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PROCESS CONTROL OPTIMIZATION FOR CENTRIFUGAL COMPRESSORS ON ACTIVE MAGNETIC BEARINGS

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ABSTRACT

During the testing activities of a subsea prototype compression station in Norway, which included full load operation of an integrated vertical motocompressor levitated by active magnetic bearings (AMBs), relevant efforts were taken to opportunely tune the process control dynamics so to increase machine protection during transient operations. For such reason compressor shutdown procedure has been modified after the first testing phase in order to improve machine reliability and availability, by opportunely acting on diverse system parameters.

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Process control systems for traditional centrifugal compressors supported by oil bearings are always designed in order to minimize (or even avoid) machine operation in the surge region also for transient operations, while it is less usual to take precautions against operation in choking condition.
For a centrifugal compressor on AMBs however, it is necessary to properly take into account the process dynamics and the limited AMB's capacity to counteract external loads in the diverse machine operative modes.
In the following sections, the machine configuration and peculiarities will be described first. Then the thrust balancing for vertical integrated motocompressor will be described. Finally, both the original and optimized process control logic for the machine will be described.

INTRODUCTION
In late 2006 Authors' Company was awarded a contract for the motocompressor for the Ormen Lange Subsea Compression Pilot. This machine represents the first subsea motocompressor unit ever built in a fully marinized version and tested in a water pit. The unit was developed specifically for the project at the customer's site, which is the subsea filed for Ormen Lange reservoir.
The prototype unit is a 12.5MW (16763HP) integrated motocompressor, which runs up to 10668 RPM in a vertical configuration with the following features:

- Single casing
- High-speed motor rigidly coupled to a multistage centrifugal compressor (3 journal bearings shaft line)
- Canned Active Magnetic Bearings (AMBs)
- Process gas used as cooling fluid for the Electric Motor
- Internal separation system (to protect the bearings and the electric motor from the intrusion of solid and liquid materials)
- Fully marinized AMBs control system

A 3D view of the machine cross section is shown here below.

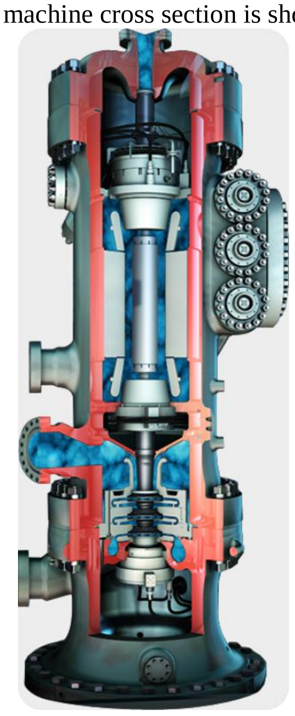

Figure 1: Compressor 3D cross section

The integrated motocompressor has been designed considering these main drivers:

- design of components has been analyzed to minimize probability of maintenance operation for a pre-defined time interval;
- the casing is designed to allow re-bundling of compressor with different size and number of impellers to adapt to change in operating conditions;
- the unit is designed to handle a defined quantity of liquid which may enter or condensate within the unit.

The electric motor is located topside and it is housed in a cast stainless steel casing, which provides interfaces for the high voltage connectors. The rotor is a single shaft line with a rigid Hirth coupling between motor and compressor shaft and is supported by three radial AMBs.
The unit is quite compact with respect to the power rating: 6m (19.7ft) in height and 2m (6.6ft) in diameter, with an overall weight of 55t (121'254lb) (not including the external seawater cooler).
The three-journal bearing rotor was considered a very robust solution for smooth rotordynamics associated with a high rotational speed. In fact, three rotor modes are crossed in this configuration but they are all critically damped. This is possible through the AMBs technology, which take advantage of the absence of the gravity load, saving load capacity for the lateral dynamics. For future reference, the AMBs are named #1, #2, #3 moving from the top to the bottom (e.g. AMB#3 is the compressor bearing).
The axial magnetic bearing is located on the top and it is a double effect bearing, since the global axial load can be reversed during operation.
The 7-axis control loop is implemented through a Digital Signal Processor at 13.8KHz, while the power electronics are based on the supplier's standard 300/30 (300 volt / 30 A).
The reason for a 7-axis control system is that the machine is equipped with three radial and one axial bearing (totaling 7 axes: 2 X 3 radial bearings and 1 X 1 axial bearing).
An internal Profinet network, a sort of real-time Ethernet communication protocol, guarantees the high throughput data exchange between the devices that are managing the raw data and the MBCSI (Magnetic Bearing Control System Interface), which is collecting all data and transferring it to the topside station.
The AMB control system is divided into two main parts:

- embedded control system, which will be installed in the subsea environment
- remote control system; which will be installed in the control room.

