Design of a Hermetically Sealed Chemical Pump With Magnetic Bearings

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ABSTRACT

Increasing demands for environmental protection and severe anti–pollution regulations have led to an extended market for hermetically sealed pumps, e.g. so called canned–motor pumps.
The usually installed medium–lubricated hydrodynamic bearings reach their limits of operation if the medium has very poor lubrication characteristics and/or contains gas. Also temperature shocks or bearing overloads may damage such conventional bearing arrangements. Active magnetic bearings seem to be an interesting alternative in such a case. The focus of this study is the possibility to integrate magnetic bearings to one–stage chemical pumps which are available typically in the range of 2 kW to 15 kW. Not only the design of one suspension system is the objective, but the comparison of conventional radial/axial magnetic bearing designs and conical bearing shapes. The influence of the stainless–steel can inside the stator and around the rotor is calculated numerically for static conditions. A test rig is introduced which allows the evaluation of the static calculations and the quantification of power losses caused by Eddy current effects in the can.

INTRODUCTION

The strict application of anti–pollution laws has enforced the development and optimization of hermetically sealed chemical pumps. Fig.1 shows the cross–section of a standard one–stage chemical pump with canned motor drive.

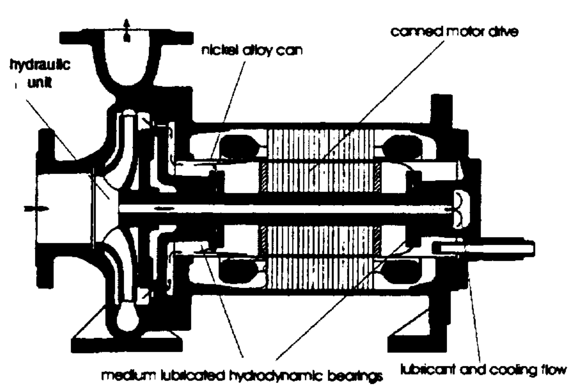


Fig. 1: Cross section of a standard canned motor pump [Neum–94]

To separate the fluid–filled regions from the dry ones of the pump, this design uses a stiff, thin (approx. 0.3 mm), coaxial housing of antimagnetic and highly corrosion–resistive material (stainless steel or nickel alloy), the can, as shown in fig.2. A photograph of a rotor can is shown in fig.3.

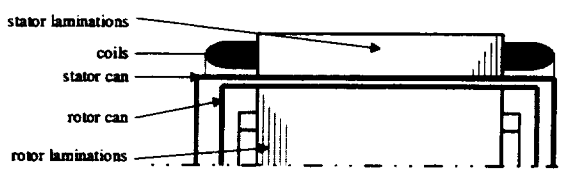


Fig. 2: Principle setup of a canned Motor

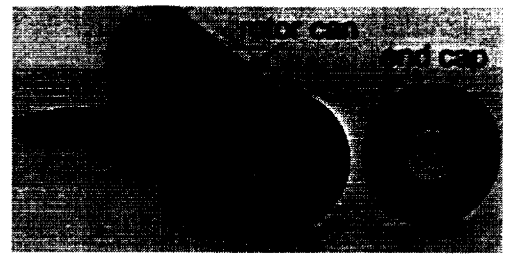


Fig. 3: Cylindrical can with two caps

The usually applied hydrodynamic bearings are located inside this can, in the fluid–filled region. Therefore they are designed as medium–lubricated bearings, often realised with very hard, highly abrasion–resistive materials, e.g. silicon–carbide (SiC). Even such advanced bearing designs reach their limit if the fluid gets very poor lubrication characteristics, extremely low viscosity or the risk of cavitation. In addition, frequent start–ups and run–downs, sudden variations in temperature and transport of abrasive solids limit the lifetime of conventional bearings. Bearing failures have been reported as one of the common problems in hermetically sealed pumps [Voll–93]. One reason to start the present research activities has been the lately published articles about the future perspectives of pump development [Neum–94, Hergt–99]. The application of active magnetic bearings as an alternative has been outlined especially for chemical pumps. Publications about successful designs for such machines are already available. In 1989, Allaire et al. [Allaire–89] reported about the design and realization of a 15 kW canned motor pump equipped with magnetic bearings. He used two 8–pole radial magnetic bearings and two single–acting thrust bearings to support the pump rotor. The two radial bearings have different geometries corresponding to the required bearing loads. Pump performance and vibration levels before and after the installation of the magnetic bearing system have been measured and compared at various locations on the pump casing. In the outlook, the installation of a similar pump in a chemical plant to determine the pump performance for several years was mentioned. A similar project with a 37 kW pump was reported by Marscher et al. [Marscher–91] in 1991. In this pump, a hydraulic balancing device has been applied to reduce the remaining hydraulic axial thrust and to renounce an active axial bearing. One year later, in 1992, Hanson et al. [Hanson–92] described the development of magnetic bearings fitted to a canned motor pump with a motor power of about 93 kW. Two identical radial and one double–acting thrust bearing have been utilized to support the pump rotor. Beside the pump characteristics and rotordynamics, the cooling requirements have been specified and the benefits of magnetic bearings for diagnosis purposes have been pointed out. Gempp et al. [Gempp-96] published the introduction of a bearingless 1.5 kW motor and 3-phase AC bearings for a small canned chemical pump. Due to the bearingless motor, only one additional radial bearing is installed beside a double–acting thrust bearing.

