DETERMINATION OF FORCES ON A COMPLETLY ACTIV MAGNETIC SUSPENDED COOLANT PUMP IN A POWER STATION*

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***Abstract* – This paper describes the retrofitting of a coolant pump from conventional bearings to active magnetic bearings and the determination of the bearing forces under real operation conditions in a power plant.**

***Index Terms* – active magnetic bearings, cooling water pump, power station, determination of bearing forces**

I. INTRODUCTION

At conventional pumps bearing damages are the most common causes for bearing failures and they lead often to the failure of the complete turbo machine. In power plants such failure kinds compromise the safety of the energy supply. By use of active magnetic bearings at units in power stations it is possible to eliminate largely the failure caused by bearing damages. The power plant relevant advantages of the magnetic bearings in opposite to conventional oil bearings like ball bearings or slide bearings are:

1) The contact less suspension induces no friction and wear. Thereby a practically unlimited lifetime and a high reliability are reachable.
2) Beside ecologically advantages the lubrication free operation of bearings leads to positive effects for the plant safety and to a reduction of the fire hazard.
3) The inherent signals rotor position and bearing current can be used for a monitoring without additional measurement. Therewith an on-line diagnosis is practicable for the detection of faults and to the prevention of damage escalations. Also statements to loads and operating point of the pump are possible.
4) By help of the controlled bearing force the improvement of rotor dynamic and automatic rotor balancing can be achieved.

For the proof of the applicability magnetic levitated pump rotors for power plant conditions in frame of a project a conventional supported pump was modified with active magnetic bearings. After successful tests the pump was installed in a 500 MW power station and tested in long-term operation for all operation cases occurred in a power plant. The aim was by help of this demonstration plant of a magnetic bearing pump to get operation experiences under real conditions and the experiences and results to use as basis for future additional applications in power stations.

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II. RETROFITTING OF A COOLANT PUMP TO ACTIVE MAGNETIC BEARING

A. Choice of the Pump

Fig. 1 shows the pump type SM 400/400 A selected for retrofitting. This type of coolant pump is used as pressure booster pump in a coolant system. Choice criterions were the existing of a redundant pump and the possibility to integrate the magnetic bearing in the area of conventional bearings. The pump SM400/400 is a horizontal, single stage, double suction centrifugal pump. The impeller has four blades. The double suction construction equalizes the loads in axial direction and minimises the axial bearing forces.
The main parameter are summarized in table 1. The operation takes place mainly with the higher nominal speed.

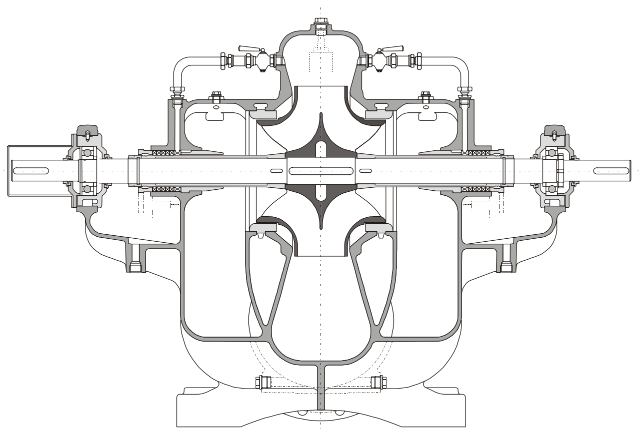

Fig. 1 Cooling Water Pump SM 400/400 with conventional Ball Bearings

TABLE I
SM 400/400 – MAIN PARAMETER

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

B. *Force Determination*

Basis for the design of the magnet bearings was the determination of the maximum loads appearing at the pump. These are determined from the geometric data, the pump parameters, manufacturer indications and by simulations. The load forces consist of static and dynamic forces in axial and radial direction.

The causes of static radial forces are the rotor weight and asymmetric pressure distribution between impeller and casing. Magnitude and direction change in dependence on operating point and flow-rate ï. Fig. 2 depicts the calculated static radial forces versus the operating point. Dynamic radial forces are caused by hydraulic and mechanical unbalances. The force frequencies are in range of the nominal frequency and blade frequency. The maximal radial force amounts 10240 N.

