas

Stefano Falomi is mechanical designer for centrifugal compressors in the Advanced Rotating Equipment Department, with main focus on wet gas compression and High Pressure Ratio Compressors. He earned M.Sc. degree in Mechanical Engineering in 2007 from University of Florence. In 2010 he received the PhD degree in Mechanical Engineering from the University of Florence, with a study on torsional vibrations of compression trains with large variable frequency drives.

Giuseppe Iurisci is Manager of Mechanical Design team inside Advanced Rotating Equipment Department, where he's responsible for the mechanical conceptual design and development of new technologies for Centrifugal and Axial Compressors. He has been with GE Oil & Gas since 2006, working as a design engineer, Engineering Manager of Performance and Applied Aerodynamic Team. He has a Bachelor's and Master of Science Degree in Aerospace Engineering from Politecnico of Turin (Italy) and a Master of Science in Thermal Power Aerospace Propulsion (Cranfield University - UK).

Stefano Fattori is currently Test Data Analysis Team Leader within Turbomachinery Laboratory. He is responsible of data acquisition, analysis and post-processing of measurements from testing and validation of new products.
Stefano holds a degree in mechanical engineering from University of Florence; he joined GE Oil & Gas in 2008 as Test Project Engineer inside the Advanced Technology organization, becoming an expert of turbomachinery validation technologies.

Angelo Grimaldi is Principal Engineer for Centrifugal Compressor Aerodynamics. His responsibility includes design, development and analyses of centrifugal compressor aerodynamic components. He joined GE Oil & Gas in 2005 as CFD specialist in the Centrifugal Compressor Research and Development team. Dr. Grimaldi graduated in Mechanical Engineering in 2001 from the Polytechnic of Bari; in 2004, he gained the Ph.D. on Numerical Fluid Dynamics from the same University.

Carlo Aringhieri is a Systems Engineer inside Advanced Technology Department for GE Oil & Gas in Florence (Italy) where he's responsible for the development of new products and technologies. He has been with GE Oil & Gas Advanced Technology group since 2008 and has worked as Program Manager for new products and technologies before covering his current position. He has a Master of Science in Aerospace Engineering from University of Pisa (Italy) and a Master of Science in Engineering and Physics of Plasmas from University of Padova (Italy).

Giuseppe Sassanelli is a Senior Engineering Manager with ~17 years' experience in Turbomachinery design R&D, product and solutions development, in the context of large scale, high tech company with large global footprint. He is currently leading the conceptual mechanical team for GE Oil & Gas turbomachinery in Florence. Prior the current role he has been global manager aerodynamics for GE Oil & Gas and previously has led the NPI team for centrifugal compressors. Giuseppe received the Edison Pioneer Award in 2012. He is author of several patents and papers.

Gianni Iannuzzi is Engineering Manager for turbomachinery testing design for GE Oil & Gas, Florence, Italy. His current responsibility is to manage the testing facilities' design for Oil & Gas rotating equipment, including plant R&D projects. Mr. Iannuzzi joined GE Oil & Gas Testing Department in 1995; he worked on string testing activities ranging from the main LNG trains to ultra-high pressure applications, such as Kashagan. Mr. Iannuzzi graduated in Mechanical Engineering at University of Florence.

ABSTRACT

The present paper describes the test activities which have validated a new compressor architecture, developed by the OEM, which is able to deliver a higher pressure ratio in a single casing with respect to traditional configurations.
The major differences with respect to the typical technologies used for high pressure machines are in the rotor design:

- Rotor stacked configuration with impellers connected through high precision toothed joint and a pre-stretched tie-rod;
- Shrouded and unshrouded impellers on a multistage between-bearings compressor;
- High peripheral speed journal and thrust bearings.

The combination of unshrouded and shrouded impellers allows achieving same pressure ratio in a shorter bearing span, giving the possibility to increase the number of sections for each unit, making possible to reduce the number of casings.
A full scale prototype has been built and tested to validate the new machine architecture. The prototype has been designed to fulfill a compression service made of three compression sections with a single unit; the same service could only be performed with at least two units if a standard compressor design was applied. The service requires serial compression of natural gas, characterized as follows:

- Design inlet volumetric flow close to 10800 m3/h
- Design pressure ratio close to 30
- Absorbed power ≈ 14 MW
- Design speed of 17619 RPM (MCS = 18500 RPM, mos = 14095 RPM);

First section comprises 2 open impellers; second section comprises 3 closed impellers and third section has 2 closed impellers.
Impellers for first and second sections are in stacked configuration; third section's impellers are mounted through shrink fitting.
Considering the increased speed requirement, the prototype has been tested in conjunction with an epicyclic gear, which allows for improved efficiency for high gear ratio (12.34 in this case) with respect to standard parallel axes gears.

TECHNOLOGY VALIDATION

HPRC validation strategy was focused on verifying new technologies injected into the centrifugal compressor train, starting from component tests to a full prototype testing in actual intended environment as per TRL4 of API 17N.
The validation activity comprised a variety of tests mainly focused on machine integrity, compressor performance, operability and response to upset conditions.
The compressor has been equipped with a dedicated special instrumentation, to fully verify compressor behavior both in steady state and transient conditions. The validation has been achieved through the following tests:

- Extensive rotordynamic test campaign, with different suction pressure levels, that included also dedicated tests to assess the

squeeze film damper influence on rotordynamic stability and forced response;
- Aeromechanical test with different inlet density levels, which has validated the methodology for structural dynamics' assessments and the design for unshrouded impellers;
- Full load compressor performance test (section by section and overall) for different pressure levels, with suction pressure varying from 1.5 barA to 5.5 barA and discharge pressure up to 130 barA;
- Validation of thrust model developed for high pressure machines through thrust assessment on different operating conditions;
- Test of upset conditions such as emergency shut-down, surge region exploration, fast startup and rotor bowing for hot restart conditions.

The present document summarizes the test results that validated the operation of the machine considering both the components that distinguish this prototype from standard compressors and the components that are operated with peripheral speed higher than existing compressors:
- Central tie-rod, with acceptable static and dynamic load in all operating conditions;
- Dry Gas Seals temperature within acceptable limits during both component test and compressor test;
- Journal bearings' temperature well below alarm values and free of any thermal instability effect;
- Thrust bearing temperature well below alarm and within vendor expected values..

TEST ENVIRONMENT

Test was performed on a permanent test bench in Florence (Figure 3) which includes three independent gas loops, a full auxiliary system (lube oil, DGS and cooling) and an electric motor driver. Maximum allowable power absorption for testing cell is about 10 MW.

In addition to the centrifugal compressor, which was the focus of this validation test campaign, the epicyclic gearbox is worth of mention. It is a 12.3 gear ratio compound style gearbox; more details can be found in Maragioglio et al.

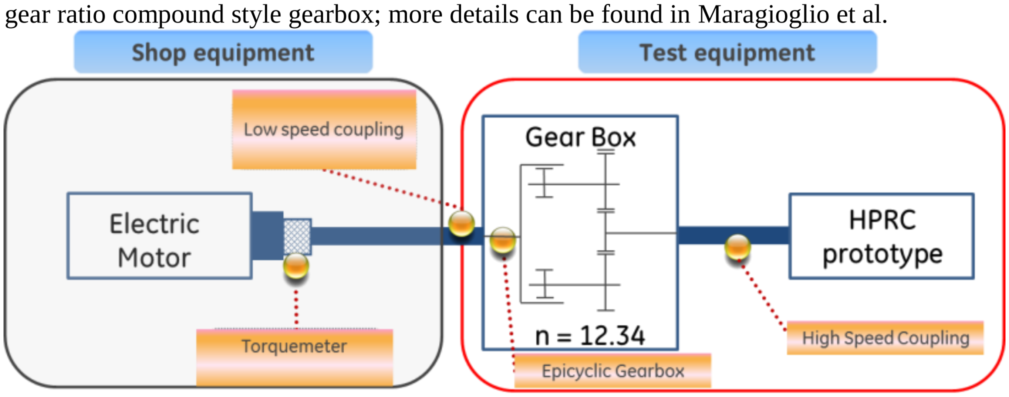

Figure 1: Compression train schematic

In order to simulate a serial operation, the three gas loops are connected by two by-pass lines provided with dedicated valves (PCV-4 and PCV-5 as shown in in Figure 2) that simulate pressure drops of inter-coolers that would have been present in actual process.
Those bypass lines also allow to compensate the leakage flows across compressor balance drums.

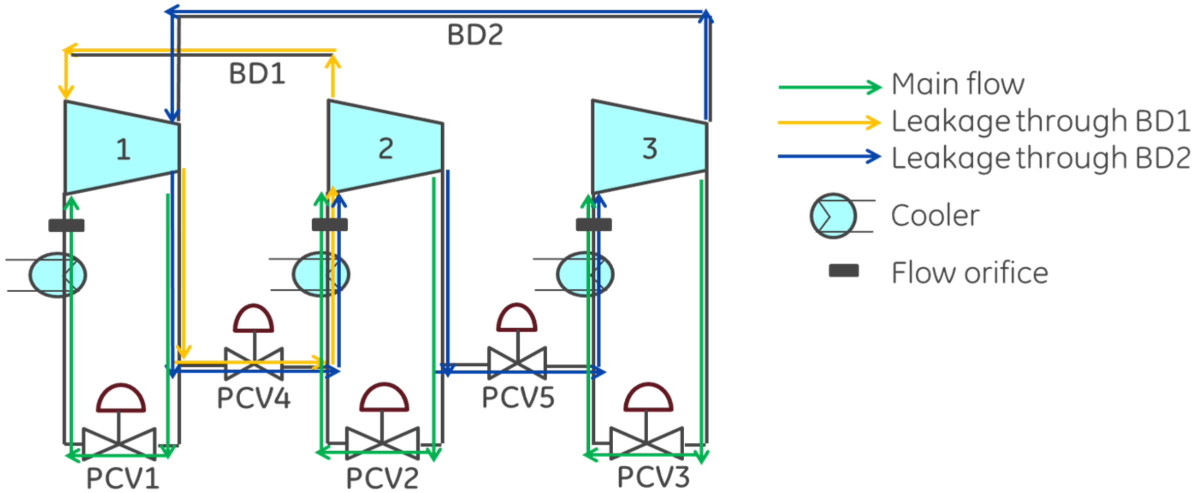

Figure 2: test loop schematic

Figure 3: test loop in Florence facility

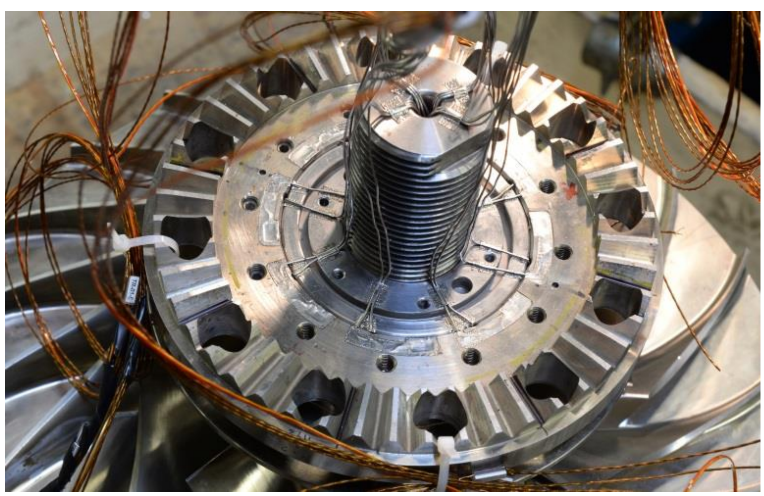

Figure 4: Instrumented rotor

INSTRUMENTATION

The instrumentation setup comprises:

- Standard instrumentation for centrifugal compressor performance test, according to ASME PTC10;
- Standard instrumentation for rotating machinery protection (API 617, API613, API670);
- Special instrumentation to validate compressor mechanical and thermo-mechanical behavior in both steady state and transient conditions.

