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LONG-TERM OPERATION OF AN AMB-SUPPORTED COOLANT PUMP IN A LIGNITE-FIRED POWER STATION - RESULTS AND OPERATIONAL EXPERIENCE -

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ABSTRACT
A conventionally suspended coolant pump was converted to a pump with complete active magnetic bearings and installed in a lignite power plant for demonstrating purposes. The main objectives were to test the possibilities of active magnetic bearings in turbo machines operating in a power plant and to proof their suitability. After commissioning the pump was used and tested in the power plant regime for this purpose. In addition, using the system-inherent signals of the magnetic bearings allows a machine and facility diagnosis and an online determination of the bearing forces.

INTRODUCTION
Bearing defects are, among sealing damages, the most common failure cause of pumps and compressors. Turbo machine malfunctions, caused by bearing defects during power plant operating, lead to reductions of plant reliability and safety of electrical power supply. Bearing defects can be excluded almost entirely when active magnetic bearings are used. In order to prove the applicability of magnetic bearings in turbo machines in power plant operation, a conventionally suspended cooling water pressure booster pump was completely retrofitted to active magnetic bearings and installed as the third coolant pump in a 500 MW block-unit in the Boxberg lignite power plant of the Vattenfall Europe Generation AG. The following objectives were set:

- Gaining operating experience under realistic conditions
- Determining of the bearings forces during pump operation
- Machine and facility diagnosis
- Model and software verification

This paper is a summary on the usage of the pump up to now, the operating experience gained with it and on the possibility facility diagnosis using inherent signals of the magnetic bearings shaft position and bearing currents.

COOLANT PUMP WITH MAGNETIC BEARINGS SM 400/400 A
Figure 1 shows the cooling water pressure booster pump SM 400/400 A being converted to complete active magnetic bearings.

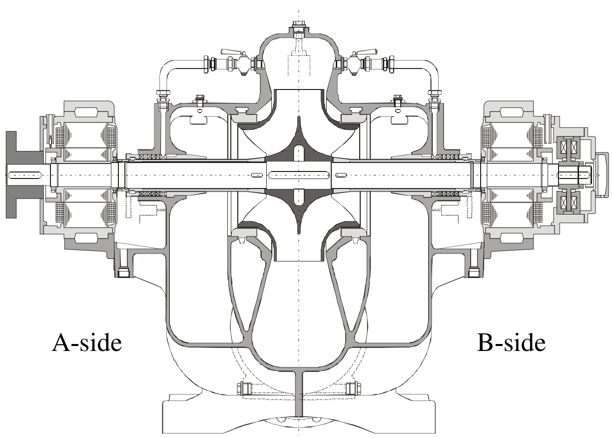

Figure 1: Cooling water pump SM 400/400 with active magnetic bearings

The pump is a single stage centrifugal pump with a single volute casing. The casing is parted horizontally in the shaft level. The bearing pedestals are outside the pump casing. Axial thrust balancing is carried out by a double section impeller. Packing glands are used for shaft sealing. The main parameters of the pump are given in Table 1.

TABLE 1: SM 400/400 A – Main parameters

Parameter	Value	Unit
Nominal speed	985/1475	rpm
Nominal capacity	915/1400	m³/h
Specific energy	137/319	J/kg
Delivery head	14/32.5	m
Shaft power	120/180	kW

Determination the utmost loads occurring at the impeller during the operation was the basis for the design of the magnetic bearings. The loads have been determined from geometrical data, pump parameters, product information and simulation calculations. The load forces consist of axial and radial acting static and dynamic forces. The dominating frequencies of loading forces were measured under various operating conditions at a pump identical in construction in the same power plant. Table 2 shows the parameters of the designed magnetic bearings.

TABLE 2: Parameters of axial and radial magnetic bearings

Parameter	Value axial	Value radial	Unit
Load capacity	12600	7500	N
Pole surface	137.6	48.6	cm²
Air gap	0.6	0.4	mm
Number of turns			
Bas coil	160	110	
Control coil	64	44	
Bias current	6	6	A
Control current	±15	±15	A

After the pump had been retrofitted to active magnetic bearings, it was commissioned in three stages:

- Dry run with a magnetic loading device
- Test on the flow-loop- test-bed for pumps at the KSB company – see Figure 2
- Installation and commissioning in the power plant

The trial operations on the test-beds were performed to proof that the magnetic bearings are functioning and to determine the attainable pump parameters.

