



## DEVELOPING MAGNETIC BEARINGS FOR SUBSEA OCEANIC ENVIRONMENTS

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### ABSTRACT

The article deals with the set-up environment especially designed and implemented to improve vibration attenuation of rotating machines installed on subsea environments. Active Magnetic Bearing Control (AMBC), including vibration attenuation or suppression associated to the passive magnetic bearings (PMB) design for subsea oceanic gas compressors and motors is the aim of the article. Gas production forecast and production was the motivation for subsea gas compression using High-speed Variable Speed Drives (VSD) in order to keep the actual gas production volume at the initially forecasted levels using underwater compressors and motors equipped with active magnetic bearings (AMB).

Robust control algorithms for AMBC using the most efficient technologies to ensure performance are demanded. With the developed test-rig, the improvement of vibration control algorithms to be applied on subsea motors and compressors equipped with hybrid magnetic bearings (HMB), (consists of active and passive magnetic bearings) is the objective of the work. The implemented test-rig provides us a test environment which can be modified to support different scenarios with regard to structural components, loads and severe operating conditions.

**Key words:** Active magnetic bearings, Passive magnetic bearings, Subsea rotating machines, Vibration control.

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## SUBSEA TECHNOLOGICAL BACKGROUND

Producing oil and gas from reservoirs located at long distances from land is a costly proposition that presents many challenges to offshore operators. Going subsea with long reach from shore or remote platform can be a very cost-efficient solution as it may eliminate the need for a fixed or floating topside installation. To help operators achieve efficient electric power supply and control at long step-out distances, some developers and manufacturers offers technical solutions of highly developed electrical products and associated services tailored for subsea production applications.

The objective of subsea gas exploitation using subsea compressors and related equipment is to minimize big investments and operational expenses in new off-shore production units by utilizing existing production units as a hub and have a significant part of the gas recompression equipment installed on the sea-bed (for boosting in the wellhead area) by means of long step-out cables. To achieve proposed objectives several technical challenges will be addressed:

High-speed VSD, long step-out power system, a very long umbilical cable between frequency converter and machine/compressor, power rating equipment for step-out system so far, rated for 6-8 MW compressor power and motor supply frequencies of up to 200Hz., and finally the motor and compressor equipped with active or hybrid magnetic bearings (R. Fantoft, 2005) (Raad, R.O.; Henriksen, T.; Raphael, H.B.; Hadler-Jacobsen, A, 1996) (Pyrhonen J., Nerg J., Kurronen P., Lauber U., 2008).

Some of the subsea solutions are based on the standard frequency converters and special designed transformers (Moreno, V. and Pigazo, A., 2007), adapted to meet the stringent requirements of topside or subsea installation.

Many specialised manufacturers are introducing subsea electrical solutions based on the gained experience since they have been involved in the development of subsea electrical equipment for many years. Feasibility studies on subsea components began two decades ago and the first commercial subsea transformer was delivered in the last decade. Since then, such manufacturers has delivered variable speed drive systems and transformers to some of the largest and most advanced offshore developments in the world.

With long experience and in-depth expertise, they offer solutions ranging from straightforward equipment supply, such as redundant monitoring systems (Ferreiro K., Haro, M. and Calvo, J. L., 2009), to full project management of the total subsea electrification network -from fixed or floating production units to subsea pipelines, wellheads or even downhole. With each delivery, the customers gain a highly qualified service and support partner to ensure that the products perform to the customers' expectations throughout the lifecycle of the field.

### The key for subsea applications

For subsea electrical consumption applications, state of the art technology provide topside variable speed drives and transformers designed to extend step-out distances

and reduce subsea cabling. Well proven topside electrification systems ensure reliable, efficient power supply to subsea power consumers and provide substantial cost savings by reducing subsea component and cabling requirements.