In particular, the special instrumentation comprises:

- Strain gauges on first section open impellers for aeromechanic validation;
- Strain gauges and thermocouples on rotor tie-rod;
- Thermocouples on diaphragms and casing;
- Temperature measurements on DGS;
- Load cells on thrust bearing;
- Accelerometers on compressor external casing.

In addition, following instrumentation was added to have more detailed analyses of compressor performance:

- Flow and temperature measurement on compressor external balancing lines;
- Pressure measurement in first section last diffuser
- Static pressure and temperature measurements on labyrinth seal between section 1 and 2.

TEST RESULTS

This paper summarizes test results for the compressor integrated in the gas loop described in previous section.
The compressor has been tested for 300+ hours in a great variety of process conditions:

- suction pressure varying from 1.5 barA to 5.5 barA and discharge pressure up to 130 barA;
- compressor absorbed power up to 10 MW;
- process gas molecular weight from ≈16.6 g/mol (methane from civil distribution pipeline) to 42.5 g/mol (98% CO_2 plus methane and nitrogen traces) to perform a reduced speed Type II performance test according to PTC-10;
- different values for startup speed ramps to simulate both direct on line startup and variable speed motor ramp up;
- performance points stabilized from mos to MCS;
- full exploration of compressor’s curves from surge to deep choke to both validate compressor operability and open impellers’ aeromechanics in different flow regimes.

The paper will mainly focus on performance test results and compressor mechanical integrity. Other aspects (rotordynamics, compressor operability and aeromechanics) will be detailed in separated publications.

Performance test results

The three sections are designed for serial operation: the compressor performance are thus presented in Figure 5 and Figure 6 referring to the serial operation of the three sections, showing three points for the design speedline: overall choke, design point and overall surge. The performance curves refer to a test condition with suction pressure close to 4 barA, with gas molar mass close to 24. g/mol.
In a serial operation, the last compressor section is the one that limits the compressor operating range, so these curves are not reflecting the actual width of the section by section curves.
Test points (green dots) are plot versus expected performance (solid blue line) in non-dimensional form with respect to compressor design point. Performance is well in agreement with predicted values both in terms of polytropic head and efficiency.
Table 1 summarizes main outcomes of thermodynamic test results: machine delivers required head with small excess (2%) that is acceptable for both fixed speed (head within 100%-105% of rated) and variable speed drivers. The absorbed power is -0.3% with respect to expected, so it is acceptable for both fixed and variable speed applications. Deviation is within API specifications.
Note that deviation in efficiency is expressed as a percentage of design efficiency. For section 3 the deviation seems excessive (+4.3%) but this is due to the fact that section 3 is designed for very low volumetric flow, that results in low efficiency for design point. The actual difference in terms of percent points is slightly more than 2%.
The overall pressure ratio delivered by compressor is well matching the expected curve, with a maximum pressure ratio for design speed close to 35.

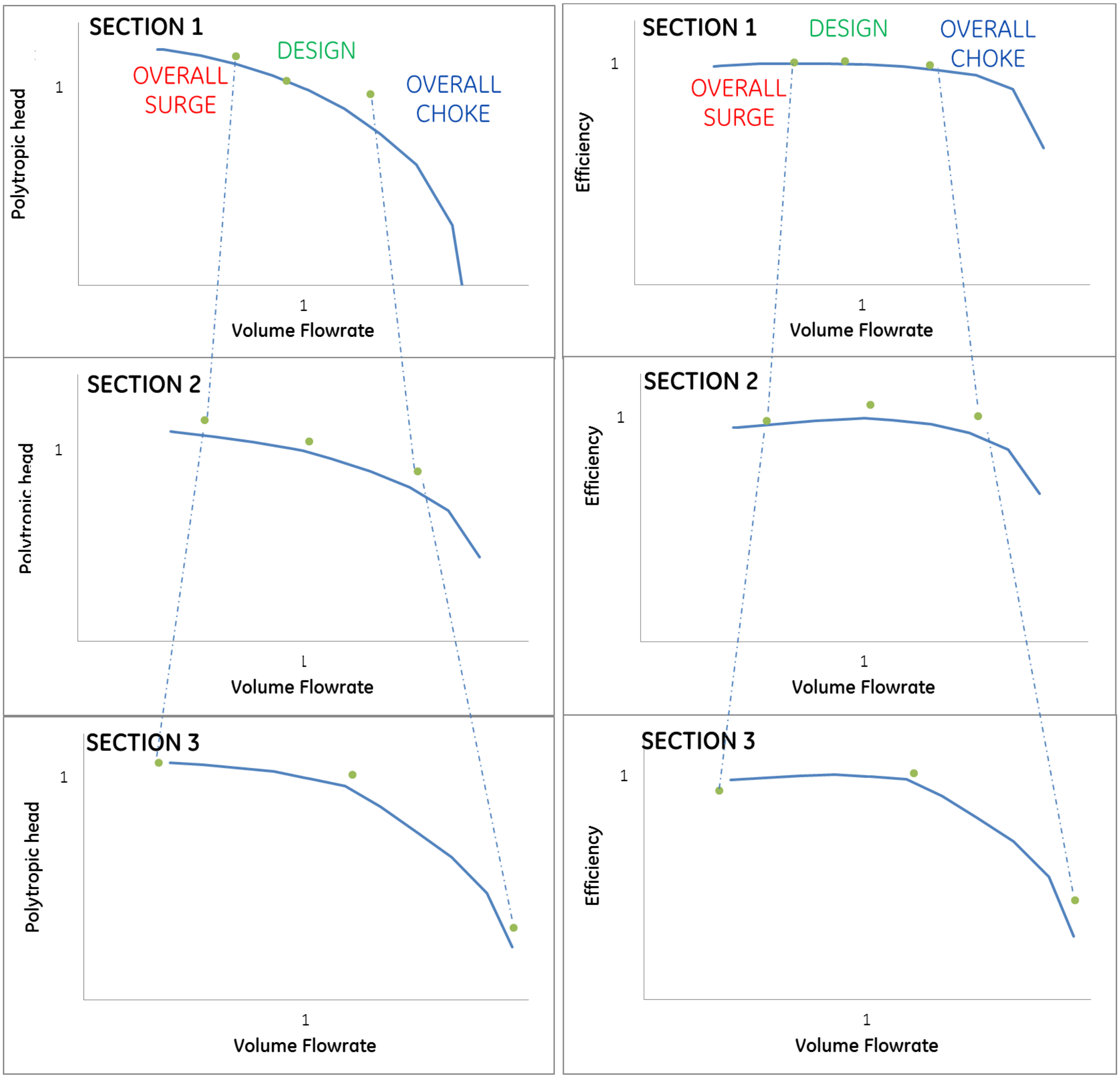

Figure 5: Section polytropic head and efficiency, design speed: expected vs. test

	Deviation (wrt expected)		
	Pol. Head	Efficiency	Power
Section 1	+0.1%	+1.1%	
Section 2	+2.7%	+2.9%	
Section 3	+7.4%	+4.3%	
Overall	+2.2%	+1.9%	-0.3%

Table 1: performance (test vs. expected) at design point

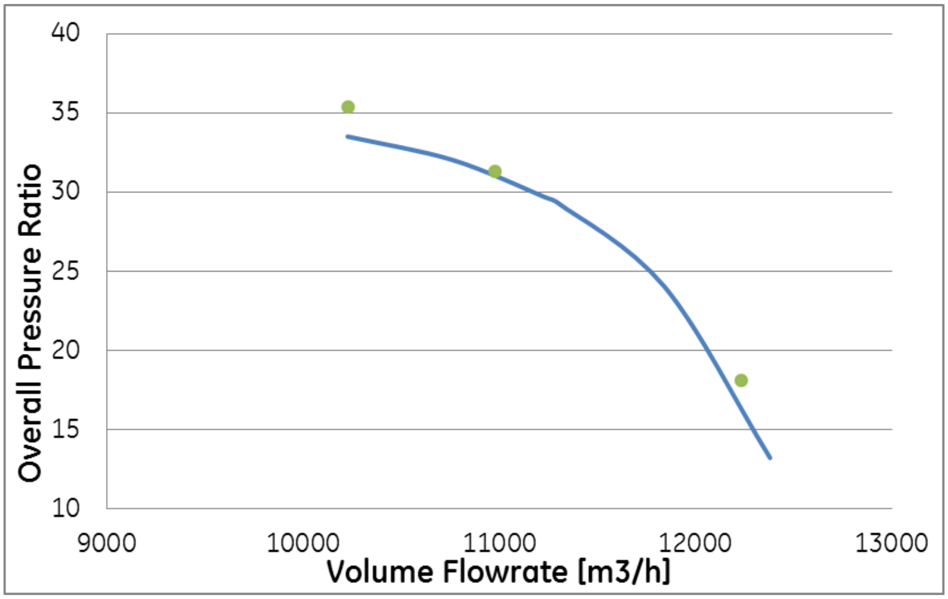

Figure 6: Overall pressure ratio vs volumetric flow, design speed: expected vs. test

In addition, the compressor has been tested section by section, to check the operating flow range for each section. In particular, it is interesting to show the comparison between expected and tested compressor curves for section 1, which is the one that includes two open impellers in a between bearing configuration, that is not typical for standard compressors.
The expected has been built through CFD for impellers, suction volute and discharge scroll; CFD results were tuned considering experimental data coming from single stage test of an open impeller of same family, but designed for higher volumetric flow.
The curves shown below refer to a full curve with suction pressure close to 4 barA and molar weight close to 19.3 g/mol; a good agreement was found between test data and expected.