Axial loads result by pressure differences between suction and pressure side of the pump. Due to the double suction principle axial forces should be minimal. Rest axial thrusts accrue by manufacturing deviations, inhomogeneous seal gaps or asymmetric flow conditions. On the basis of the manufacturer information the axial loads amounts about 25 % of the maximal static radial force (1750 N).

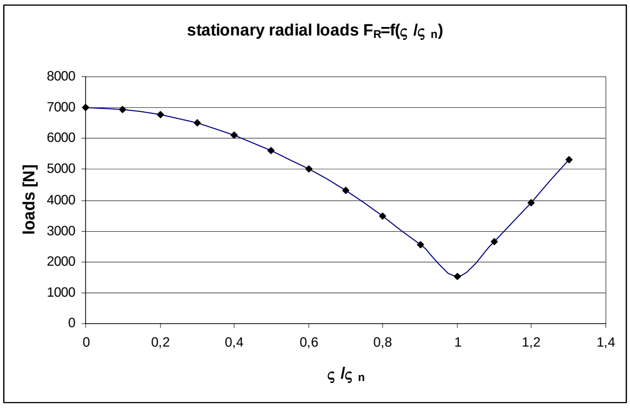

Fig. 2 Stationary radial Loads vs. Ratio between Flow-Rate and nominal Flow Rate $q = ï°ï_n$

The dominate load frequency was measured at a operation pump in a power plant and lie in the range of the nominal rotation frequency of 25 Hz - 200 Hz [1].

C. *Design, Construction and Test of the Magnetic Bearings*

On the basis of the estimated loads the magnetic bearings were designed. Under consideration of the rotor part of the magnetic bearings the maximum static force was determined to 10840 N. The proof of the reliability performance is based on the criterion of dynamic stiffness (DDS) by help of simulation [1]. Table II shows the specifications of the magnetic bearings.

TABLE II
PARAMETER OF AXIAL AND RADIAL MAGNETIC BEARINGS

Parameter	Value axial	Value radial	Unit
load capacity	2100	7500	N
pole surface	41.2	48.6	cm²
air gap	0.8	0.4	mm
number of turns bias coil control coil	 90 36	 110 44	
bias current control current	6 ±15	6 ±15	A A

Fig. 3 shows the pump equipped with magnetic bearings.

The start-up of the pump was performed in three phases:

- dry test with a magnetic load device in the lab
- test by using the flow loop test bed at KSB in Frankenthal (Germany)
- Installation and start-up in the power station

Fig. 4 shows the pump installed in flow loop test bed at KSB.

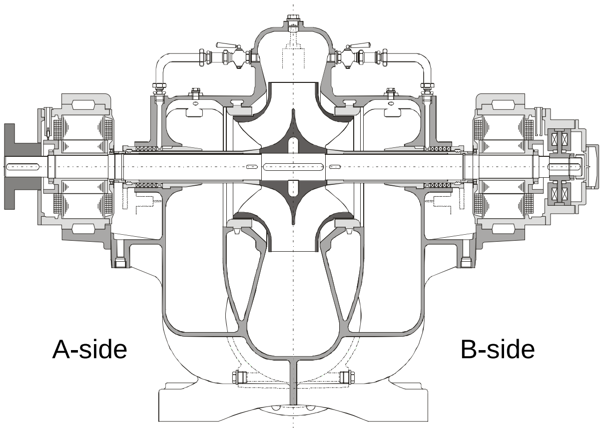

Fig. 3 Cooling Water Pump SM 400/400 with Active Magnetic Bearings

Fig. 4 Pump SM 400/400 in the Flow Loop Test Setup at KSB

The flow loop tests have shown that the magnetic bearings work without problems in range of nominal speed and nominal flow. At deviations from the nominal values due to throttling of the flow rate axial loads occur higher than the designed forces of the thrust bearing. The result was the automatic shut-off. To reach a higher load capacity in a first step the air gap was decreased by help of a stronger thrust disc. Thereby the load capacity could be increased to approx. 3600 N.

D. Installation and Pump Start-Up in a Power Plant

The pump was installed as a third full 100 % pump in the power station. Fig. 5 shows the pump installed in the power station.