The embedded part is an assembly of electronic devices that are mainly used to implement the control loop, protection of the machine and the interface to the remote systems.
The remote part, based on an industrial PC and dedicated software, is used specifically for the remote tuning of the machine and remote monitoring that continuously collects and stores the data analyzed by the embedded system; the stored data can be post-processed and used for diagnostic purposes.
The AMBs are all canned, which means that the stator parts are completely enveloped with stainless steel protective cans. This technology was qualified during the project and the authors believe it is a big plus for the subsea compression application.

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The machine comprises an auxiliary bearing system, that is the protection system that engages as the AMBs are no more able to maintain the rotor position on the control set-point (E.G. loss of feeding power, or external forces exceeding bearing static or dynamic capacity); the auxiliary bearing system avoids rubbing of rotor surfaces with the machine stator parts. The condition of rotor touching the auxiliary bearings will be referred to as "landing" in the next sections.
On the electric motor side, state-of-the-art technology in terms of high speed and process cooled motors was used. The motor was integrated in the motocompressor environment and made as robust as possible and insensitive to potential process upset conditions. For instance, the cooling gas is filtered and then maintained in a closed loop, the casing material is made of stainless steel to avoid rust and deposits on the magnetized parts, and the layout is vertical to allow drainage of liquids.
The unit allows for the collection of liquid slugs on the bottom sump which capacity is 500 liters (132.1gallon).
The compressor is conceived as a cartridge, which can be removed from the electric motor section if a re-bundle is needed.
The maximum number of impellers is dictated by the available space in the suction plenum and by a compromise with the rotordynamic behavior. The compression service for the current application is fulfilled with a three-wheel rotor.
The compressor duty is based on many operating points, which represent the evolution of the gas reservoir life. The delivery pressure is always constant at 140bar (2031psi) while the suction pressure decreases over time: the starting suction pressure is just below 140bar and the target final plateau value is 80bar (1160.3psi). This means the compressor starts to operate at its minimum rotational speed and increase the speed along the years. For this reason the motocompressor operating range is very broad with a minimum operating speed (mos) of 3048rpm and a Maximum Continuous Speed (MCS) of 10668rpm.
Finally, this solution is fully marinized and ready for submerged testing.

THRUST BALANCING & AUXILIARY BEARINGS

Thrust on rotor of integrated motocompressor is the result of three different terms, which are evaluated separately:

- Motor-compressor rotor weight;
- Thrust due to differential pressure acting on motor surfaces because of cooling flows, which is evaluated through a detailed secondary flows model;
- Thrust on compressor impellers and balance drum, evaluated with the standard approach for centrifugal compressors.

For a centrifugal compressor, the main thrust terms are:

- Primary thrust: thrust acting on the impeller because of the pressure differential across each stage; this term is usually directed towards suction, and is maximum at compressor surge, while it is virtually zero in choking conditions;
- Momentum thrust: this term is related to fluid deviation which happens in rotor parts: in each impeller, the gas flow is deviated from axial direction (impeller suction) to radial direction (impeller discharge) causing a variation of momentum in process gas that results in a thrust term towards discharge side. This term is maximum in choking conditions, since it increases with machine flowrate.
- Balance drum thrust: it is a force term towards machine discharge due to pressure differential across machine balance drum. This term is maximum close to surge line, since machine pressure ratio is maximum in that region, while it reduced towards zero when operating in choking conditions.

The procedure for thrust evaluation of the integrated machine is not in the scope of the present paper; the resulting thrust curves for the design inlet pressure (80 barA) are shown in figure below.