FIGURE 2: Pump in the flow-loop-test-bed at KSB

The testing on the test-bed showed that the magnetic bearings have mastered the pumping operations at the nominal working point without problems. When the delivery flow was throttled, axial loads beyond the design force of the axial magnetic bearing occurred, resulting in a safety shut-down of the pump. After pump modification – an axial magnetic bearing with a higher bearing load capacity was assembled – the pump was installed as the third fully adequate operational pump in the block-unit of the power plant. To realize this, the pump was integrated in the facility considering processing and controlling technology aspects. Figure 3 shows the pump in the power plant.

FIGURE 3: AMB-pump operating in the power plant

Figure 4 shows a detail from the facility schematic diagram including the pump being the third pump in the cooling system. Each side of the pump contains a gate valve; a swing-type check valve is in addition at the pressure side. A regulating valve which sets the delivery flow depending of the temperature in the gas cooler of the generator has been inserted in the main discharge line DN 500.

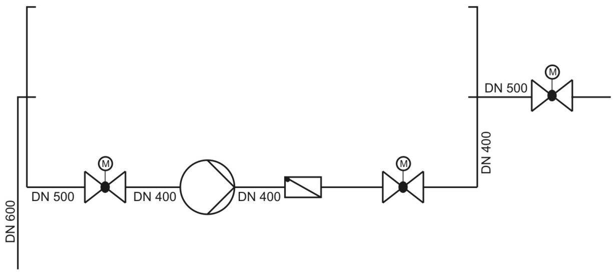

Figure 4: Facility schematic diagram

OPERATION IN THE POWER PLANT

Pump usage is subject to the operating regime of the power plant. There are three different operating modes:

- Standstill
  Driving engine and pump are switched off. Magnetic bearings are also switched off; the shaft is located in its touch down bearings.
- Reserve pump
  Magnetic bearings are switched on; the shaft is levitating in its magnetic bearings. The pump is filled and can be connected anytime in this operating mode.
- Operating pump
  The engine is on and is running at a preselected nominal speed. The pump is working in the pump operation.

Depending on operation mode and outside temperatures, the pump is operated at different working points. So, during the warm summer months the pump usually works for weeks without any change by the operating personnel at the design point, being the point with the highest level of efficiency and with the lowest bearing loads. During the cool winter months, the temperature in the cooling system is kept constant by reducing the delivery flow with the control valve, which is done by the operator at the master display. With further reduced cooling requirements, e.g. when the boilers operate in mono-unit operation, the pump can even be operated at the lower speed – cp. table 1.

Every alteration of the pump's working point affects the forces acting on the impeller and therefore also affects the bearing loads. The highest forces occur when the pump is starting up against the shut pressure-sided valve and when two pumps are operated parallel.
An algorithm especially developed for this purpose allows a determination of the bearings forces during operation online, using the system-inherent signals shaft positions and bearing currents [1].

Not only the signals of shaft positions and bearing currents, but also the pressures occurring on the suction side and on the pressure side in front of and behind the pump and the position of the regulating valve are recorded. The operating modes of the pump and the facility are diagnosed from these signals. Using the software-tool MLDia, a diagnosis system especially developed for this purpose, feature developing is generated from the logged signals. Via knowledge-based procedures, simple diagnosis predications are made from the signals. Fuzzy-based algorithms are used for generating complex diagnosis predications from the feature developing [2]. From the analyzing of typically failures at active magnetic suspended rotating machines follows the derivation of the occurrences, which to be diagnoses:

- Control loop of the magnetic bearings:
  Failures in sensors, signal conditioning, controllers, power amplifiers and bearing coils
- Pump/process:
  Static/dynamic unbalances, alignment failures, loads, loose parts of rotor, processes of touch and resonances
- Drive:
  Electromagnetic tensile forces, coil damages, damages at rotor or stator, unequal air gaps, rotational speed, acceleration, rotating direction
- Touch down bearings:
  Wear, damages, bearing plays
- Operating regime:
  Differences from normal operation, switch-on/off, operating mode, alterations in pressure and rotational speed

Figure 5 shows the monitoring system, which forms the basis for the diagnosis system. On the left-hand side is to see the orbits of the radial shaft positions (green) and corresponding bearing currents (red) during the pump operation. Beneath are the signals of the axial magnetic bearing. On the right-hand side, limit values for shaft position and bearing currents are supervised.