Fig. 5 Pump SM 400/400 installed in the Power Station

The start-up tests contained the test of the electric and control imbedding of the pump into the operation regime of the plant and the required locking. After that all operation cases were tested like starting against swing-type check valve, against closed gate valve, pumps parallel operation as well as continuous operation as operation and redundant pump.

The tests have shown, that the load capacity is exceed during operation caused by throttling of the flow rate and due to the automatic pump shut-down now safety operation is possible. The radial magnetic bearings control the occurred loads also during displacements of the nominal air gap without any problems. A new axial magnet bearing with greater load capacity was necessary for the protection of long-term operation of the magnetic bearing pump.

E. Thrust Bearing with higher Load Capacity

Based on the design principle of the existing thrust bearing and on the given geometric environment a new axial magnetic bearing was designed. The load capacity is approx. 12.5 kN. Fig. 6 shows the old and the new thrust bearing at the pump.

Fig. 6 Previous and new Axial Magnetic Bearing

Table III lists the parameter of the new thrust bearing.

TABLE III
PARAMETER OF THE NEW AXIAL MAGNETIC BEARINGS

Parameter	Value axial	Unit
load capacity	12660	N
pole surface	137.6	cm²
air gap	0.6	mm
number of turns bias coil control coil	 160 64	
bias current control current	6 ±15	A A

III. Bearing Load Calculation

A. Motivation

For conventional supported pumps the determination of bearing loads occurring during the operation is not possible or only with high efforts. The retrofitted pump SM400/400 uses for this tasks the system inherent signals rotor position and bearing currents. The determination of the forces is realised by the bearing specific force-current-displacement-field.

The knowledge of the quantity of the load forces and the period of effectiveness for different operating points of the pump allows the user to carry out estimates for the life time of the conventional bearings of structurally identical pumps and to include these in the maintenance planning. Additional it is a diagnosis of process states possible differing from the normal operation.

For pump developer and manufacturer the knowledge of the loads is useful for the verification of analytic calculations and allows the optimization of materials and construction.

B. Force-Current-Displacement-Characteristic of a Bearing Magnet

The determination of bearing loads is based on the force-current-displacement-characteristic of the bearing magnets. This characteristic considers the non-linear relationship between bearing force, air gap and current as well as the non-linear influences of the core material saturation and hysteresis. For the calculation of the characteristic field a Mathcad®-Program has been developed. With that a calculation of the characteristic diagrams of bearing magnets is possible in consideration of arbitrary geometry and of all nonlinearities on the basis of the method introduced into [2]. Input parameters are the bearing geometry (rotor part and stator part) include the air gaps (magnetic and auxiliary bearing), the turns of the bias current and control coils, the bias and control current and the magnetization curves of materials.

Fig. 7 shows the calculated characteristic field of a 12.5-kN axial bearing magnet. The superposition of the magnetically effect of the bias current coils and the control coils causes a magnetic force from 0 N (at -15 A control current) and maximal forces depending on displacement at 15 A control current. At nominal air gap 0.6 mm (0 µm displacement) and 15 A control current the load capacity amounts 13 kN. The maximal values at the margin positions are 8.1 kN and 17.3 kN resp.

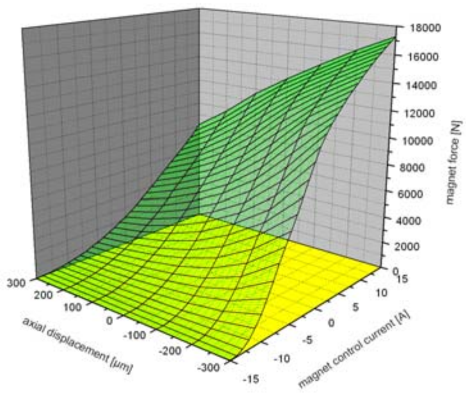

Fig. 7 Force-Current-Position-Characteristic 12.5 kN Axial Bearing Magnet

C. Characteristic Field for a Axial Bearing

The axial magnet bearing consists of two ring magnets arranged diametrically and between a thrust disc placed on the rotor – compare Fig. 3. The difference control of coils ensures, that the magnetic effect of a control winding superposes the bias field (amplifying or relieving). Fig. 8 depicts the characteristic field for both bearing magnets.