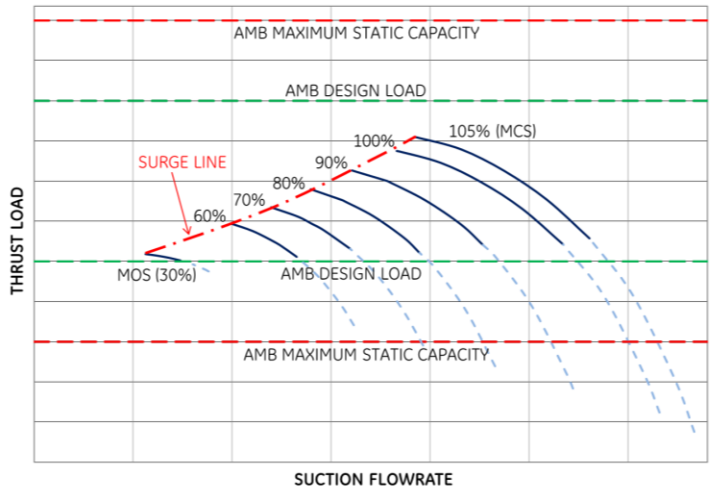

Figure 2: Compressor thrust curves

The overall thrust for the integrated machine is plot as a function of machine flow rate. Several curves are drawn for different operating speeds. The solid curves are plotted considering the acceptable operating range of the machine and refer to points where the machine can continuously operate without any reliability concern.
On the left side the curves are limited by the surge limit line; operating points are then considered as being acceptable until the efficiency reduction observed for high flowrate is not related to possible flow instabilities.
Nevertheless, in particular process conditions, the operating point may move outside the acceptable solid curves and fall into the dashed curves zone for a limited period of time. Since the machine pressure ratio in the dashed region falls rapidly towards 1, the compressor may operate there only for very low circuit resistance.
As specified in the previous section, the motocompressor thrust is balanced through a symmetric double-effect magnetic thrust bearing. Thrust bearing is symmetric since it has same capacity on both sides.
The load capacity for AMBs is characterized by the following two different aspects:

- The *static capacity* (or peak load capacity), referring to the maximum static load that the magnetic bearing can handle;
- The *dynamic capacity* that can be defined considering the maximum load rate that the AMB can provide.

The dynamic capacity is a parameter which is usually not considered for oil bearings: as a matter of fact, the active magnetic bearings are balancing external loads through

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magnetic forces which magnitude is given by feeding currents on AMBs' coils. If the load changes suddenly, the system may not be able to modify the bearing current with the same speed, resulting in an unbalance between magnetic bearing force and external force which results in rotor displacements that may lead to short touch or even longer landings on the auxiliary bearings.
In addition, it must be noted that, differently from oil bearings, AMBs are not able to provide any overload capacity, even for a short time: therefore, if the process force variations induced on the rotor exceed AMB capacity, a landing event may occur.
The auxiliary bearings' clearance is sized in a way that the rotor, when is no more sustained by AMBs, will land on auxiliary bearing without touching any other stator part. This prevents both rotor and stator parts of compressor from being damaged during the landing event.
As matter of fact the landing bearing system is however a consumable machine apparatus and is designed to sustain only a finite number of landings.
Considering the expected mean time between maintenance activities for a subsea compressor station (that is in the order of some years) it is therefore of key importance to avoid AMBs overload capacity during operation, so to limit the use of auxiliary bearings only for the unlikely case of electrical feeding system failure.
By referring to Figure 2, it is thus necessary to avoid the operation of compressor in the region where thrust exceeds the *maximum static capacity*.
In addition, sudden thrust variations have also to be avoided, in order not to overcome the AMB *dynamic capacity* during operation.

TESTING FACILITY AND CONDITIONS

After extensive testing in the authors' company facility (Vannini et al., 2011), the motocompressor unit has been shipped to the customers's facility for an Extended Factory acceptance test. The testing activity started in February 2012 and comprised a large variety of tests mainly focused on machine integrity, compressor performance, operability and response to upset conditions. In particular some of the most important tests which has been performed are:

- AMBs tuning and setting during machine rotation;
- Vibration monitoring for all the operating speed and suction pressure;
- Motorcompressor test with dry gas;
- Motorcompressor test with humid gas in the loop;
- Performance test in the entire compressor performance map:
    - Speed from 30448 RPM (mos) to 10668 RPM (MCS);
    - Suction pressure from 25barA up to 98.5 barA;
    - Flow from surge line to overflow conditions;
- Maximum continuous speed test;
- Maximum compressor power;
- Loss of electrical power;
- Low suction pressure operation;
- Verification of actual surge line;
- Slug simulation;
- Process excursion;
- Motor cooler testing;
- Repeated Start up sequences
- Shutdown sequences, initiated by different station equipment
- Start up after prolonged stand still.

Figure 3: Extended Factory Acceptance Test in the pit at customer facility in Nyhamna, Norway

A full report for testing activity is outside of this paper's scope; results of EFAT activity have been summarized in Bigi et al., 2013.
What is important to remark is that the scope of the test was to operate the compressor in the actual conditions it will operate subsea; the gas loop was designed including the usual anti-surge system installed for centrifugal compressors and the whole compressor envelope was tested to extensively assess the motocompressor operability.