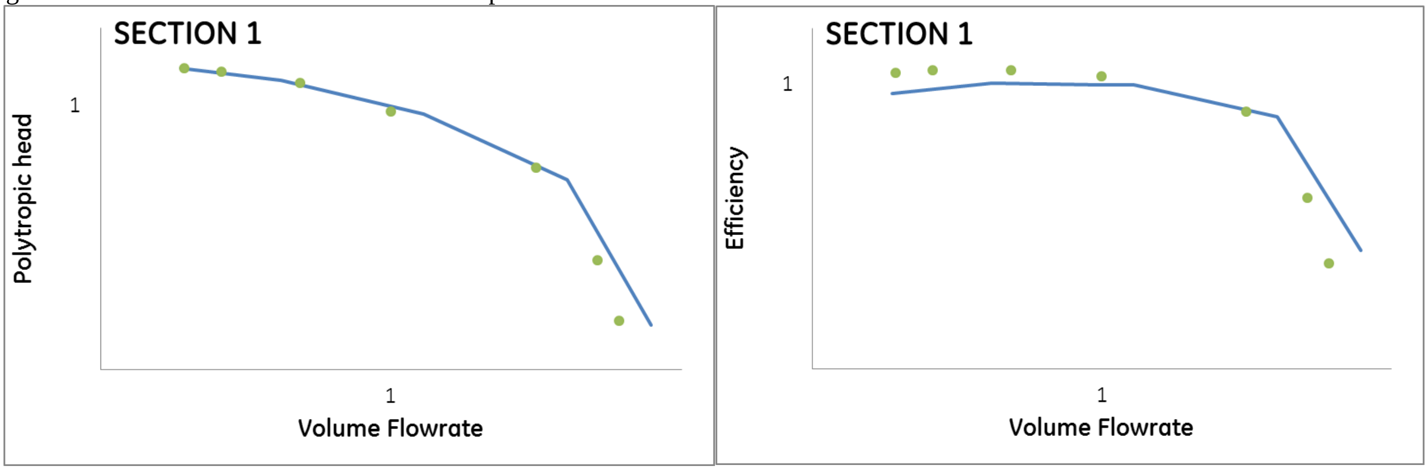

Figure 7: Polytropic head: head and efficiency, design speed: expected vs. test

Mechanical behavior
The aspects covered within this section are those that differ from standard between bearings centrifugal compressors. In particular, following aspects will be detailed:

- Thrust bearing temperature, with peripheral speed at MCS close to 140 m/s (at average pad diameter);
- Axial thrust, to validate thrust balancing scheme and prediction tools;
- DGS temperature during component test and compressor test with tip peripheral speed at MCS close to 170 m/s;
- Axial load on rotor central tie-rod in operating conditions.

Mechanical behavior: thrust bearing

Before testing the unit, the thrust bearing vendor has provided expected pad temperature for different test conditions, based on following hypotheses:

- Oil is ISO VG32
- Oil inlet temperature is 50°C,
- Oil flowrate is constant with speed and equal to design flow.

Expected pad temperatures from supplier have been interpolated as a function of speed [RPM] and load [N] to build an online expected function and compare measurements vs. expected in real-time. The fitting is linear with load and quadratic with rotating speed.

Here below (Figure 8) a first comparison of expected and measured pad temperature (average temperature of pads on loaded side) is shown. The expected is built from the above mentioned fitting, with the axial load on bearing measured from thrust bearing's load cells. It is possible to note that speed is the primary driver for pad temperature: the temperature steep variations occur simultaneously to speed ramps (at time 5000, from mos to 100%, at time 10000 from 100% to 105%) performed for aeromechanical validation. Load has a secondary influence on pad temperature: dependency is visible for time 4000 where the axial load increase results in both expected and measured pad temperature increase.

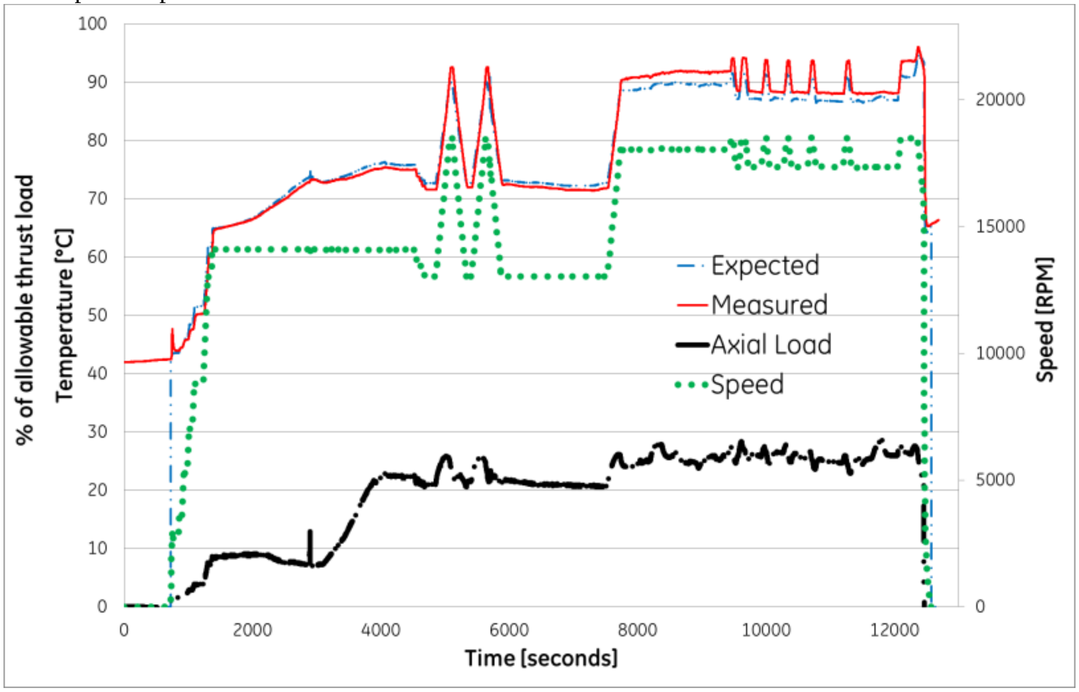

Figure 8: Thrust bearing pad temperature: expected vs test

Measurements show a good agreement with expected temperatures.

As summarized in the beginning of this paragraph, the compressor has been operated in a variety of conditions, leading to identify the most severe conditions for all components.

For the thrust bearing, it is of great interest to analyze temperature trend for the test condition where the highest pad temperature has been observed (Figure 9). Operating conditions were far from the compressor serial operation: compressor was running at MCS with section 1 at surge and 2nd and 3rd section in deep choke (to explore full 1st section curve without incurring in motor power limitation). During design phase this condition was already forecast as the most critical condition for thrust thus an auxiliary balancing line was added to avoid thrust bearing overload; anyhow, since both thrust and pad temperature were far from ultimate values, the auxiliary balancing line was not activated. As shown in Figure 9, during part of this test on 1st section, thrust load overcame the 50% of thrust bearing capacity, with pad temperature reaching approx. 98°C, thus still well acceptable for continuous operation.

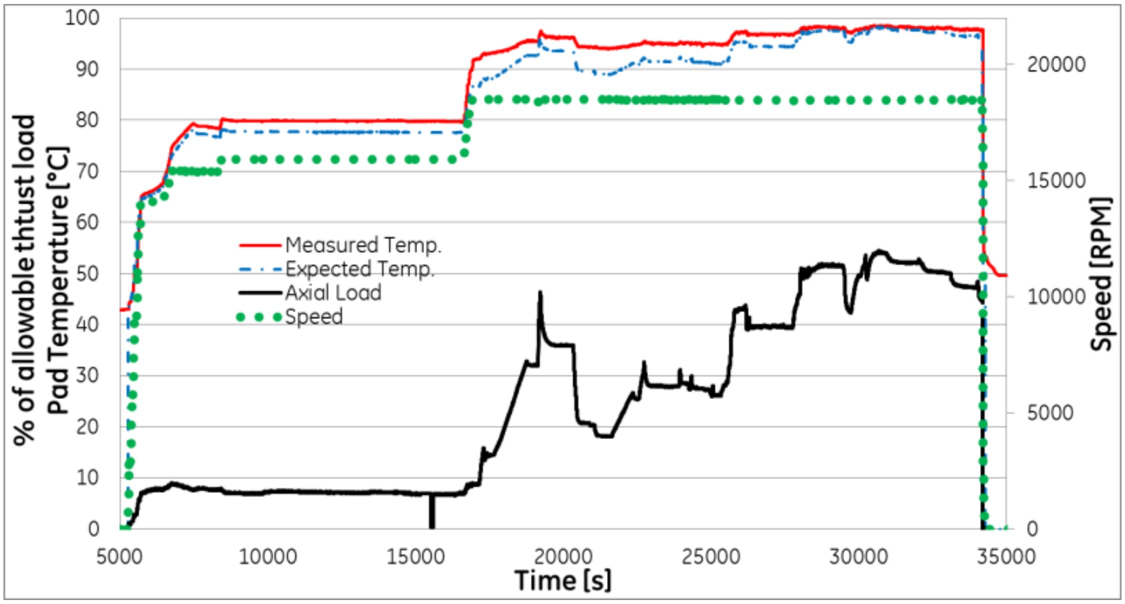

Figure 9: Thrust bearing pad temperature: expected vs test

Mechanical behavior: thrust balancing.

HPRC prototype is equipped with 2 balance drums:

- 1st balance drum operating between Suction 2 and Suction 1 pressure;
- 2nd balance drum operating between Discharge 3 and Discharge 1 pressure.

Balance drums and labyrinth seals between sections have been sized to have a residual thrust between 0 and 50% of bearing allowable load during serial compressor operation, including pressurized start-up and shutdown with transient settle-out pressure on the three sections.

Measured thrust on a serial compressor curve (choke, design and surge) has been compared with expected thrust, which has been evaluated with two different hypotheses on seals: "Min clearance" refers to compressor thrust evaluated considering all seals at minimum allowable clearance from assembly drawings, while "Max clearance" refers to thrust evaluated for maximum seals' clearance.

Thrust is plot as percentage of thrust bearing's maximum allowable load. Thrust trend and values are well captured by thrust model; the error at design point between measured and average forecast is 7% of thrust bearing capacity.

The thrust balancing scheme selected for this unit with three sections has proven to be effective in all operating conditions.