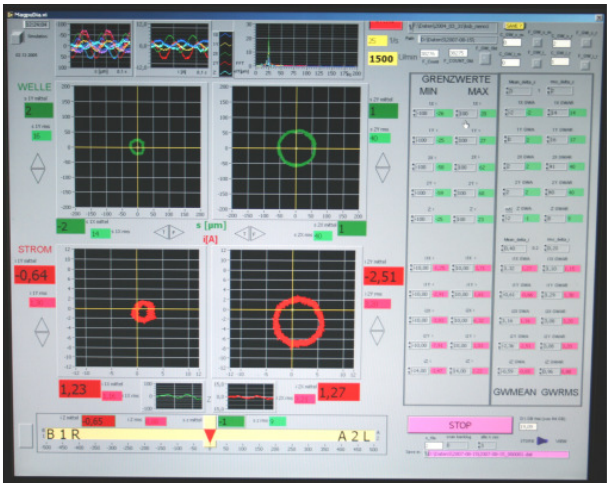

FIGURE 5: Monitoring system

Shaft position and bearing current signals are logged and stored. Occurring failures and operating modes are catalogued and form the basis for the diagnosis system.

SELECTED OPERATING MODES AND FAILURES

Operating modes and failures having occurred at the pump during commissioning and in steady operation in the power plant are described below.

Loose shaft parts

When the pump was being commissioned in the power plant, an assembly error caused that a shaft nut from the rotor sheet package on the A-side loosened, resulting in severe resonance vibrations of the pump when running. These vibrations could be compensated over a period of time and partly attenuated, but in the end led to a safety shut-off.

On the basis of shaft position signals, figure 6 shows the stochastic oscillations which occurred in the A-sided radial magnetic bearing in x- and y- direction (green and red lines) and the resonance vibrations interfere the shaft position signals on the A-side. These failures cannot causally be ascribed to alterations regarding the process parameters. Error detection in the functional chain sensors, signal processing, controlling electronics, power electronics in the end located the failure in the mechanical part of the magnetic bearings.

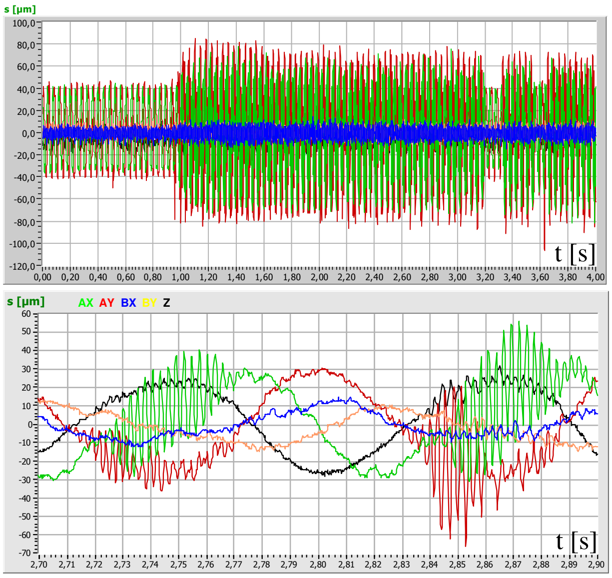

FIGURE 6: Stochastic shaft vibrations (upper graph) and resonance (lower graph) during pump operation

BX	radial B-sided in x-direction
BY	radial B-sided in y-direction
AX	radial A-sided in x-direction
AY	radial A-sided in y-direction
Z	axial in Z- direction

Those failures can be recognized from the high level of vibration in the amplitude spectrum of the shaft position and current signals, as shown in figure 7.