PROCESS CONTROL FOR INTEGRATED MOTOCOMPRESSOR

Process control systems for traditional centrifugal compressors supported by oil bearings are always designed in order to minimize (or even avoid) machine operation in the surge region both for continuous and transient conditions. Considering Figure 2, the operation on the left part of thrust curves is thus limited by the compressor protection against surge conditions. As a matter of fact, in surge conditions, the compressor behavior is no more well predictable and sudden flowrate variations (up to reverse flow conditions) are accompanied by pressure ratio fluctuations and high amplitude rotor vibrations. All these phenomena represent an undesired operating condition both for the machine and the plant operability; an antisurge system is thus always implemented through a recycle valve which opening leads to compressor suction flow increase that moves compressor operating point towards the high flow region.
It is much less usual to take precautions against operation in maximum flow conditions, known as "choke". As a matter of fact, most of centrifugal compressors can safely operate in choking conditions without relevant concerns.
By looking again to Figure 2, it is however possible to see that operation in maximum flow region results in overcoming the maximum static capacity for the axial magnetic bearing. Continuous operation in that region, at least for this prototype, is thus avoided during steady operation.
However, the system characteristics and the protection settings led to exploration of the high thrust region during some of the machine's shutdown sequences:

- The overall circuit has a low resistance, that means that, by fully opening the antisurge valve, it is possible

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to operate the compressor in the extreme right part of performance (and thrust) curves;

- The machine shutdown procedure was implemented to move the operating point at shut-down initiation in a safe condition, far from surge area, where unfortunately the magnitude of the thrust load is increasing. Then, the motor stop was initiated.

Indeed, when the process control triggered the compressor shut down, the anti-surge valve was the first to fast open; then, after approximately two seconds, the electrical motor trip was triggered and speed started to drop down (Fig 4). As already explained, the main aim of this logic was to move the compressor operating point far from surge line before triggering the motor shutdown; by doing speed, during the speed ramp down, the risk of entering the surge region is minimized.

This process control logic resulted in a rapid increase of machine flow rate, with the operating point moving into the dashed curve region and eventually overcoming the axial AMB static capacity: as a result the rotor landed on the auxiliary bearing for more than one second (Fig. 5).

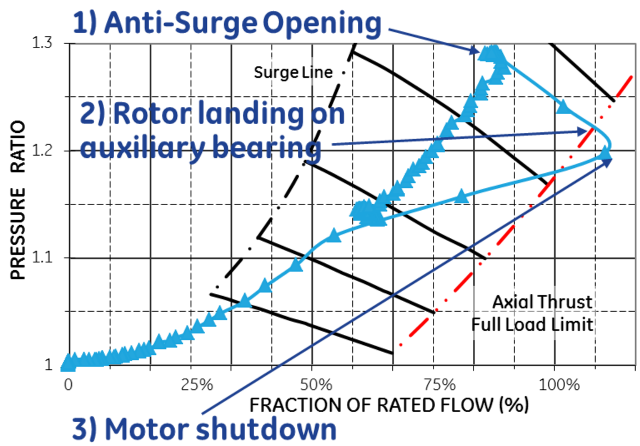

Figure 4: Compressor operating point during shutdown

Figure 5 represents the rotor axial position during the process shutdown procedure; a careful analysis of this quantity gives relevant information about the compressor operation during the shutdown.

The first thing to be noted is the first peak (with the tag #1, Anti-surge Opening): the sudden flowrate increase due to the antisurge valve opening results in a temporary overcome of axial AMB dynamic capacity: the AMB is no more able to maintain the target axial position (the 0 in the y-axis) and the rotor suddenly moves towards compressor discharge side, because of the rapidly increasing momentum thrust. The axial AMB is however able to fast recover the rotor position and bring it back to target.

As shown in Figure 4 the increase in flowrate leads to cross the maximum AMB static capacity: the axial AMB is thus no more able to maintain the target position and the rotor lands on the auxiliary bearing. After approximately two seconds from antisurge valve opening, the motor shutdown is triggered. The rotor has been however leaning on auxiliary bearing for more than 1 second, resulting in a significant damage of landing bearing after several trip sequences.