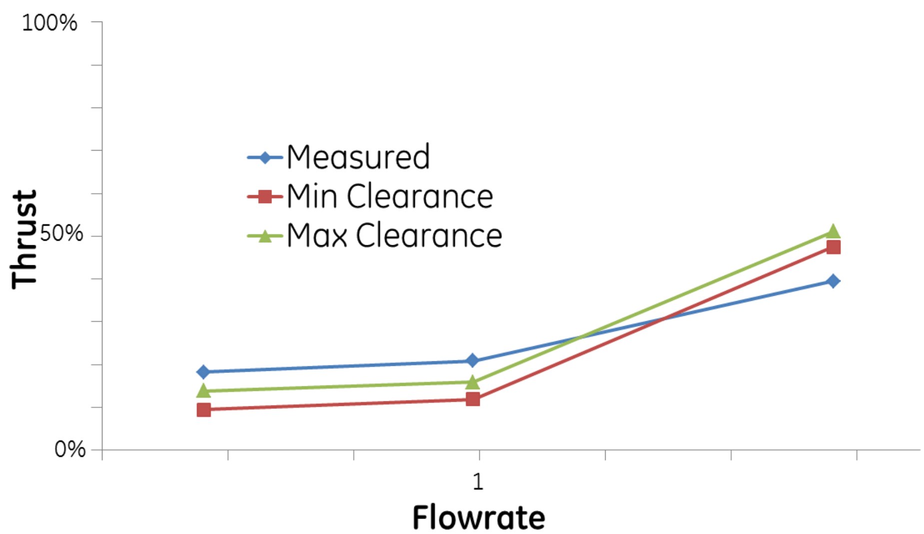

Figure 10: Axial thrust, expected vs. test

Mechanical behavior: Dry gas seals

HPRC prototype has been equipped with tandem dry gas seals with standard design and materials for high speed/high pressure applications.

Dry gas seals have been instrumented to monitor operating temperature during different operating conditions: T1 measures temperature on inboard ring; T2 is on outboard ring, while T3 and T4 are closer to compressor end head, in regions of sealing between DGS stator and end head. Each measurement is duplicated on each seal.

Results of dry gas seals' test at vendor's facility are shown below (Figure 11). As it's possible to see, maximum temperature during component test was observed on outboard ring (T2) with a value close to 211°C. This value is considered well below maximum acceptable value. Limiting components are stator O-rings (close to T3 and T4 locations) which ensure full mechanical characteristics up to 250°C.

The dotted rectangle in figure below shows a test period performed with speed equal to compressor MCS. Measured temperatures have been used to build an expected value for operating temperatures during full scale compressor test. It must be noted that primary vent backpressure during this component test is 2.8 barA, while during compressor test was set to 1.5 barA.

Expected is built as a function of compressor speed and buffer gas temperature (expected seal temperature = seal gas temperature plus a speed dependent term), based on temperature levels shown in Figure 12.

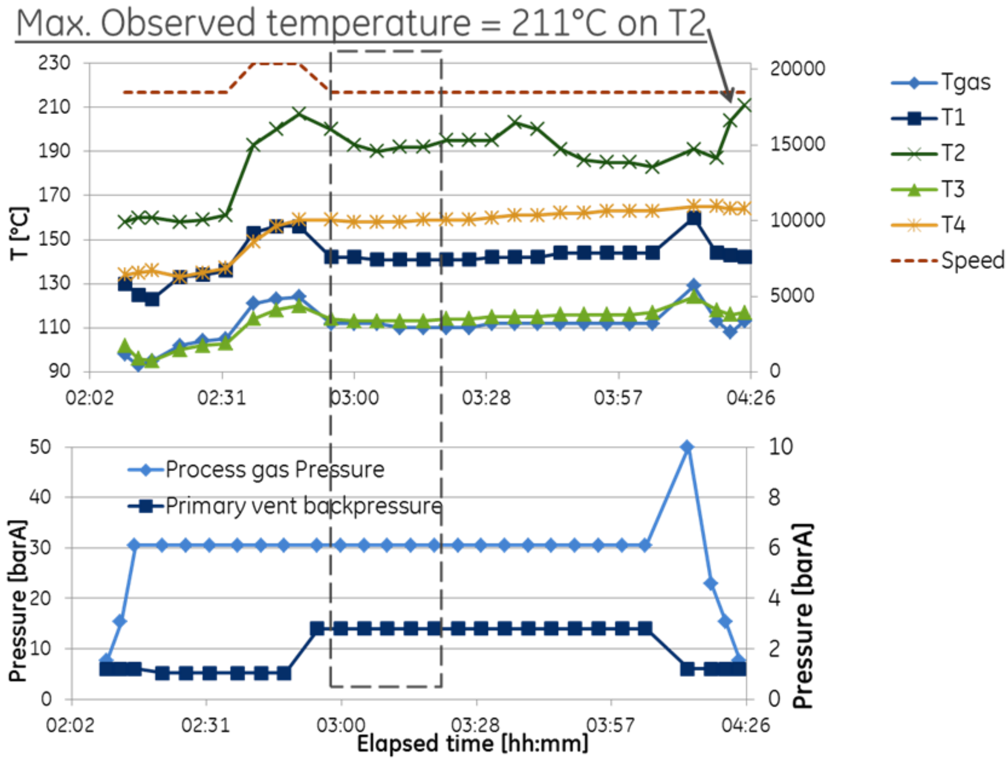

Figure 11: Dry gas seals temperature, component test

Measurements on inboard rings during HPRC prototype test are shown in Figure 12. The oscillations in buffer gas temperature (due to heater start/stop cycles) result in some oscillations on rings' temperature. Measured temperature is always well below expected. Possible explanation is that the component test rig has a far less effective heat removal if compared with a full machine, where for instance the oil circulation in nearby bearings results in continuous heat removal that leads to lower temperature.

As stated before each measurement is duplicated: the two lines for "T1 drive end" represent the two measurements on inboard ring on two symmetric points on the drive end dry gas seal. The consistent temperature trend on the corresponding sections confirms that dry gas seal is operating without any significant non-uniformity (e.g. ovalization of stator or non-uniform clearance).

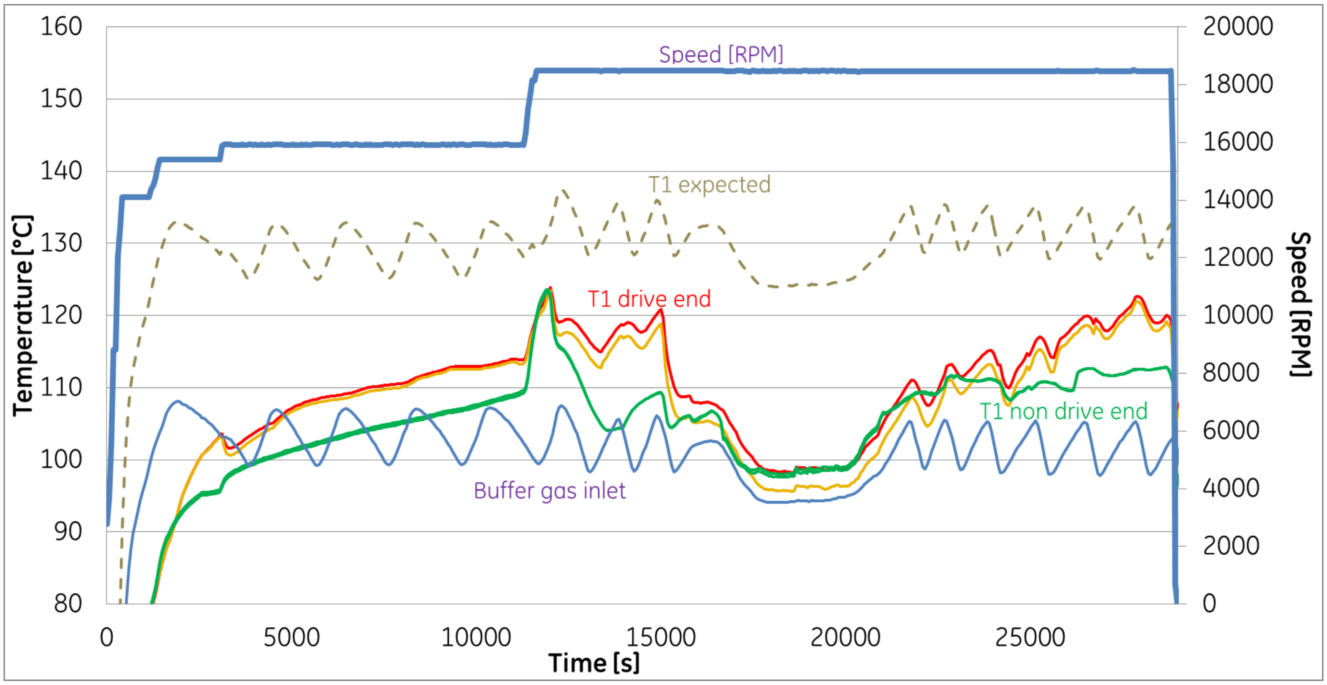

Figure 12: Dry gas seals temperature, inboard ring

For outboard rings the measured temperature (Figure 13) is slightly higher than expected since, as anticipated, primary vent backpressure during component test was 2.8 barA, while during compressor test it was set to 1.5 barA. The expected has thus been built in a condition that is not well representative of compressor actual operating condition. The different backpressure results in a far lower gas leakage during compressor test across the outboard ring (differential pressure on that ring was 1.8 bar during component test and 0.5 bar during compressor test), thus resulting in a far lesser heat removal. Anyway, the maximum observed temperature during tests has been always lower than 211°C, that were recorded during component test with 1.3 barA primary vent backpressure. The component has thus operated within acceptable limits during the whole test.