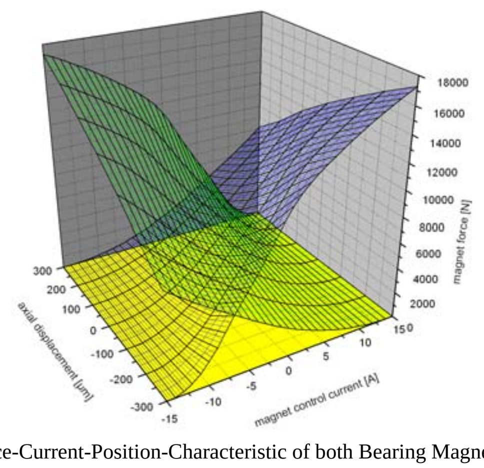

Fig. 8 Force-Current-Position-Characteristic of both Bearing Magnets

The total bearing force is the vector summation of the single forces acting on the thrust disc in dependence on the displacement and consequently the bearing current. Fig. 9 shows the characteristic field calculated on the basis of the characteristic fields for the single forces. The positive or negative sign of the bearing force indicates the force direction. For a better illustration the level is pictured where the value of the bearing forces is zero. A linearization at the operating point in wide ranges is achieved, because of in difference acting forces. The nonlinearities caused by the saturation are remaining.

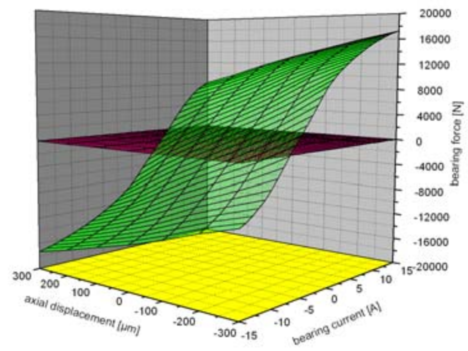

Fig. 9 Force-Current-Position-Characteristic 12.5 kN Axial Bearing

IV. Approximation by a Polynomial Approach

For online calculations a analytical function has been developed.

For the approximation of the characteristic field the following approach was chosen (1).

$$F_\nu(s,i) = \sum_{k=0}^{m} \sum_{l=0}^{n} s_\nu^k \cdot i_\nu^l \cdot a_{(n+1)*k+l} \tag{1}$$

Inputs data are the bearing forces F_ν, the bearing air gap s and the bearing current i at mesh points ν (Fig. 9). The parameter m and n are the polynomial degrees in the air gap and current direction resp. The vector a is determined by help of the values $F(i,s)$, i and s for all mesh points ν. Because polynomial with higher degree have a disposition to oscillate and also increase the computation effort, a compromises was searched to minimize the deviations in acceptable limits. For a polynomial with degrees m=4 and n=5 (2) the maximal absolute fault is ±229 N. According to (1) 30 coefficients a must be calculated.

$$F(i,s) = \left[s^0 \cdot i^0 \ldots s^4 \cdot i^5\right] \cdot \begin{bmatrix} a^0 \\ \vdots \\ a^{29} \end{bmatrix} \tag{2}$$

In a wide range of the characteristic field the relative fault is less than ±2 %. Fig. 10 shows the initial field and the approximated field.

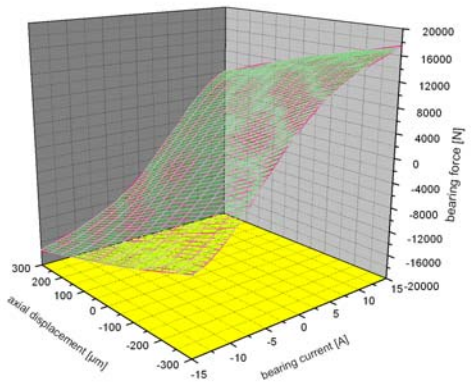

Fig. 10 Initial Field (pink) and its Approximation (green)

V. Determination of Bearing Forces for selected Operating Cases

A. *Speed Increase to Nominal Speed*

Fig. 11 shows the calculated forces for the two ring magnets FzA and FzB during the increase of the rotational speed to nominal speed.