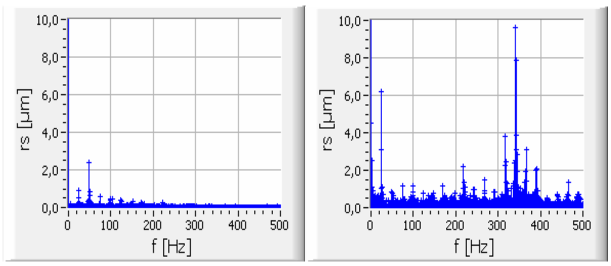

FIGURE 7: Amplitude spectrum of shaft position at the A-side
without (left graph) and with failure (right graph)

Misalignment of pump and engine

Figure 8 shows the A-sided radial shaft position for the start-up process when there is misalignment between pump and motor.

At the time of 1.25 s the engine is connected and the pump speeds up until it reaches its nominal speed. Synchronously to the speed the A-sided radial shaft position are going up in x-direction. At 4.6 s deviation is approx. 100 µm, reaching the maximum control current. After further 0.6 s follows a safety shut-off of magnetic bearings and drive system, the shaft gradually comes to a stop in its touch down bearings.

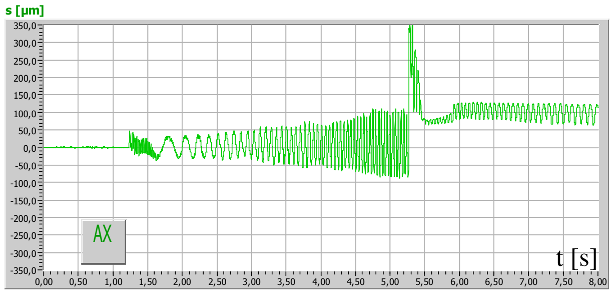

FIGURE 8: Radial shaft position A-side x-direction at start-up process when pump and engine are misaligned

Misalignments between drive system and working machine lead to higher bearing loads. With conventionally suspended engines, misalignment is only indicated when the bearing is already defect. With the magnetic suspended pump, misalignment has been detected through shaft position signals so that a defect escalation could be averted. Pump and engine were realigned, as shown in figure 9, and the pump was commissioned again.

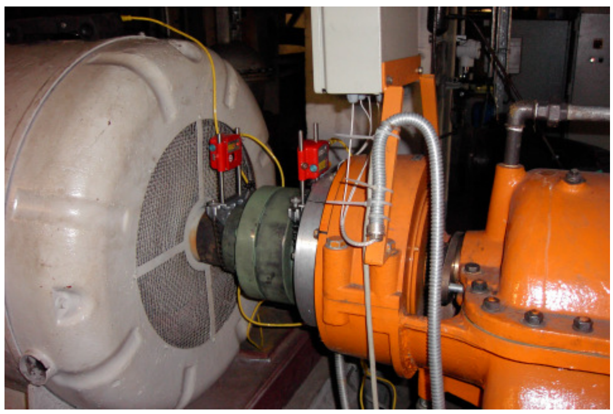

FIGURE 9: Realignment of pump and engine

Start-up in operating mode reserve pump

Figure 10 shows the commissioning of the pump from the operating mode reserve pump to the nominal working point on the basis of the data of bearing currents and pressure signals. The pump is connected at 11:22 a.m. with the regulating valve being opened to 100%. The pressure behind the pump rises from 1.6 bar to 4.22 bar, the pump pressure P decreases from 1.6 bar to 1.3 bar. After approx. 2 hours new thermal balance in the bearings has been reached, the bearing currents remain almost constant.