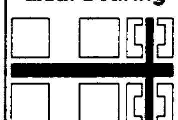
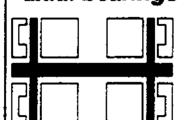
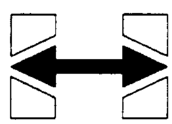
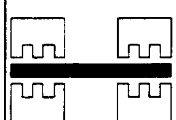
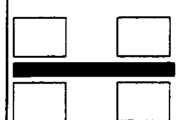
To leverage active magnetic bearings to become a competitive element in commercial canned motor pumps, the additional investments and especially the life–cycle costs have to be reduced. The present study focuses on the comparison of different geometrical bearing designs. Conical and teeth pole contour bearings are investigated by means of their practicability, especially by means of installation efforts, and the possible reduction of electronic hardware. The investigation of the influence of the can on the magnetic circuit will enable the necessary enlargement of the bearing load capability. For this purpose, a test rig to realize rundown tests with different bearing configurations is designed.

MAGNETIC BEARING DESIGN

A selection of the different targets which have been defined at the beginning of the study is listed below:

- The compatibility with existing commercial pumps will ensure the realization of a laboratory test installation with justifiable effort.
- Diameters of rotor and stator parts have to be similar to the motor diameter to use continuous cans and to avoid the necessity of complicated welded joints on the thin material.
- The pump design should be modular, which means that an easy exchange of different bearing geometries without modifications on hydraulic and drive units should be ensured.
- To simplify the assembly of the pump, it is desirable to balance the pump rotor with mounted laminations and can, and to insert it without any dismantling into the pump housing.
- The shaft length extention should be kept as small as possible to preserve the conditions for a dynamically stiff rotor.

To respect all these requirements, a set of principle arrangements of magnetic bearings mounted on a rotor with an overhung load are compared in table 1. The comparison has been carried out for a pump bearing with a load ratio (f_{rad}/f_{ax}) of about 1.25.

Version	***A***	***B***	***C***	***D***	***E***
	2 Radial bearings, 1 double-acting axial bearing 	***2 Radial bearings, 2 single-acting axial bearings*** 	***2 conical shaped bearings*** 	***2 teeth pole contour bearings*** 	***conventional hydrodynamic bearings*** 
adequate radial load capacity	available, depending on diameter, length	available, depending on diameter, length	available, depending on diameter, length, conical angle,	available, depending on diameter, length	not available with critical fluid parameters
adequate axial load capacity	available, depending on diameter	available, depending on diameter	available, reasonable only for high force ratios (radial/axial)	only available for very high force ratios (radial/axial)	not available with critical fluid parameters
bearing diameter = motor diameter	possible	possible	diameter not constant	possible	not necessary
conditions for dynamically stiff rotor design	fulfilled	fulfilled	fulfilled	fulfilled	fulfilled
assembly without rotor dismantling	possible without additional restrictions	possible, if outer diameter of axial and inner diameter of radial bearing are identical	possible	possible	possible
shaft length extension	1,15	1,2	1,1	?	1,0 (original length)

Version	A	B	C	D	E
shaft weight extension	1,65	1,55	1,4	?	1,0 (original weight)
installation of can	stator and rotor can have to be assembled and welded out of different parts	can may have a constant diameter, if outer diameter of axial bearing and inner diameter of radial bearing are identical	stator and rotor can have to be manufactured and installed with conical shape	possible, in principle, influence of axial force development not yet clarified	standard procedure