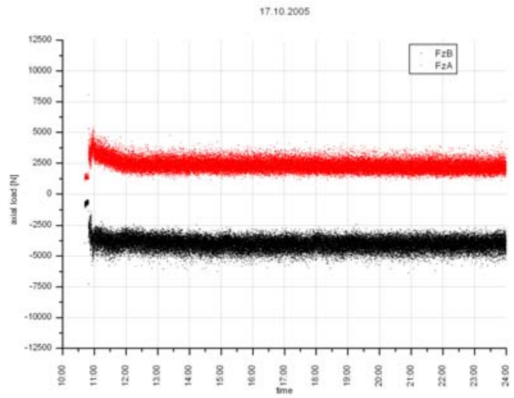

Fig. 11 Forces during starting Process of the Pump

After switch on it comes to a settling process, because of the heating of the magnetic bearing components. It is finished approx. 1.00 p.m. (13:00 h). After that the mean loads amounts approx. 2300 N and -4050 N resp. During the start phase peaks occur up to 8200 N and -7300 N resp.

B. *Pump Operation at Nominal Operating Point*

Fig. 12 shows the calculated axial forces for the ring magnets FzA and FzB during pump operation at the operating point. The mean forces amounts at the A-side 1850 N and at the B-Side -3300 N. The pressure values are 1.33 bar at the suction (Ps) and 4.25 bar at the pressure side (Pd).

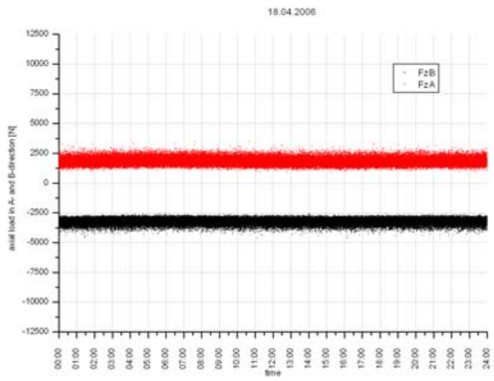

Fig. 12 Bearing Loads at Pump Operation at Operating Point

C. *Pump Operation by Changes of the Operation Point*

Fig. 13 shows the influences of operating point changes to the bearing forces. Table IV summarizes the appendant environment conditions like valve lift, pressures (Ps, Pd) as well as the mean forces (FzA, FzB) occurred. The maximal differences of the forces are 1250 N and 700N resp.

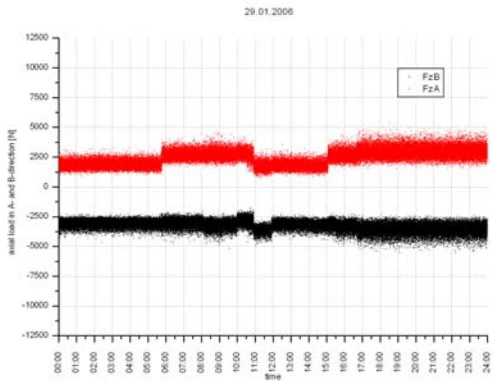

Fig. 13 Axial Forces in Dependence on the Operating Point

TABLE IV
Influence of Alterations of Operating point at 29.01.2006

time	valve lift [%]	Ps [bar]	Pd [bar]	FzA [N]	FzA max [N]	FzB [N]	FzB max [N]
00:00	61	1.44	4.9	1900	4000	-3100	-4650
05:48	46	1.49	5.27	2700	4450	-3050	-4700
08:10	40	1.52	5.38	2750	4700	-3250	-5200
10:00	48	1.48	5.16	2700	4200	-2900	-4600
10:58	53	1.48	5.02	1700	3200	-3600	-4900
11:58	60	1.45	4.9	1800	3550	-3150	-4450
15:06	45	1.43	5.19	2650	4450	-3300	-4900
16:48	35	1.47	5.44	2950	5400	-3300	-5500

D. *Switching between lower and higher Nominal Speed*

Under cold weather conditions the pump is operated with the lower nominal speed. It is possible to switch direct from lower to higher speed. Fig. 14 shows the loads occurred during this process. The valve lift of the throttle valve is 44 %.