### *Topside Variable Speed Drives*

Based on the market-leading AC drives, topside drive systems are air or liquid-cooled and feature high robustness in a compact size. Depending on load characteristics, the topside drive system provides a step-out distance of up to 47 km. All components must be qualified and meet international standards and marine classification requirements. Typical selection of frequency converters includes:

- Drives for load up to 2 MW with 14 km reach-out distance, 11kV transmission voltage
- Drive for 2-4 MW load, 31 km reach-out distance, 25kV transmission voltage
- Drives for 8 MW load, 47 km reach-out distance, 36-52 kV transmission voltage

### *Topside Transformer*

Among the existing input transformers there are some of them that can be for 6, 12 or 24 pulse converter input. The step-up transformer is tuned for optimal voltage in the umbilical. The special developed topside transformer from is combining input and step-up transformer into one single tank. It is of a high temperature design. This gives a significant reduction of weight and volume compared to ordinary transformer solutions. They are delivered with an integrated earth fault monitoring system for the umbilical.

## **Design studies for subsea electrical systems**

Stimulating subsea oil and gas reservoirs effectively through boosting, injection and compression is critical for achieving stable production and extending the feasibility of the field.

For reservoirs with long step-out distances, powering the subsea equipment that performs these functions is a challenger task, requiring electrical equipment that is powerful, rugged and reliable.

Producing oil and gas from reservoirs located at long distances from land is a costly proposition that presents many challenges to offshore operators. Going subsea with long reach from shore or remote platform can be a very cost-efficient solution as it may eliminate the need for a fixed or floating topside installation. To help operators achieve efficient electric power supply and control at long step-out distances, some manufacturer's offers technical solutions comprising highly developed electrical products and associated services tailored for subsea production applications.

Conquering subsea gas resources with the use of cost effective technology requires being part of the world's first full-scale subsea gas compression equipment.

Some manufacturers have submitted subsea electrical equipment for full scale testing, analysis and qualification for a subsea compression application to boost gas production from the North Sea oil and gas fields. Project parameters include a tie-in distance of 47 km, and an 8MW compressor powered by a 200 Hz motor.

### The challenges for subsea compressors and motors set-up

Traditionally an electrical driven gas compressor package contains stand alone electrical motor, couplings, gear box and compressor(s) with shaft seals. A lubrication system is normally required for bearings in rotating parts. In the compact gas compressors external couplings, gearbox and shaft seals are removed, as shown in figure 14.

Magnetic bearings are used instead of lubricating bearings. The magnetic bearings can either be exposed to the transported gas or be separated from it by using a can. The electrical motor is running at the same speed as the compressor and the motor is cooled by pressurized gas alone or in combination with insulating liquid.

Rated motor speed for compact compressors can be up to 12 000 RPM. For a 2-pole motor this means a supply frequency slightly above 200 Hz. Motor windage is the highest motor loss component for the electrical motor in a compact gas compressor. It increases with speed and pressure. The loss in the long step-out power system does also increase with increased frequency. From an energy efficiency point of view, it is preferred to reduce the rated motor speed. Reduced speed will however increase the size of the compressor. In the technology qualification program, compressors with rated speed 9500 RPM and 11200 RPM have been tested. Operating speed is typically in the range of 30-100% of rated speed.