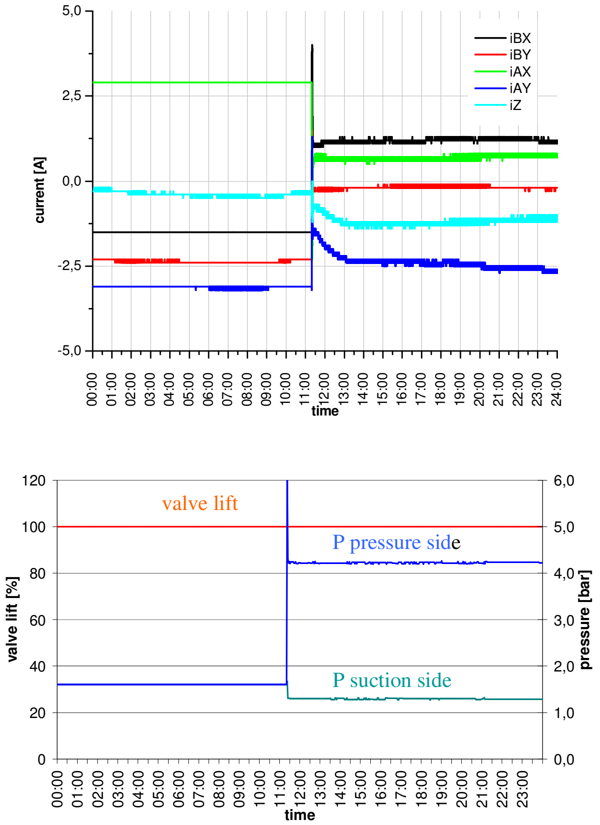

FIGURE 10: Bearing currents (upper graph) and pressures (lower graph) when pump is starting up to its nominal working point

Operating with strongly throttled delivery flow

The delivery flow had to be strongly throttled due to weather conditions in order to keep the temperature in the cooling system stable, when the pump operating at a high nominal speed. Therefore the regulating valve was gradually shut from 40.3% to 26.7%, as shown in figure 11. As a result, pressure behind the pump increased from 5.31 bar to 5.72 bar with a possible maximum pressure of 6 bar.

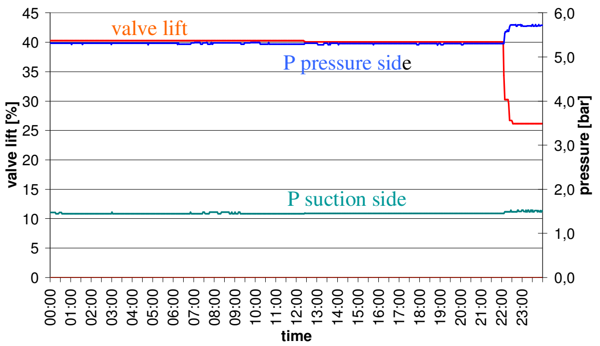

FIGURE 11: Regulating valve position and pressures with strongly choked delivery flow

Fig. 12 shows the effect of the working point alteration on the bearing currents and axial shaft displacement.

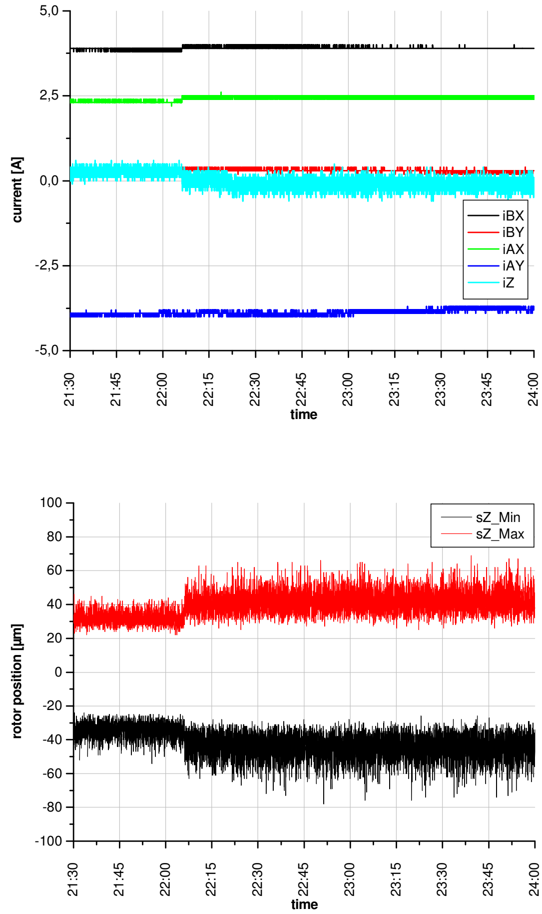

FIGURE 12: Bearing currents (upper graph) and axial shaft deviation (lower graph) with choked delivery flow

A working point close to maximum pressure does not mean an extraordinary load for the magnetic bearings. The bearing loads in the radial magnetic bearings slightly increases in x-direction. The pressure rise affects an increase of axial forces in A-side direction. The axial bearing current iZ changes from initially 0.3 A to -0.05 A on average. Pressure increase leads to a higher shaft deviation. Minimum values are changing from -30 µm to -41.5 µm, maximum values from 30 µm to 45 µm on average.