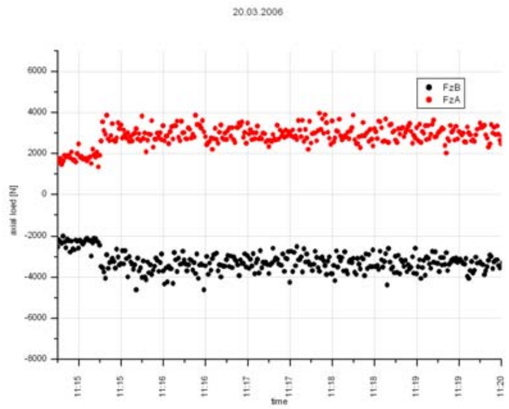

Fig. 14 Load Change by Speed Switching

In Table V contains the mean and maximal forces measured before and after switching.

TABLE V
LOADS AT SWITCHING FROM LOW TO HIGH SPEED

Speed [rpm]	Ps [bar]	Pd [bar]	FzA [N]	FzB [N]	FzAmax [N]	FzBmax [N]
985	1.61	3.4	1800	-2350	2450	-3000
1475	1.44	5.19	2950	-3350	3950	-4600

E. Parallel Operation

By parallel operation of a second Pump together with the magnetic bearing pump the maximal pressure (6 bar) was achieved. Fig. 15 shows the bearing forces before and after the switch on of the second pump. As result the mean values increase FzA from 2800 N to 9450 N and FzB from -3500 N to -9400 N. The Peaks appeared are 11900 N and -12050 N resp. and are controlled by the axial bearing.

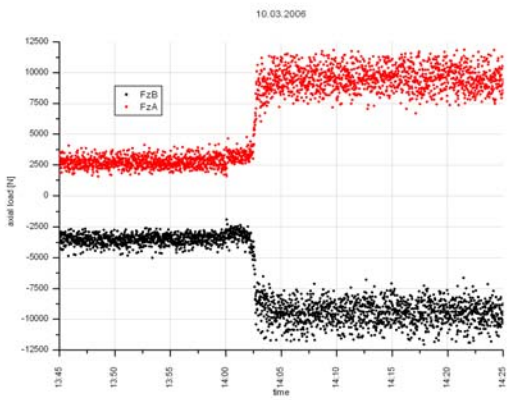

Fig. 15 Bearing Forces during parallel Operation

F. Scheduled Pump Switch-off

Fig. 16 pictured the axial bearing forces by a scheduled pump switch-off. During the process peaks up to 10 kN occur in both directions. The bearing forces (approx. ±1000 N) are to lead back to vibrations caused by the system pipelines.

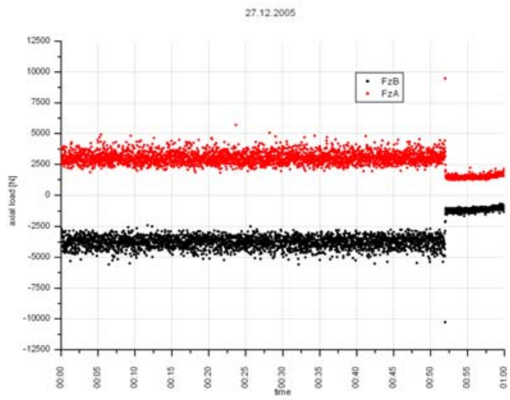

Fig. 16 Axial Bearing Forces by a scheduled Pump Switch-off.

VI. Conclusion

A conventional supported pump was modified with active magnetic bearings and installed in a power plant for demonstration reasons. On the basis of non-linear characteristic fields analytical functions have been developed for the calculation of the bearing forces. By help of the system inherent signals air gap and bearing current the forces occurred during the operation can be online calculated. Operation tests have shown that the axial loads are higher than estimated.

References

[1] F. Worlitz, St. Gärtner and T. Rottenbach, “Completely active magnetic suspended coolant pump for application in power stations” *8th International Symposium on Magnetic Suspension Technology* pp. 157-161, September 2005.

[2] H. Stegemann, "Neuartige Verfahren für die Modellierung und digitale Regelung aktiv magnetgelagerter Systeme", Shaker Verlag, 2003, pp. 46-49.