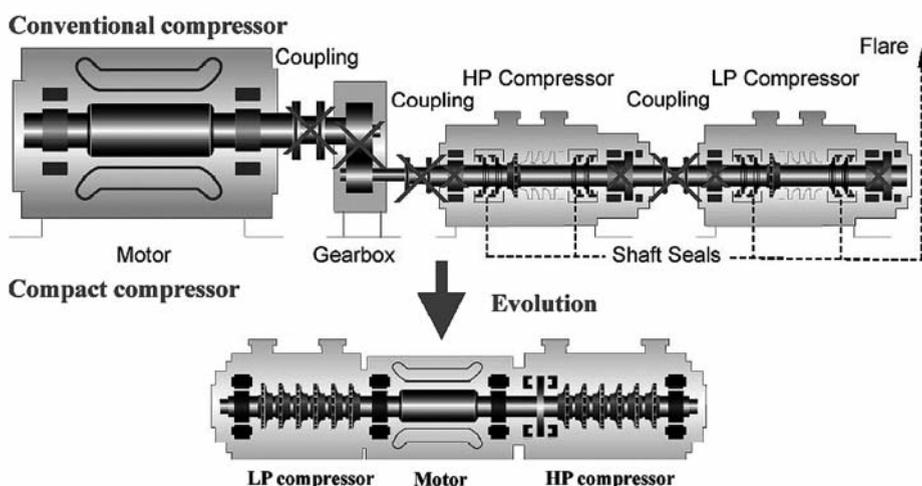


Fig. 1. The evolution of a conventional gas compression station to a compact subsea gas compression station.

Motor rotor in compact compressors can either be of solid metal design with or without a copper squirrel cage or a laminated rotor design with a copper squirrel cage.

From a rotor dynamical point of view, solid rotor is preferable. A solid rotor has a high degree of nonlinear magnetic property.

Motors that are cooled by pressurized well stream gas only, can have either conditioned gas, semi-conditioned gas or use the same gas that is going through the compressor. Compatibility between the cooling gas and the insulation material in motor and magnetic bearings is a critical success factor for the qualification of compact compressors on wet gas.

Motors that are cooled by pressurized gas and insulated liquid, separates the stator from the gas. Motor stator is then cooled by insulated liquid.

The pressure inside the motor will be slightly above gas compressor inlet pressure (any gas leakage will go from motor through labyrinths to the compressor).

Magnetic bearings have been considered as the very suitable solution for subsea applications as a lube oil system can then be omitted. In figure 1 (Gerald Scheuer et al. 2005) it is shown the evolution from a conventional gas compression station to a compact actual compression station equipped with active magnetic bearings for subsea applications.

Among the key technical items focused in the design of a gas compression module is a set of reliable components such as seals and bearings needing special attention.

Active magnetic bearings applied on motors and compressor requires also a position control system capable to support shaft unbalance and inherent vibration dynamics. Although passive magnetic bearings don't need position control, in practical applications vibration control to compensate unbalance and cavitations disturbances is required.

With the aim of improve reliability and efficiency on subsea magnetic bearings applications, next section is devoted to the description of the necessary basic equipment (test rig) to develop passive magnetic bearings and in section 3 vibration attenuation by means of a vibration control system is described.

## **PROPOSED PMB STRUCTURES.**

Magnetic bearings allow contact-free levitation. This is an attractive feature which offers some interesting advantages, such as no friction, no lubrication, low maintenance cost, long life, etc. In recent years, magnetic bearings have found applications in flywheels, pumps, compressor drives, and so on.

There are mainly two types of magnetic bearings: the active magnetic bearings and the passive magnetic bearings. The passive magnetic bearings don't need active control and extra input energy, so they are more compact and easy to use.

There are several ways to achieve passive magnetic bearings: superconductor, diamagnetic, eddy current, and permanent magnet. The simplest type of passive

magnetic bearing is only using permanent magnets. This type has the advantages of potential to miniaturization, high stiffness, reliability, and cost effectiveness. Although Earnshaw's theorem states that there is no stable and static configuration of levitating permanent magnets, permanent magnet can be used as either axial magnetic bearing or radial magnetic bearing. Actually, lots of practical magnetic bearings consist of active magnetic bearing as well as permanent magnetic bearing to obtain total stability and relatively low cost. Common configurations of permanent magnetic bearing are composed of two monolithic permanent magnetic rings with either axial magnetization or radial magnetization.

Magnetic bearings have been widely used due to their advantage of without mechanical wear and noise, which are beneficial to many applications such as subsea pumps and compressors with high speed and power. There are many different types of non-contact rotary machines. It is hopeful of designing a magnetically suspended motor with compact structure and simple control. A new structure of motor with PMB is introduced in this paper and is shown in figure 2.