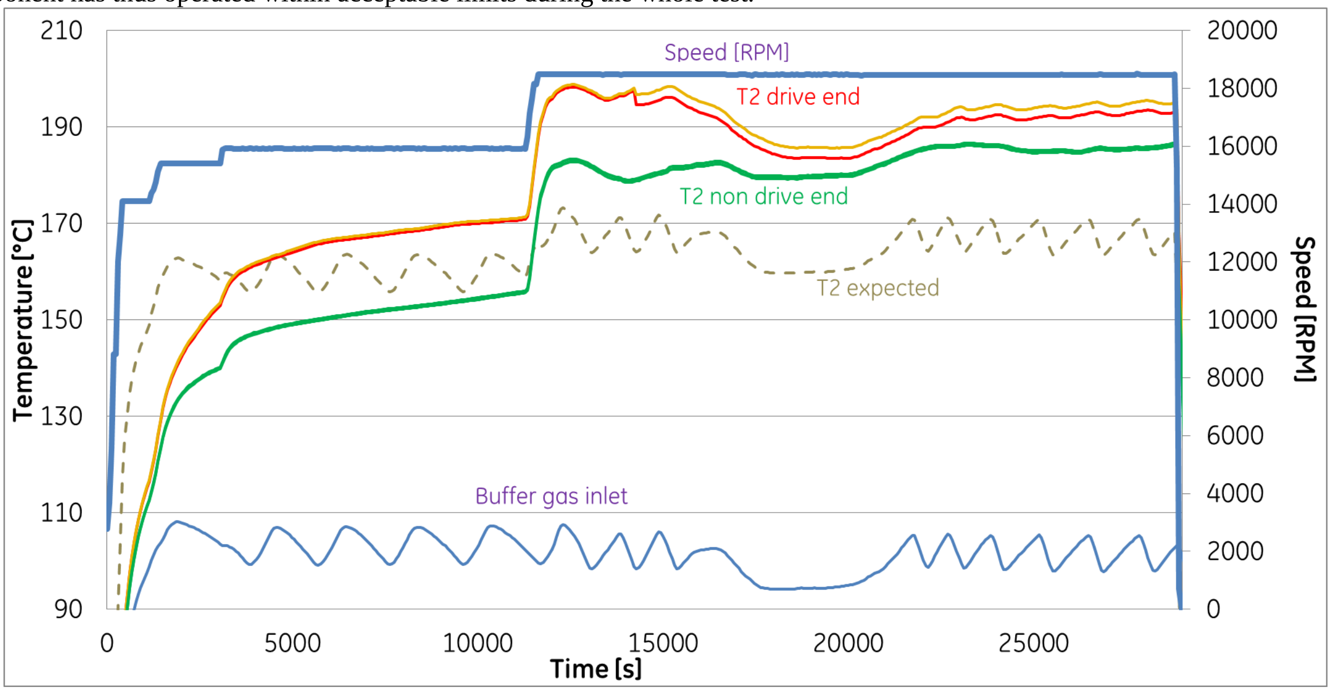

Figure 13: Dry gas seals temperature, outboard ring

Temperature levels on stator parts (T3 and T4) have been always well below expected values, thus well below maximum temperatures recorded during component test. Temperature plots for normal operation are not shown for the sake of brevity, but they are shown during compressor transient operation.
Temperature trend is shown for both startup and shutdown, to highlight that no unexpected temperature behavior has been ever observed. From first shutdown (on March) to last (on September) the behavior has always been coherent, with no sudden temperature rise (a phenomenon that has been observed on other units highlighting a sudden contact on DGS rings during coast down).
It is thus possible to conclude that dry gas seals have always operated in a reliable way.

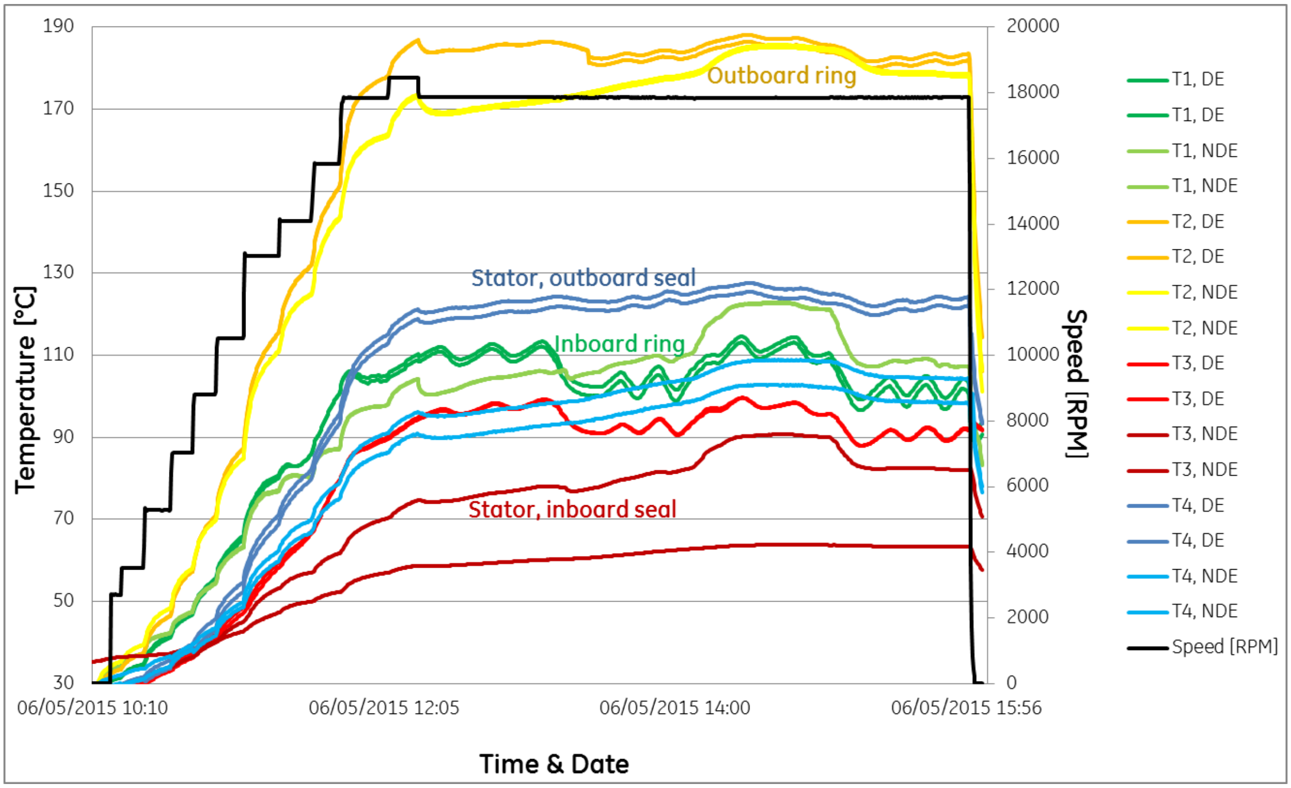

Figure 14: Dry gas seals temperature, start-up

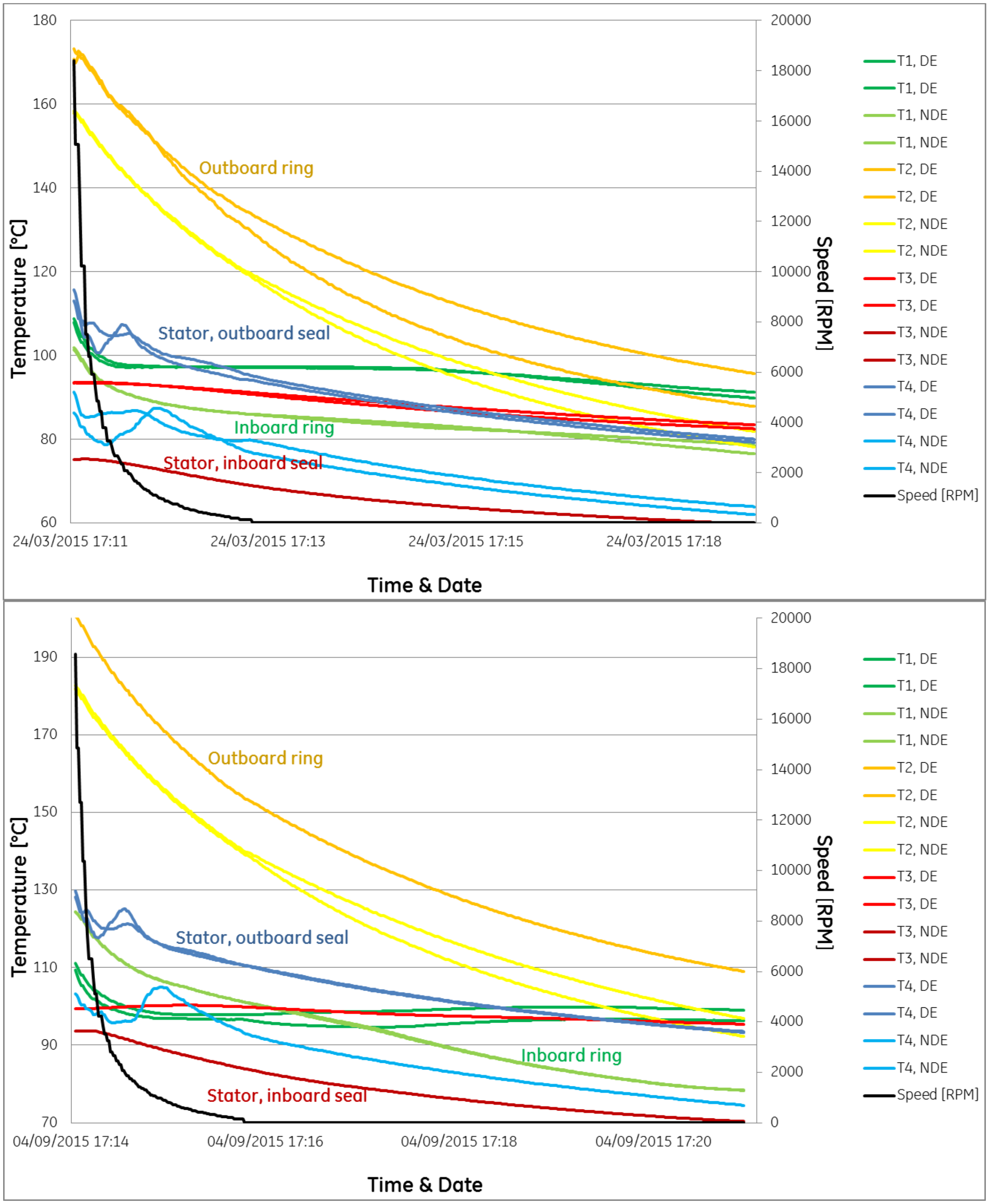

Figure 15: Dry gas seals temperature, shut-down

Mechanical behavior: central tie-rod

Last point covered in this paragraph is the tie-rod axial load during operation.

During design phase, a thermo-structural model for HPRC compressor has been built to analyze the axial load variations on tie-rod due to compressor operation in order to define the required tie-rod pre-stretch at assembly.

Main concerns during design phase were:

- Conditions that may lead to tie-rod load increase, leading to unacceptable axial stress;
- Conditions that may lead to tie-rod load excessive decrease, reducing the torque transfer capability.

Thermo-structural analysis has shown that no situation for axial load increase was expected; in addition, the minimum tie-rod axial load predicted by calculation was giving a safety factor on torque transmittal higher than 6. As a matter of fact the high precision teeth coupling has been selected to minimize the unbalance of built-up rotor due to coaxiality error among subsequent impellers; from a torque transmission standpoint the teeth coupling has high safety factor for this application.

Preload value for tie-rod has been set to 60% of tie-rod material yield. An overload test with 110% of maximum forecast stretch in operation is required by API 617. The overload test has been performed to 120% of defined preload to have a margin of 10% load increase with respect to preload.