Mono-unit operation

Because of a low energy requirement the block operation system-inherently had to be transformed into mono-unit operation due to. As a consequence, the required cooling capacity decreases. So the pump was switched from high-speed operation to low-speed operation, cp. table 1. Figure 13 shows valve position and pressures before and after switching.

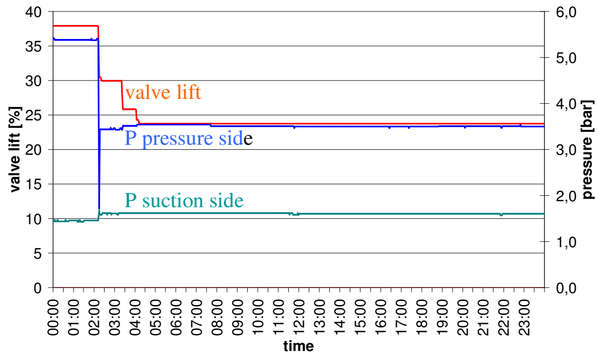

FIGURE 13: Pressures and regulating valve position with speed switching

Initially, the pump is working with a high speed at a regulating valve position of 37.9% and a pressure of 5.38 bar. About 2:15 a.m. the pump is switched to the lower rotational speed. Thus the pressure temporarily falls to the level of the pressure on the suction side. Simultaneously the regulating valve is phased down to 30.5%. Figure 14 displays the A-sided radial shaft position when switching. It is clear from the figure that the pump had to be shortly shut down for switching. At a time of 0.8 s the engine is shut off, at 4 s the non-return valve is shut. Then the shaft gradually comes to a stop in its magnetic bearings. At a time of 31.75 s the low-speed pump is connected. After that the valve phases down to 23.7%, pressure reaches 3.5 bar.

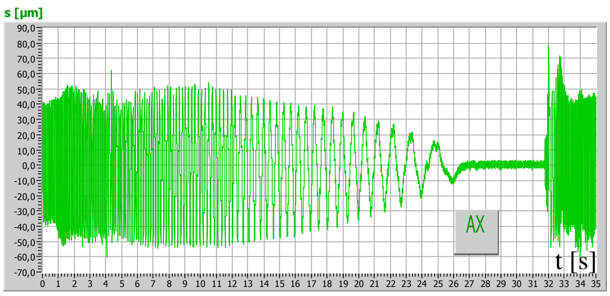

FIGURE 14: Radial shaft position A-side x-direction signal during changeover the rotational speed

Shut-down with flawed return valve

When switching off, the pump is separated from the pressure side of the cooling circuit by a swing-type check valve. For repair works on the pump, additional gate valves separating the aggregate safely from the cooling system have been inserted in the suction and pressure lines. Figure 15 shows the non-return valve and the slider on the pressure side of the pump.

The pump was switched off for a scheduled condenser check which included the emptying of the coolant system. Fig. 16 shows the cut-off on the basis of A-sided radial shaft position signal in x-direction.

FIGURE 15: Non-return valve and slider in the pressure line

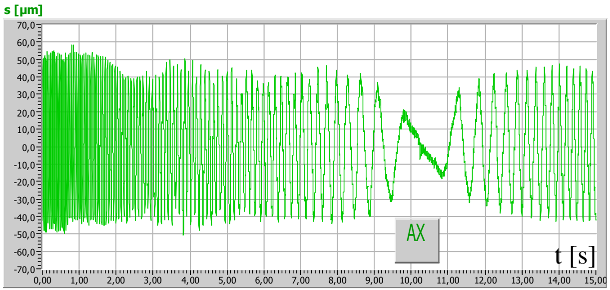

FIGURE 16: Radial shaft position A-side x-direction during cut-off with flawed non-return valve