Although one passive magnetic bearing is unstable, a stable PMB can be constructed by combination of a pair of two passive magnetic rings coupled in opposition

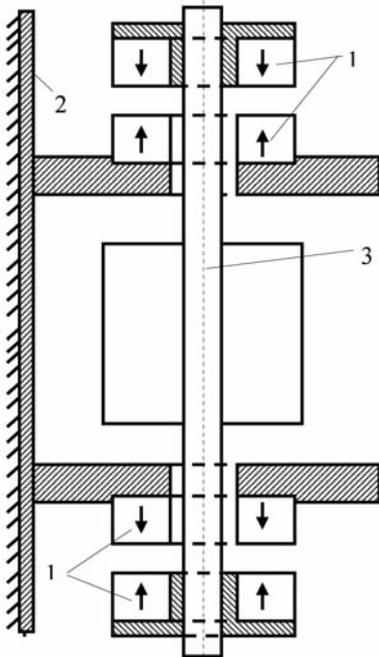


Fig. 2. Structure of a motor rotor equipped with the prototyped PMB. Axial magnetized magnetic rings, 1. Stator support, 2. Rotor body, 3.

between them, which yields the proposed prototype named the inverse PMB (IPMB). As shown in figures 2, 3 and 4 the axially polarized magnetic rings are assembled by pairs of rings in opposition, so that stability of the rotor is inherently ensured. The axial stability of the rotor can be obtained by the radial force of the passive magnetic bearings and by the instinctive behaviour of the IPMB rotor since the axial inverse magnetic forces try to keep the shaft centered with respect to the stator.

In figure 3, three stages axial IPMB is shown. The basic idea is to construct axial IPMB with the required magnetic stages for a particular application. In this prototype the supported load approaches 300 N, which is a considerable force in comparison with the magnetic mass. The axial force is then determined by the size of the magnetic mass of the magnetic rings and the number of magnetic stages.

The layout of a prototyped two stages axial passive magnetic bearing is shown in figure 4. A pair of such an IPMB is necessary at least for every shaft or rotor.

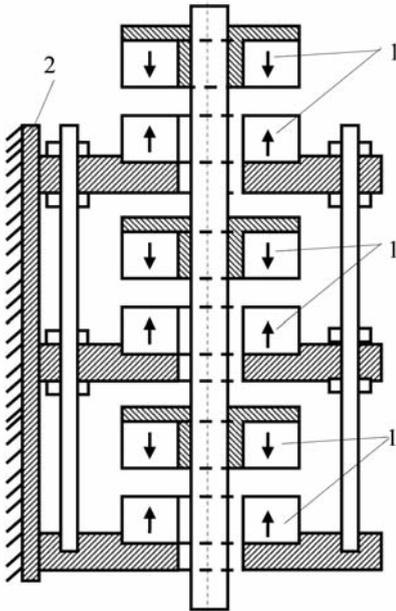


Fig. 3. Structure of a prototyped IPMB of three magnetic stages.

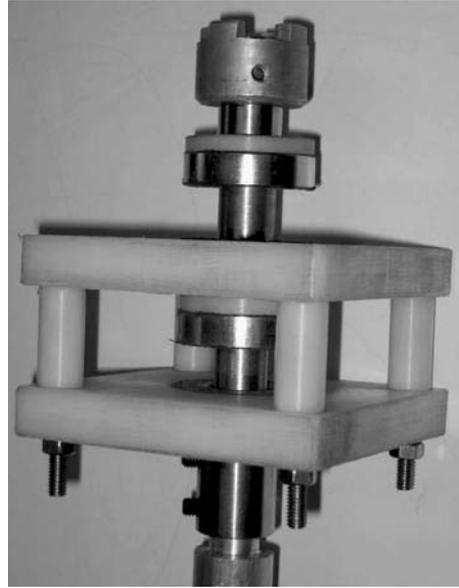


Fig. 4. Prototyped two stages axial passive magnetic bearing.