The thermo-structural model has been built considering the “maximum case” for tie-rod axial load: fast start-up to MCS (acceleration to MCS in 80 seconds) with the threee sections in deep choke; then operating point is fast moved to surge condition for all sections.

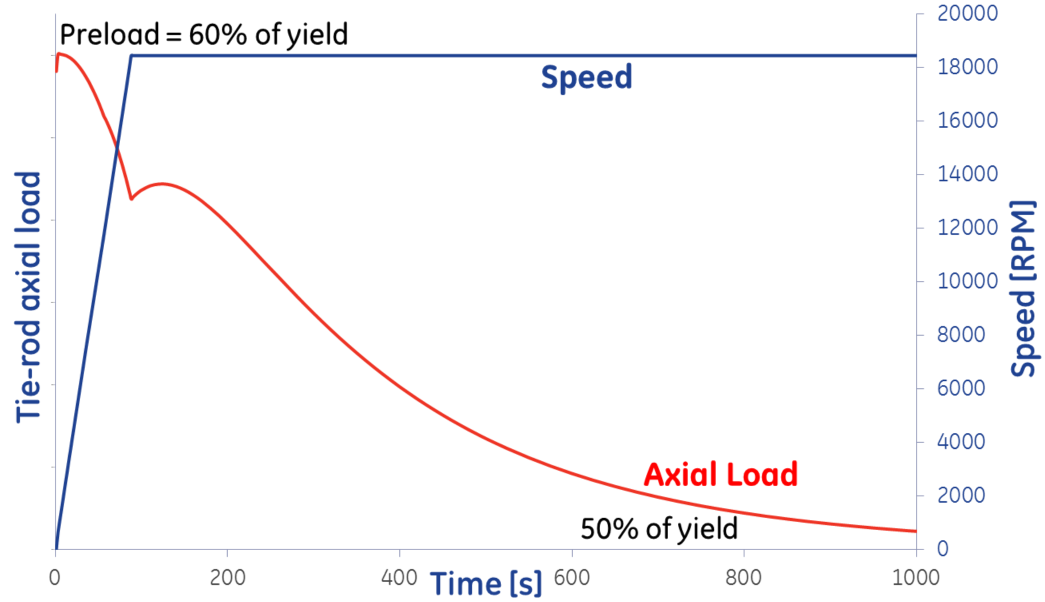

Figure 16: Tie-rod tension during startup, expected

The expected axial load variations during ramp up are shown in Figure 16.

The first effect on load is given by acceleration to MCS that leads to axial load reduction due to impellers’ axial shrinkage caused by centrifugal loads.

Then there is a slight increase given by the fact that impellers are closer to main flow path, so they’ll heat up faster than tie-rod; their axial growth slightly increases axial load on tie-rod. Then, as the whole rotor reaches the thermal equilibrium, the load reaches approximately 86% of initial load. Note that this value depends on final thermodynamic point. Since during tests usually the final point was not exactly same of thermo-structural model, there is a small difference between predicted and actual axial load.

Figure 17 shows tie-rod measured tension during a start-up similar to a compressor mechanical running test, with 10% speed ramps up to MCS and then stabilization of a performance curve at 100% speed. Final shutdown is triggered from compressor minimum operating speed. Load on tie-rod is the red line while compressor speed is the blue line. Starting from rest, the preload reduces after each speed increase; then, when speed is maintained equal to a constant value, the temperature stabilization leads to a slight axial load change. Final load corresponds approximately to 50% of material yield.

As it’s possible to see, load variability on the compressor 100% speedline, from choke to surge, is minor. Residual axial load during operation, as stated above, is almost 6 times the minimum axial force required to transfer the steady state torque.

Finally, axial load during emergency shutdown is shown. Part of initial load is recovered after reaching zero speed since centrifugal force on impellers reduces to zero. Then the load will slowly get back to initial preload value after thermal stabilization is reached.

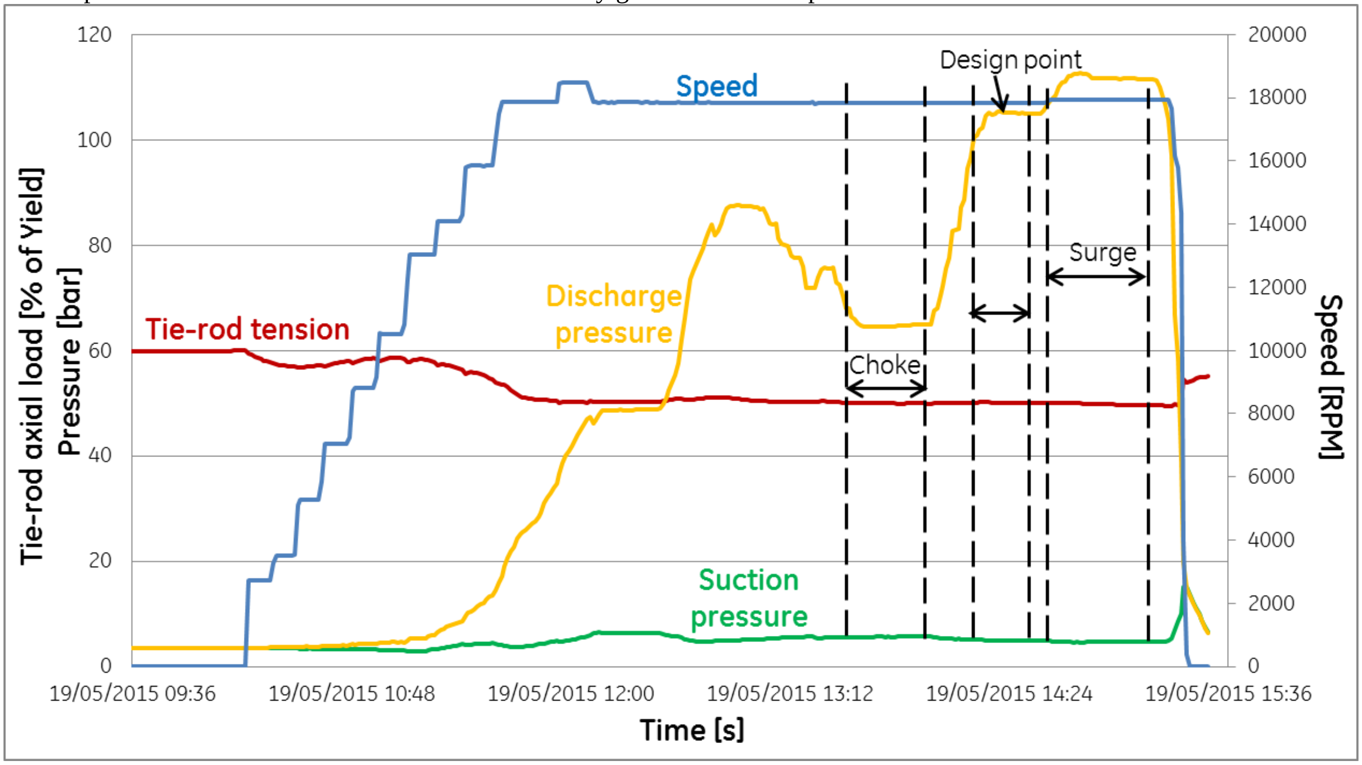

Figure 17: Tie-rod tension during start-up with speed steps, measurement

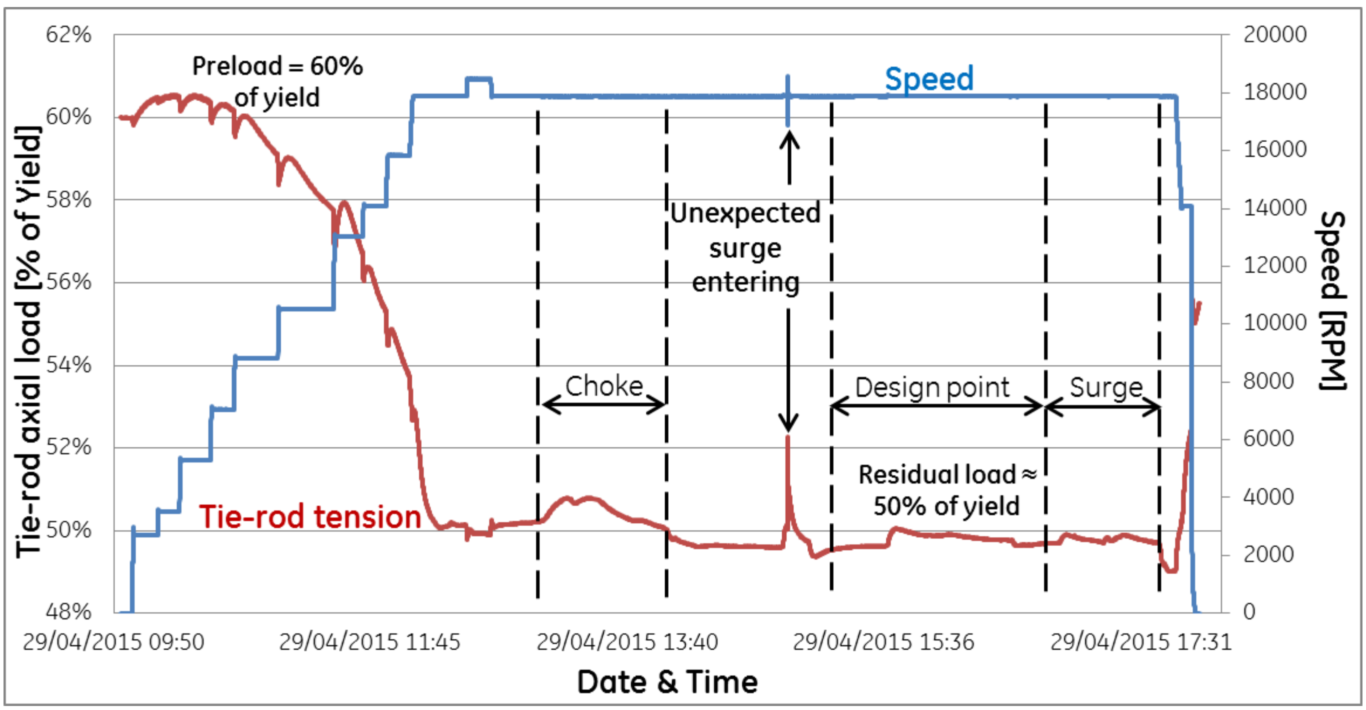

Figure 18: Tie-rod tension in surge conditions, measurement

Figure 18 shows a trend for same testing procedure, with an unexpected surge entering due to a fault on monitoring system occurred during the test (system was not updating the flowrate measurement). The tie-rod load suddenly changes and increases, but the axial load is still well below initial preload. It should be noted that the increase in axial load due to surge occurrence is less than 3% of material yield, with a static load that is close to 50% of material yield; if this excitation is thus drawn in a Goodman diagram, it will be a point well below the fatigue limit for tie-rod material, so even the occurrence of several surge cycles will not have an impact on the fatigue life of central tie-rod.