In addition, after motor shutdown, the compressor thrust reduces below the AMB static capacity. The AMB is thus able to re-levitate the rotor. The proportional-integral controller for axial AMB has however accumulated a large error after more than one second of landing: the axial force from AMB thus remains equal to maximum value for some tenths of a second, pushing the rotor on the other side of auxiliary bearing. Finally, the control system recovers the target position and the machine stops with rotor in full levitation.

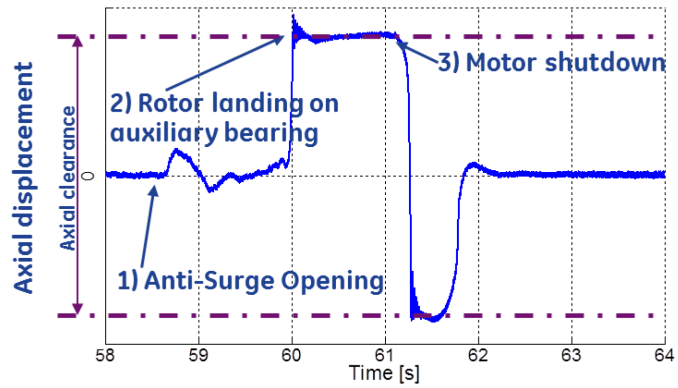

Figure 5: Rotor axial displacement during shutdown

Figure 6 shows the radial vibration on radial bearing #3 (as a portion of radial clearance on auxiliary bearing). Note that radial AMB#3 is the one closer to the compressor thus the effects of compressor dynamics are more evident.

As it's possible to see from the inner part of the orbit, the vibration is usually within 30% of radial clearance. During the shutdown, because of the dynamic upset, some increased vibration is present, which is however 60% of radial clearance as a maximum.

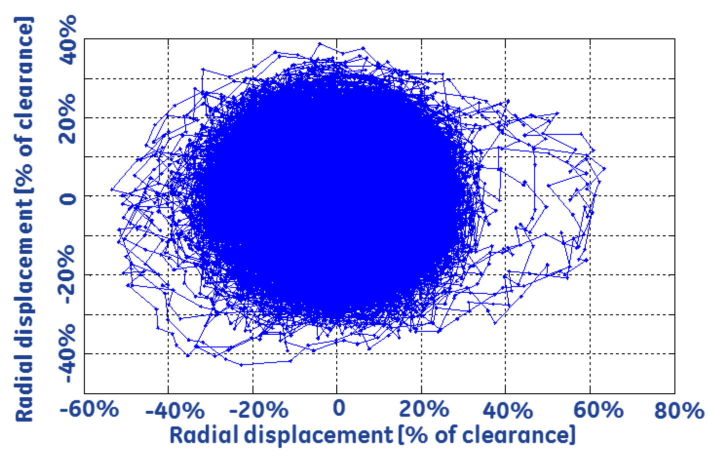

Figure 6: Rotor radial vibration on AMB#3 during shutdown

During the first testing phase of the subsea station, this process shut down logic was triggered several times, overcoming the maximum allowed number of landings, resulting in an excessive wear of the auxiliary bearing carrying the axial loads.

This excessive wear finally led to an excessive auxiliary bearing clearance which prevented the capability to levitate the shaft after a rotor de-levitation.

The testing activities were then stopped in order to replace the worn auxiliary bearing. Above presented study, obtained by test data post processing, demonstrated that the reason for auxiliary bearing excessive wear was originated by the adopted specific

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shutdown procedure, forcing the compressor to operate in the dashed region of compressor curves, where AMB maximum static capacity is overcome.

MODIFIED CONTROL LOGIC FOR AMB SUPPORTED MACHINE

The trip procedure was thus modified in the second testing phase to avoid the repetition of these process originated landing events.

In the new procedure, the opening command for anti-surge valve is only effective when the machine operating point is approaching the surge control line on the left part of the compressor map, regardless of the electrical motor trip initiation. Depending on the position of the operating point before emergency shutdown initiation, when the motor trip is triggered it might happen that the machine operating point moves in the surge region for very short time, since the anti-surge valve opening will be effective only after a delay time that is dependent from both valve response and circuit volumes (Figure 7).