## VIBRATION CONTROL

### Introduction to Vibration Control

Motors, compressors, and turbines or pumps and blowers including all rotating machinery in general, is commonly used in process industry, including machining tools, power generation, as well as aircraft and marine propulsion among the most important industrial applications. Mass imbalance is commonly responsible for rotating machinery vibration. When the principal axis of inertia of the rotor is not coincident with its geometric axis imbalance occurs. Nevertheless there are some more causes for rotating machinery vibration such as operation near resonant frequencies or critical speeds.

There are two major categories in AVC techniques for rotating machinery:

- Direct active vibration control (DAVC) techniques in which directly apply a lateral control force to the rotor.
- Active balancing techniques which adjust the mass distribution of a mass redistribution actuator. Active balancing isn't under the scope of this work.

The control variable in DAVC techniques is a lateral force generated by a force actuator based on a magnetic bearing. The advantage of DAVC techniques is that the input control force to the system can be changed according to vibration characteristics.

By applying a fast changing lateral force to the rotating machinery, the total vibration, including the synchronous vibration, the transient free vibration, and other nonsynchronous vibration modes of the rotating machinery, can be attenuated or suppressed. The limitation of most force actuators is the maximum force they can provide. In high rotating speed, the imbalance-induced force could reach a very high level. As most force actuators cannot provide sufficient force to compensate for this imbalance-induced force, active balancing methods are well justified. Although active balancing methods can eliminate imbalance-induced synchronous vibration, they cannot suppress transient vibration and other nonsynchronous vibration.

(Maslen and Bielk, 1992) (R. Larsson, 1998) presented a stability model for flexible rotors with magnetic bearings. Besides the flexible rotor model itself, their model included the dynamics of the magnetic bearing and the sensor-actuator noncollocation. This model can be used for stability analysis and active vibration synthesis.

Most recently, an analytical imbalance response of the Jeffcott rotor with constant acceleration was developed by (Zhou and Shi, 2001). They concluded that a satisfactory solution quantitatively shows that the motion consists of three parts:

- the transient vibration at damped natural frequency,
- the synchronous vibration with the frequency of instantaneous rotating speed,
- and a suddenly occurring vibration at damped natural frequency.

Such mentioned technique provides physical insight into the imbalance-induced vibration of the rotor during acceleration. For this reason it can be used for the synthesis of AVC schemes.

For the synthesis of DAVC techniques, most it is common to use simplified low-order finite element models of the rotor system. Although the techniques developed can be extended to a high-order system theoretically, the computational load and consequently the signal-to-noise ratio will have to be higher. The DAVC techniques can be difficult to implement for the high-order system. Therefore, it is conveniently to use a reduced order models to approximate the high-order system models. Applied model reduction techniques have a specific impact on the performance of the DAVC schemes that must be considered if expected performance cannot be achieved.

### **AVC with Magnetic Actuators**

This section presents the test environment for active vibration control of rotating machinery. The principal idea is to control bending vibrations of a flexible rotor, supported by AMBs based on two sets of non-contacting electromagnetic actuators located at both shaft ends as shown in figure 5.

The test environment is composed of the following parts; a rotor test rig, two sets of magnetic actuators assembled to operate as both electromagnetic actuators and

AMBs, and a programmable control unit (C.R. Fuller, S.J. Elliot, and P.A. Nelson, 1996), (C.R. Knospe, et.al, 1997), (S.J. Elliot, 2001) to be applied on vibration attenuation or vibration suppression by means of feedback control applied to decrease the dynamic response of the rotor assumed as active magnetic dynamic damping. The main studies to be carried out on the described test rig deals with the dynamic response in the range of

velocities of interest, especially near the resonant frequency region which can be reduced with a conventional velocity-feedback controller, or alternatively feedback filtering based control (K. Tammi (a), 2003)

The use of a velocity feedback controller decreases the response of the rotor significantly. The active control brings the possibility to run the rotor across the critical speed. A feedforward system, based on an adaptive finite-impulse-response filter (K. Tammi (b), 2003), may also be designed to compensate disturbances caused by the mass imbalance if a reliable model of imbalance is available.