Figure 19 shows a compressor ramp-up with test-bench variable speed drive maximum slope (15 RPM/s corresponding to 185 RPM/s on compressor), to simulate a direct on line startup. Machine reaches 100% speed in 95 seconds.

This startup procedure is the one closest to thermo-mechanical simulation shown in Figure 16. Three phases are clearly detected during ramp-up:

1) tie-rod tension decreases since acceleration results in impellers' axial shrinkage due to centrifugal loads;
2) impellers' temperature increases first, leading to tie-rod elongation;
3) tie-rod temperature increases later; the subsequent elongation results in axial load reduction.

Those three phases were correctly captured by thermo-mechanical simulation.

Figure 20 represents a detail of tie-rod axial load during compressor shutdown. Also in this case it is possible to identify three phases:

1) tie-rod tension increases since absence of centrifugal load leads impellers to recover original axial length;
2) impellers' temperature decreases first, leading to tie-rod shortening thus axial load reduction;
3) tie-rod temperature decreases later; the subsequent shortening results in axial load increase which eventually leads to preload recovery.

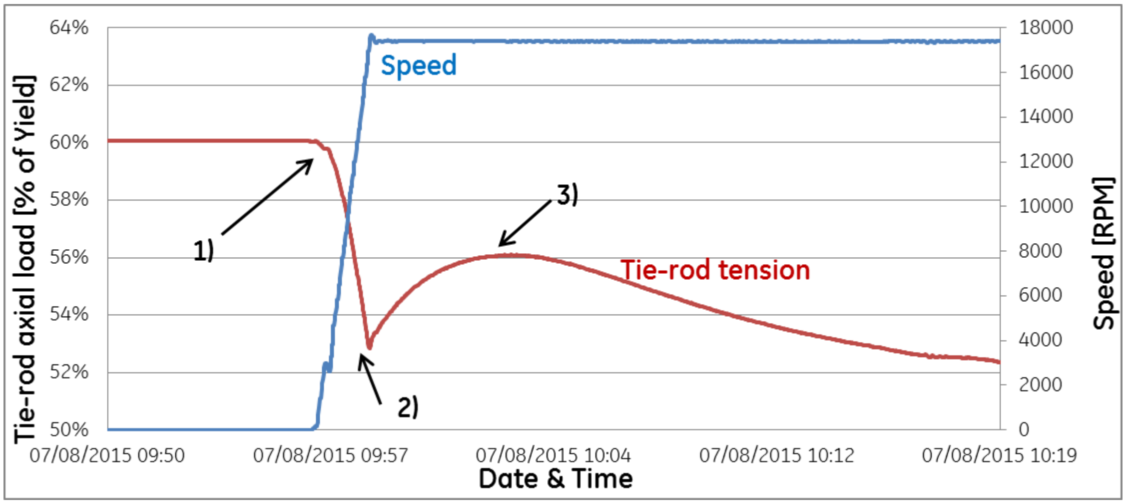

Figure 19: Tie-rod tension during direct on line startup

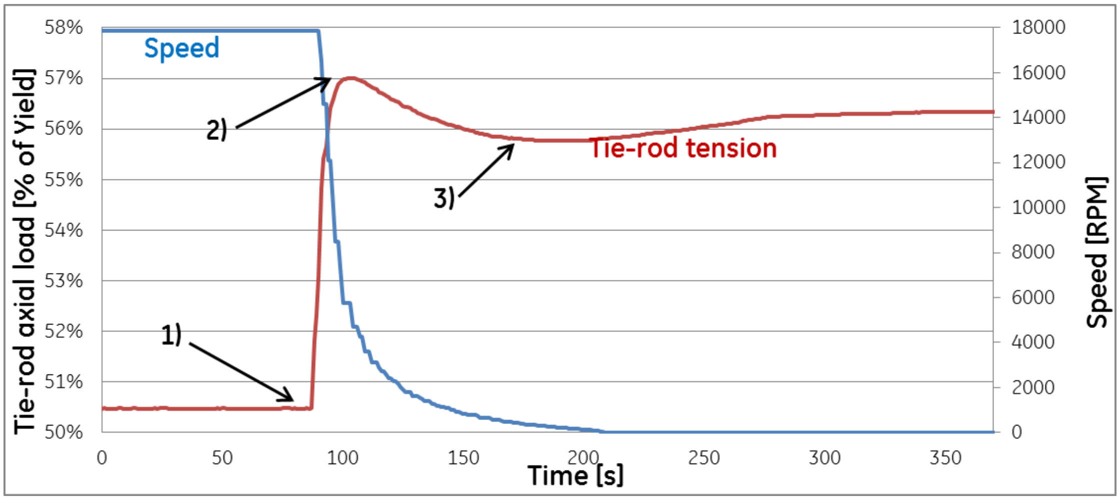

Figure 20: Tie-rod tension during shutdown

Tie-rod dynamic behavior has also been addressed. The tie-rod has been designed to have a subcritical behavior (revolution speed well below tie-rod bending frequency), to avoid excitation of its first natural frequency due to 1 X REV excitation. The tie-rod natural frequency is slightly higher than 500 Hz. It has been observed that almost no dynamic component is present on strain gages for the tie-rod natural frequency during steady state operation; the component with highest amplitude is the 1X REV, that reaches ≈1% of initial preload. Also in this case, this dynamic excitation does not represent any concern in terms of fatigue life.

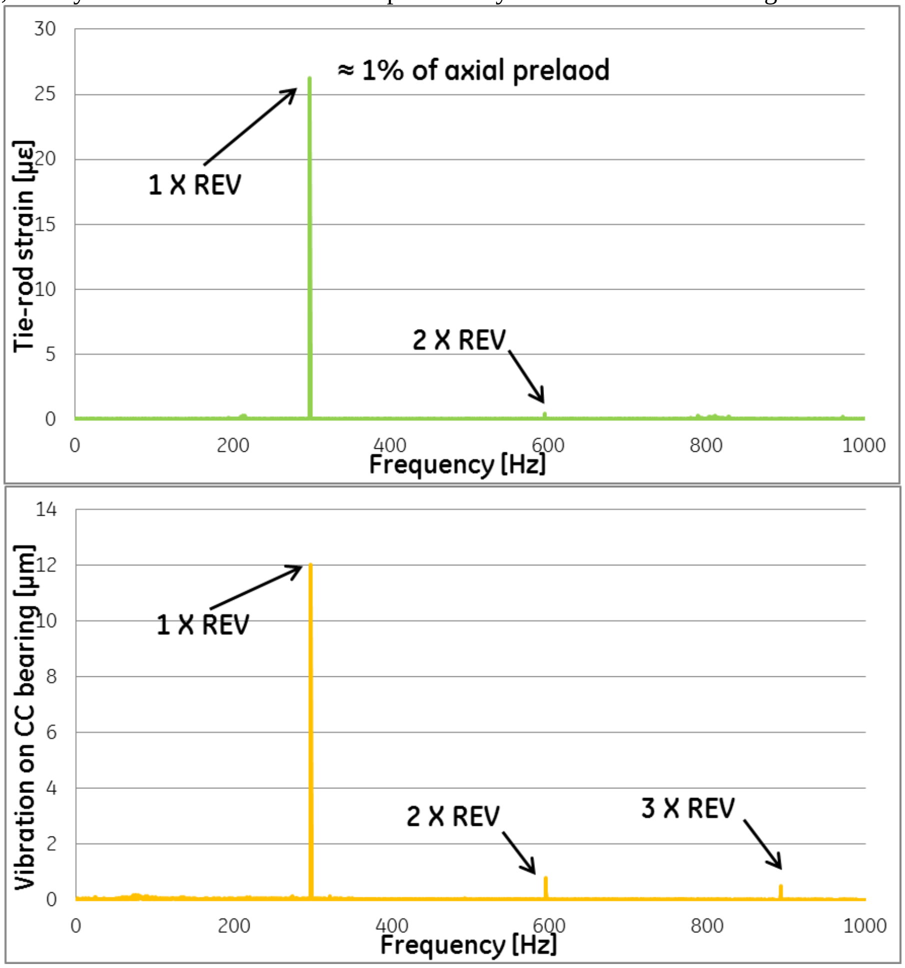

Figure 21: Tie-rod dynamic behavior

Aeromechanics

A detailed analysis of aeromechanical test is included in Toni et al.; it is interesting however to summarize the key results that validate compressor design obtained using strain gages on open impellers (two sectors instrumented on each stage):

- Flowrate sweep in compressor curves have shown maximum response in choking conditions;
- Modes' frequency are not significantly affected by process gas conditions (gas density and flowrate);
- Response level shows a clear trend with inlet gas density: Figure 22 shows for one of the observed modes that variability of test conditions led to identify an asymptotic response.

During the whole compressor test, the response level never exceeded the limits defined by internal mechanical criteria; the asymptote in response with inlet density allows concluding that even in increased suction pressure conditions, the response level will still be well below the maximum allowable levels.

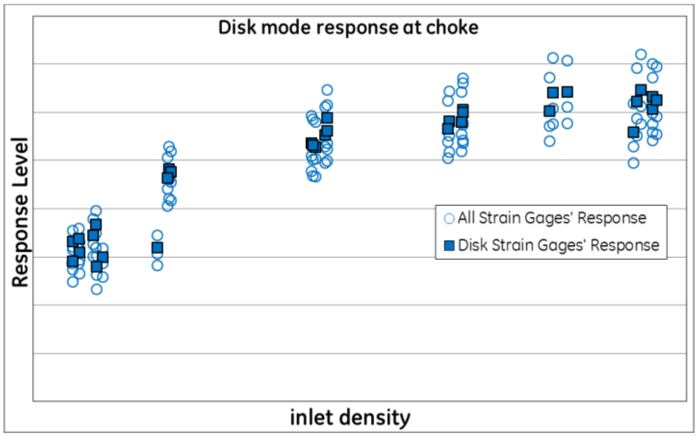

Figure 22: Disk mode response at choke vs. inlet gas density

Rotordynamics

Test results related to component level validation test (e.g. ping test and rotor balancing) have been detailed in Vannini (2014).

Compressor test has shown good agreement with rotordynamic predictions in terms of rotor modes frequency and damping: a dedicated Operational Modal Analysis (OMA) campaign has been performed to evaluate modal parameters for different compressor operating conditions.

OMA technique was formerly applied and validated at the Authors' Company, see Guglielmo et al.