At the time of 0.6 s the engine is shut down, the shaft run out in its magnetic bearings until it stops at 10.2 s. Due to a defect, the non-return valve does not shut. The pump is now functioning as a turbine, driven to a speed of 700 rpm by a reverse flow of water. Simultaneously with the engine cut-off, the gate valve on the pressure side shuts. This process lasts approx. 3 min., until the shaft comes to a stop, as shown in fig. 17

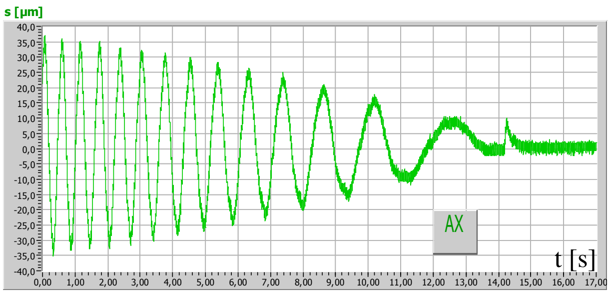

FIGURE 17: Radial shaft position A-side x-direction signal during shaft phasing down until it stops

Start-up against the shut pressure valve

Due to a defect non-return valve, the pressure-sided valve could not, as usual, be opened before the pump was connected. So the pump had to be put into operation with the pressure slider shut. The driving engine and the slider on the pressure side are switched on simultaneously. The slider is fully open after approx.

3 min. The start-up process means a heavy load on the bearings, as the pump has its maximum delivery height with zero delivery flow. In accordance to this, the energy supplied to the medium via the driving engine is transformed into thermal, but mainly into pressure energy.
Figure 18 shows the shaft positions and bearing currents in radial x-direction of the A-side and in axial direction during the start-up process. There are axial shaft deviations up to 100 µm during the start-up; axial current briefly reaches 14.8 A.

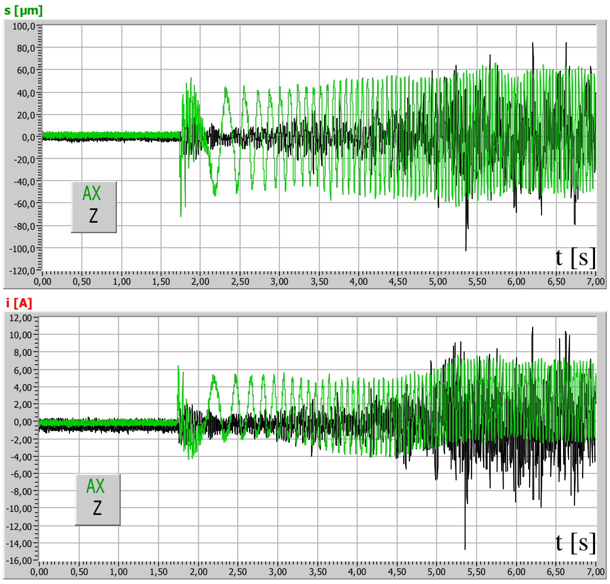

FIGURE 18: Shaft positions radial AX and axial Z (upper graph) and corresponding bearing currents (lower graph) during the start-up process against the shut pressure valve

Figure 19 shows the orbits of radial shaft positions and the axial shaft position signal approx. 10 s after connection and when the slider is fully open.

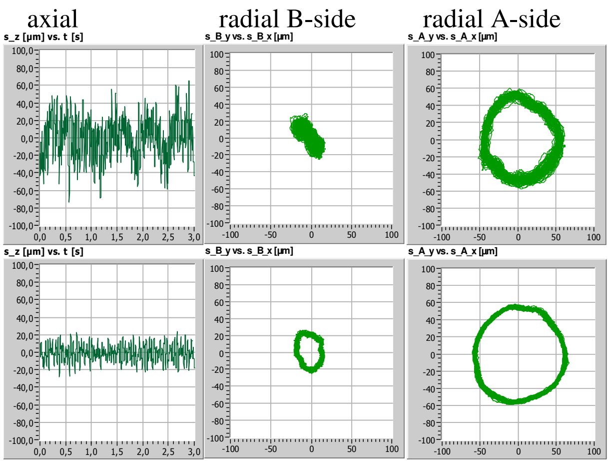

FIGURE 19: Shaft positions at slider opening and with fully opened slider

During the start-up process the shaft vibrates considerably more than when the pump is in steady operation. Furthermore, the B-sided orbit is clearly deformed. Axial deviation reaches values of over 70 µm, in steady operation it is approx. 20 µm.