As shown in figure 6, every degree of freedom to be controlled requires a feedback control loop. The control system applies the force commands to attenuate shaft vibration while keeping the shaft into the radial position centre. The implementation of a shaft end vibration and position control scheme is shown in figure 6. It consists in an Agilent Technologies based hardware programmed under Matlab-Simulink V.9(a).

Every shaft end should be equipped with a control system comprising at least the parts shown in figure 5. It consists in two independent closed loop controllers to attenuate or suppress the shaft vibration in the normal plane of the shaft. The other shaft end should be equipped with a similar system.

### Control Loop hardware

Control loops accessories such as data acquisition and final control elements or actuation devices are implemented with specifically designed hardware based components.

Radial displacement is sensed by means of a data acquisition system which is based on a set of Eddy current probes. Axial displacement is measured under the same technology. Eddy Current Probe (ECP) systems are integral components,

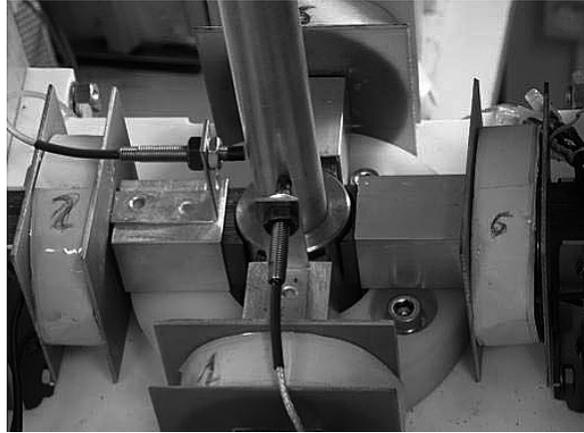


Fig. 5. A detail view of the test-rig electro-magnetic actuators for vibration control.

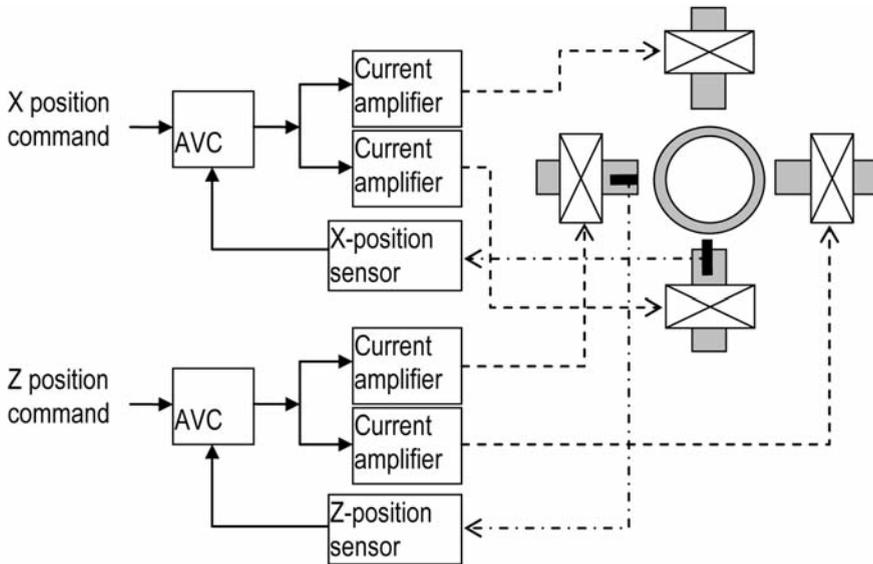


Fig. 6. Control structure for every shaft end

which typically consists of a non-contacting probe, an extension cable and a driver. An ECP typically senses mechanical movement and converts this movement (displacement) into a usable electrical signal.