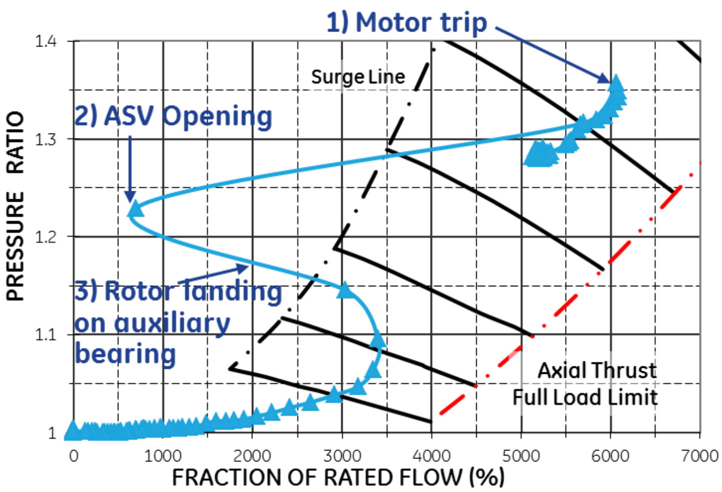

Figure 7: Operating point during shutdown (new procedure)

In this region, the absolute value of thrust is far below the magnetic bearing capacity, but the fast fluctuations in flow rate and machine pressure ratio (both resulting in fast variations of axial thrust) may lead to overcome for short instants the magnetic bearing dynamic capacity.

This condition results in large rotor axial movements that, in the case shown in Figure 8, lead to short landing of rotor on auxiliary bearing. Time duration is however far lower than previously experienced landing in the choke zone and the effect on auxiliary bearing's life can be considered negligible.

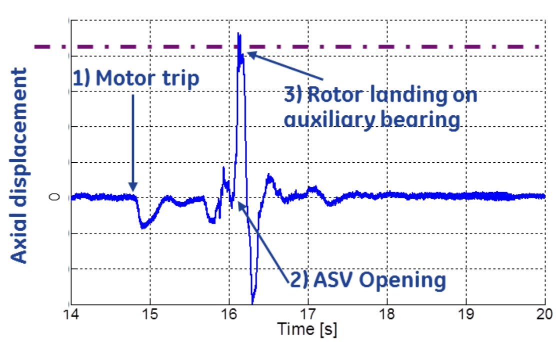

Figure 8: Rotor axial displacement during shutdown

Figure 9 shows the radial vibration on bearing #3. As it's possible to see, the occurrence of surge conditions result in high amplitude vibration which are close or equal to nominal radial clearance on auxiliary bearing. Also in this case, the touch (if actually present) is really fast and no prolonged landing is observed. The effect on auxiliary bearing's life is thus negligible.

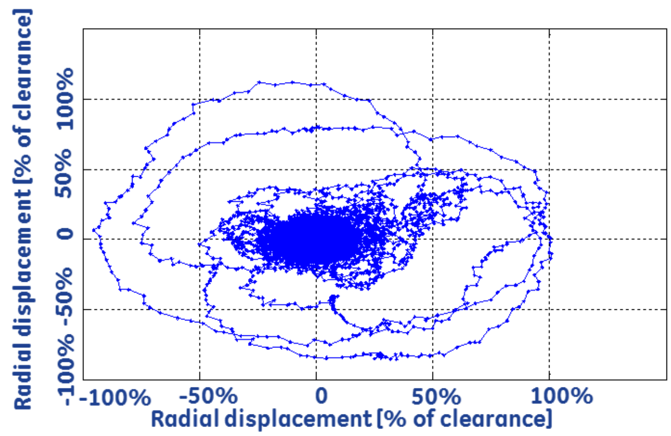

Figure 9: Rotor radial vibration on AMB#3 during shutdown

CONCLUSIONS

Concluding, control logic criteria to protect the centrifugal compressor from undesired process upsets, such as surge, have to be reassessed for machines on AMBs. For the mentioned testing campaign i.e. the initial adopted shutdown procedure, that was intended to keep the operating point far from surge region, led to an overcome of axial AMB static capacity which resulted in prolonged landings on auxiliary bearing. The final effect was an excessive wear of the axial auxiliary bearing which prevented the capability to levitate the shaft after a rotor de-levitation.

The control logic for emergency shutdown has thus been modified to take into account the axial AMB capability and to avoid the overcoming of axial AMB static capacity: no landing which resulted to be life-consuming for the auxiliary bearing system has then been experienced after this modification.

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Our gratitude goes also to our customers (AkerSolutions, Statoil and Shell) for giving us the possibility to go through this project and build together a strong know-how on Subsea compression.

NOMENCLATURE

AMBs: Active magnetic bearings;
Landing: event defined by the rotor touching the auxiliary bearing system;
ESD: Emergency shutdown;
FAT: Factory acceptance test;
EFAT: Extended factory acceptance test;

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