Several tests have also been performed to assess rotordynamic behavior in conditions representative of plant operation:

- Fast (direct on line) startup;
- Emergency shutdown from compressor surge control line;
- Compressor hot restart sensitivity to different idle time;
- Trim balancing to minimize vibration amplitude at service speed.

Those tests confirmed a safe operation in whole compressor envelope.

It is worth to mention that compressor operating range is between third and fourth critical speed. As it is possible to see in the Bode plot below, only the first mode (approx. critical speed ≈ 4000 RPM) is clearly visible during both ramp-up and shutdown; second mode (approx. critical speed ≈ 8000 RPM) and third mode (approx. critical speed ≈ 11000 RPM) are critically damped and are also hardly visible on phase lag plot.

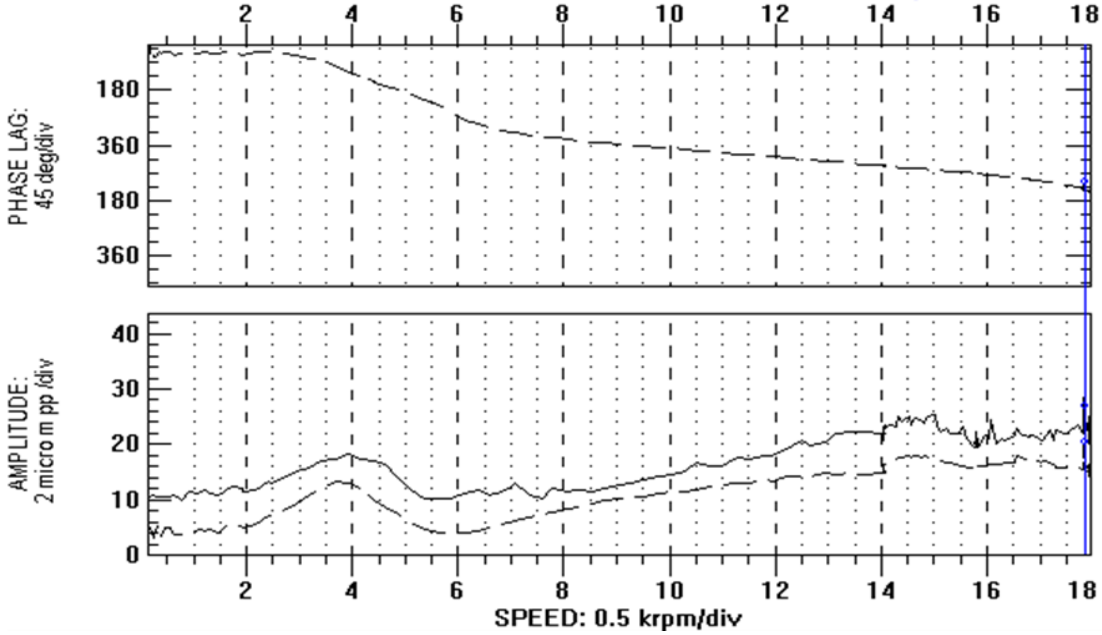

Figure 23: BODE plot during compressor ramp up

Vibration probes show a subsynchronous free vibration spectrum, confirming that all the three modes below operating range are well damped. The plot represents a ramp up to MCS with typical 10% speed steps. Then the speed is maintained equal to rated speed and operating point is moved from choke to surge. In the whole compressor envelope, no sub-synchronous vibration is visible.

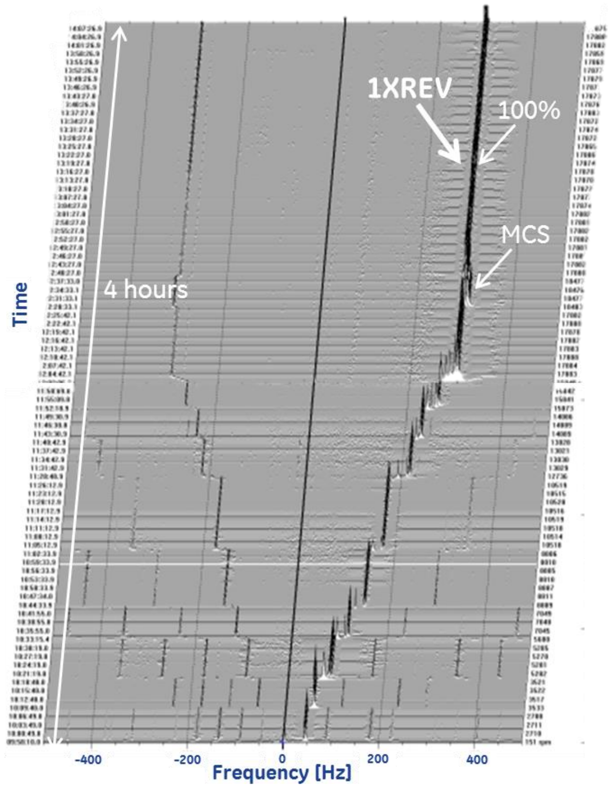

Figure 24: Waterfall plot for compressor radial vibration

Journal bearings' temperature always shows a stable behavior. Maximum pad temperature is 85°C at MCS;this temperature level is far below typical alarm values. The journal bearings' operation is thus safe. Rotordynamics and journal bearings behavior will be subject of a dedicated follow-up paper.

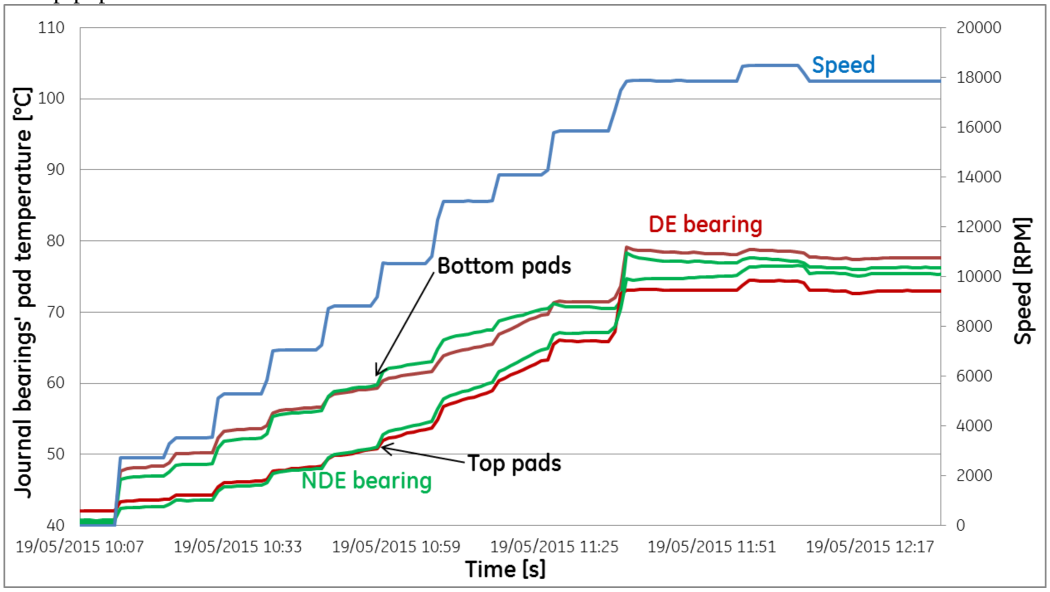

Figure 25: Journal bearings' pad temperature

CONCLUSIONS

The test activities have demonstrated the effectiveness of the High Pressure Ratio Compressor both for mechanical and performance aspects and opened new technological frontiers for compressors' manufacturing.

HPRC validation strategy was focused on verifying new technologies injected into the centrifugal compressor train, starting from component tests (new open impeller tests on dedicated single stage rig, DGS component test, Epicyclic gearbox mechanical running test) to a full prototype testing in actual intended environment (TRL4 as per API 17N).

HPRC prototype test was performed on a permanent test bench in Florence, accumulating 300+ hours of running and monitoring more than 100 sensors.

Thermodynamic performance test proved the performance level according to expected values. From a mechanical point of view, it has been proven that the system (bearings, DGS, tie-rod, open impellers in between bearings' configuration, thrust balancing scheme) is able to perform safely and within expected during both steady state and transient conditions.

From rotordynamic viewpoint the compressor performed as expected showing vibrations within the alarm levels during both run-up and run-down and a sub synchronous free behavior up to the maximum load tested. HPRC journal bearing temperatures were well within safe limits.

The final outcome is the validation of this new architecture for centrifugal compressors, which allows to increase the pressure ratio in a single casing. Considering the design case of the prototype, the required pressure ratio can be achieved with a single compressor casing instead of two, resulting in lower weight and footprint; at the same time, compressor reliability is increased through the reduction of equipment and related auxiliaries. These benefits become more evident for off-shore applications, where footprint and reliability represent main critical aspects.

In general, the test has demonstrated that the stacked rotor configuration for a high pressure centrifugal compressor that performs a typical upstream service can be used, resulting in a more cost-effective solution when compared with traditional configuration.

NOMENCLATURE

HPRC	High pressure ratio compressor
DGS	Dry gas seals
CC	Centrifugal compressor
DE	Drive end
NDE	Non drive end
HPRC	High Pressure Ratio Compressor
MRT	Mechanical running test
TRL	Technology Readiness Level

REFERENCES

Guglielmo A., Catanzaro M., Baldassarre L., Accorsi Miranda M., de Oliveira Zague L., Ishimoto L., Timbó Silva R., *Operational Modal Analysis Application For The Measure of Logarithmic Decrement In Centrifugal Compressor*, Proceeding of 44th Turbomachinery & 31st Pump Users Symposia, Houston TX, September 2015.

Maragioglio G., Vannini G., Buckley S., Bradley P., *Design Validation Of High Speed Ratio Epicyclic Gear Technology In Compression Systems*, Proceedings of 1st Asia Turbomachinery & Pump Symposium, Singapore, 22-25 February 2016.

Toni L., F. Moyroud, G. Gatta, D. Rubino, A. Guglielmo; *"Aero-Damping Measurements and Computation in a Full-Scale Multistage Centrifugal Compressor"*, Proceeding of 45th Turbomachinery & 32nd Pump Users Symposia, Houston TX, September 2016.

Vannini G., *Rotordynamic Validation of an Ultra High Speed Multistage Centrifugal Compressor Stacked Rotor*, Proceedings of ASME IGTI TurboExpo, Dusseldorf, June 2014.

ACKNOWLEDGEMENTS

Authors would like to thank all the groups that cooperated for the design and test of prototype; in particular: Systems Engineering and Advanced Technology, Advanced Train integration, Auxiliaries' systems design and New Product Introduction that carried on the design of the whole compression train, Oil & Gas Technology Laboratory (OGTL), that led the special instrumentation design and installation, Manufacturing and String test dept. for the test execution.