**THE AVC UNDER UNBALANCE INFLUENCES**

To initiate the discussion, it is appropriate to consider the traditional diagram of a Jeffcott rotor as shown in figure 10 (a).

At very low speeds, unbalance forces are negligible. The shaft turns around bearing centreline and all rotating elements are concentric. This condition is depicted in the detail (b) of figure 10. As rotor speed increases, the straight shaft will deflect into the predictable mode shape shown in figure 7.

The only driving force in the system is the centrifugal force due to the unbalance mass  $M$ .

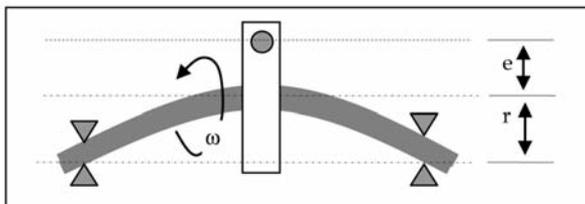


Fig.7. Shaft deflection mode under centrifugal forces due to an imbalanced mass.

The maximum bending deflection of the shaft is identified by  $r$  and the mass eccentricity by  $e$ . Furthermore, the rotational speed is indicated by  $\omega$ . By inspection of the figure 11 it can be seen that the shaft and disk are rotating at the

operating speed  $\omega$ . Simultaneously, the deflected shaft is whirling in the magnetic bearings at this speed. The mechanism driving this whirl is the centrifugal force generated by the eccentric mass on the disc. As rotor speed increases, the outward force increases in accordance with the normal centrifugal force  $F_c$  equation

$$F_c = M \cdot (r + e) \cdot \omega^2 \quad (1)$$

With regard to expression (1), the total radius of the mass unbalance  $M$  is composed of the shaft bending  $r$ , and the eccentricity of the mass with respect to the shaft centreline  $e$ . Such centrifugal forces will be compensated as much as possible by the active magnetic forces developed by the control algorithm. Since the shaft speed is squared in expression (1), the shaft rotational speed has a strong influence on the AVC algorithm. Nevertheless, such influence is attenuated due to the inertial effect of the rotor which causes the response magnitude to decrease as rotational speed increases. The developed test rig has been subjected to experimental validation where a feedback control action provided by a PID is implemented. Several controller gains have been applied so that the time response is achieved for variable rotational speeds.

As shown in figure 8 a vibration control test is performed under variable rotational speed. The rotating speed is varying from zero at the start point to 16 rad/sec. in about 50 seconds. At same time, different controller gains have been applied. As consequence, after three tests with different controller gains given as  $K_p= 3, 5$  and  $7$  respectively, three time responses were achieved and shown in figure 9 As depicted in figure 9, analytical or theoretical prediction of the optimum controller gain  $K_p$ , is

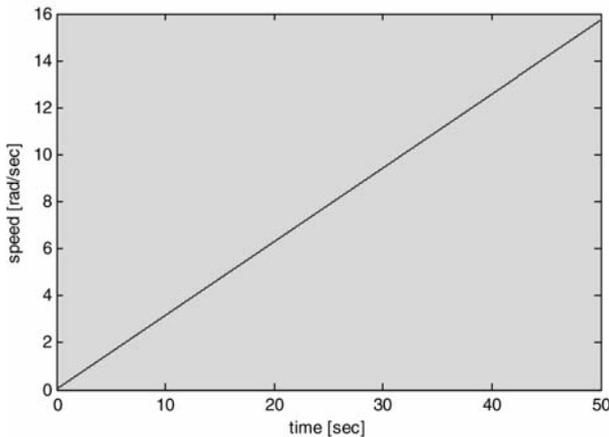


Fig. 8. Rotational speed (Rad/sec) as function of time.

not trivial. Instead, the selection of a controller gain  $K_p$  such that for a known rotating speed the response be acceptable, appears to be a satisfactory solution.