Pumps parallel operation
A more continuous and an even heavier load compared to a start-up against the shut slider is the parallel operation of two pumps. They simultaneously pump into the cooling system with a pressure of over 6 bar. Fig. 20 shows the axial bearing loads before and when the second pump operates parallel. The average values of load FzA increase from 2800 to 9450 N, average values of load FzB from -3500 to -9400 N. The highest peak demands are 11900 N and -12050 N. They are controlled by the axial magnetic bearing [1].

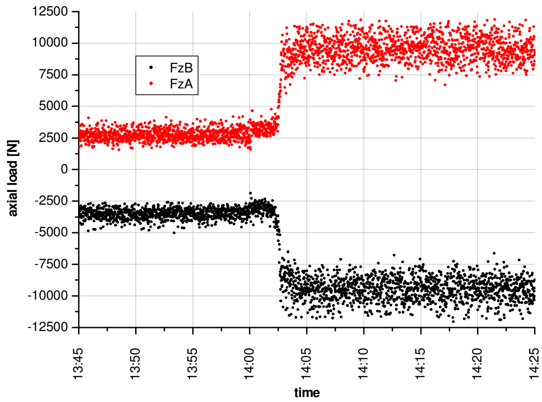

FIGURE 20: Axial bearing loads during parallel pump operation

OPERATION TIME OF PUMP
After commissioning in the power plant and successful start-up tests the pump began continuous working as a operating pump on 17.10.2005. Table 3 shows the operation and standstill periods from the commissioning on until June 2008 and reveals its availability.

Idle periods result from scheduled or non-scheduled unit standstill for maintenance or repairing works and have not been caused by the magnetic bearing. Magnetic bearings, sensors as well as power and controlling electronics have worked stable without disturbances since they have been commissioned.

TABLE 3: Operation of AMB supported coolant pump

Operating mode	**2005 since 17th Oct.**	**01/01/06 – 31/12/06**	**01/01/07 – 31/12/07**	**01/01/08 – 31/05/08**
Operation pump	45 d 8,75 h	319 d 18,75 h	324 d 5 h	147 d 3,25 h
Reserve pump	20d 6,25 h	25 d 17,5 h	17 d 3,75 h	3 d 15,5 h
Standstill	9 d 7 h	19 d 11,75 h	23 d 12,25 h	1 d 5,25 h
availability	86,8%	94,7%	93,5 %	99,2 %

CONCLUSION AND OUTLOOK

A pump retrofitted to complete active magnetic bearings has been used and tested as an operating pump in a lignite power plant for more than 2.5 years. All operating conditions occurring in the plant regime have been run and mastered without problems by the magnetic bearings. The magnetic bearings have never been the cause of a failure. The paper shows the possibilities of machine and facility diagnosis on the basis of selected operating conditions as well as an early failure recognition using the inherent signals of shaft position and bearing current in connection with the process signals pressure in front and behind the pump and the regulating valve position. Moreover, conclusions on bearing loads and thus on the state of pump and facility by online determining the bearing forces are possible anytime. The logged data form the basis for the diagnosis system and are used also for verifying the design models and simulation tools.

The experience gained during the development process, e.g. concerning design, construction and assembly of the magnetic bearings as well as from retrofitting, testing, installation and commissioning of the pump in the power plant are useful for developing further magnetic suspended plant components. Relating also to economical and ecological aspects, which were not crucial at the demonstration facility, aggregates which fully utilise the advantages of magnetic bearings are to choose. Replacing oil-lubricated sleeve bearings by magnetic bearings in aggregates such as turbines or boiler feed pumps is of great advantage concerning bearing losses, but also safety aspects, because then the whole oil system will not be necessary. Test operation results show that with this technology, the loads occurring in operation can be controlled and high availability can be achieved.

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2. Gärtner S., Neuartige Verfahren für die Überwachung und Diagnose von aktiv magnetgelagerten rotierenden Maschinen, Dissertation, Universität Kassel, Germany, October 5, 2007