Generally, for very low rotating speeds, a low gain value is better than a high one. As rotational speed increases the effect of varying the controller gain is decreasing. This means that for high frequency vibration the vari-

ation of the AVC algorithm gain is not effective at all. An interesting topic to be taken into account with regard to the vibration attenuation is the vibration effect of the shaft on the shaft support rig. If the rotor mass insignificant with respect to the bedplate mass, then vibration attenuation may be considered effective. On the other hand, shaft vibration is transmitted to the bedplate, with dramatic consequences.

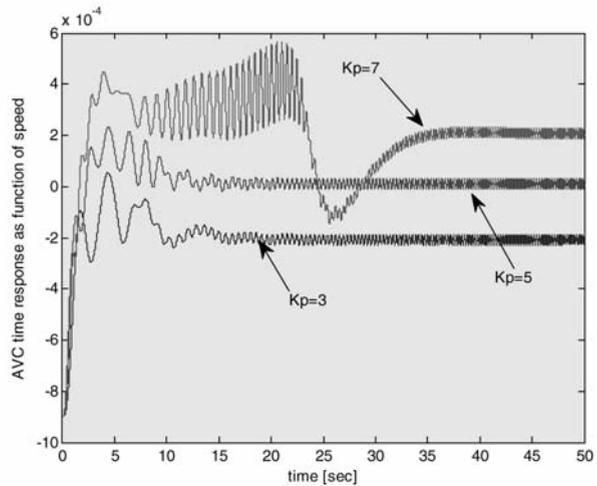


Fig. 9. Time response of the AVC as function of rotation speed.

## DISCUSSION AND CONCLUSIONS

A test rig has been designed and implemented to improve vibration attenuation of rotating machines installed on subsea environments.

Active balancing techniques promise solution to many of these problems and lead to significant economic benefits through increased reliability of machinery and the enabling of other advanced technologies.

Previous state-of-the-art non-adaptive active balancing control methods required extensive *a priori* modelling of system dynamics. Existing adaptive control methods for active balancing were not able to take advantage of the most recent data fast enough to ensure good performance and stability in the event of time-varying or nonlinear dynamics. This means that it is necessary a great research effort on this field, which must be associated to efficient and sophisticated test rigs to accurately improve and verify results.

According our experience, vibrations around the critical can be efficiently damped by velocity feedback control. It provided a possibility to run the rotor at critical speed by virtue of increased damping. It also provided smoother phase characteristics, which made feedforward compensation easier.

Control algorithms based on velocity feedback are one of the simplest examples for active vibration control in general. An important reason for this is the characteristically low damping of mechanical systems; a significant reduction in response can be achieved by a simple controller acting against vibration velocity. According to the literature review, the control method has also been applied to rotors. It has been shown experimentally that the reduction is significant in the resonance region for a rotor with low external damping.

The resonance can also be shifted with the control system by implementing a control force proportional to the displacement of the rotor. A load-carrying function is thus applied. This was briefly tested and found to work in the test environment. However, in heavy rotating machines, very large forces would be required and electromagnet based actuators under such conditions are not useful.

Velocity feedback control can also be successfully associated to feedforward control. Feedforward compensation converges at low frequencies, and outside the range of resonant frequencies, but diverges when the resonance frequencies are approached.

As mentioned, advanced control techniques and algorithms are being applied in order to render efficient productivity under the increasing industrial demands.

A variety of sophisticated control algorithms using the most efficient techniques to identify and estimate the rotating machine parameters, observer design, and advanced filtering is being applied. Nevertheless the AVC continues being an intensive research area. The improvement of control algorithms as well as control strategies will be validated by means of the developed test-rig, which can be modified to support a diversity of scenarios with different loads and changing severe operating conditions.

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