Table 1: Comparison of different bearing configurations

The estimated performance of each design is evaluated. Two designs, version B and C are selected as the most promising ways to realize a future prototype pump.
The main reason for version B is the possibility to install the rotor after the balance procedure without the need of dismantling. A simple can geometry for stator and rotor is sufficient for this configuration. In principle, the rotor can only has to be extended if both radial bearings have identical diameters. This is also valid for the stator can, if easy disassembly of the pump is not required. If the disassembly is necessary, e.g. for a scheduled test pump, the continuous can can be replaced with three discrete cans. A diameter similar to the motor can diameter and equal length are required. Radial and axial bearing geometry can be designed with validated calculation methods. The decentralized control is not expected to be complicated because interactions between the different degrees of freedom are of minor order. The shaft length extension factor is slightly higher than for version A with a double–acting axial bearing, but this disadvantage is compensated by the easier installation procedure.
On one hand, version C requires additional effort to install the can around a conical shaped lamination. But, on the other hand, it enables the reduction of hardware components because the axial and the radial force development is coupled via the conical angle. Therefore, no additional thrust bearing is required. The coupling may complicate the design of a decentralized control for the system. The shaft length extension is only about 10 % for the given load ratio and conical angle. The calculation of the optimum conical angle has been described by Lähteenmäki [Lähteen–98]. The shaft length extension strongly depends on the load ratio, because this relation leads to the selected conical angle.
Version D, namely teeth contour pole bearings, seem to be an interesting idea. Their design has been introduced by Canders et al. [Canders–97]. Unfortunately, so far, no publications about measurements to verify the calculations have been found. Nevertheless, the teeth pole stator contour bearings will only be an alternative if axial loads are negligible to the radial loads.

MAGNETIC FORCE CALCULATION

Static forces

In the design phase of the electromagnets, the static magnetic forces have first been calculated analytically, as it is described in [SchTr–93]. The static force losses caused by the stainless steel cans have to be estimated. Therefore, the airgap used in the calculation is increased by the thickness of 2 cans (rotor and stator can). The nominal airgap, required by the minimum circulation flow to ensure an adequate cooling flow for the motor, is about 0.4 mm. The resulting airgap with two cans is 1.0 mm. The results of this analytical calculation are compared with results derived from a numerical calculation with the FE–

software FLUX2D [FLUX2D–98]. The numerical calculations have been performed under consideration of the material properties of the can material, table 2.

Material	*ISO NiMo16Cr16Ti (UNS N06455)*
Density	8600 kg/m³
Relative Permeability at 0.3 T (approx.)	40
Specific Resistance at 373 K	1,3 10^{-6} Ωm

Table 2: Material properties of nickel alloy can

To receive comparable results in the analytic and the numeric calculation, material properties and magnet geometries have been kept identical. The coil current density and the winding density in the coils are similar to the ones of the asynchronous motor drive. Calculation results are shown in comparison in table 3. Fig. 4 and fig. 5 illustrate the results of the numerical calculation. The numerical results without consideration of the can material properties are performed under the assumption of an airgap of $1*10^{-3}$ m. With the can integrated, the nominal airgap decreases to $0.4*10^{-3}$ m, the remaining $0.6*10^{-3}$ m are filled with the can material. The slightly higher relative permeability of the nickel alloy can explain the higher calculated magnetic force in comparison with a similar gap filled with air.

	Analytical calculation	*Numerical Calculation*	*Numerical Calculation under consideration of can material properties*
airgap	$1*10^{-3}$ m	$1*10^{-3}$ m	$0.4*10^{-3}$ m
can thickness	–	–	2 x $0.3*10^{-3}$ m
stray fluxes	not considered	calculated	calculated
inhomogenity of flux distribution	not considered	calculated	calculated
nonlinear material properties	B/H curve	B/H curve	B/H curve
estimated max. force	100%	84%	90%

Table 3: Comparison of analytic and numeric calculation

Beside the force characteristics, the temperature distribution inside the pump will be determined. The calculations will be based on a finite–element model of the pump under consideration of heat transfer between fluid, asynchronous motor drive and bearings. If the cooling based on convection is not sufficient, an additional external cooling loop will be added to the pump to enable the operation of the bearings with satisfiable high coil current densities.

Dynamic Effects, Eddy Current Losses

The electric properties of the can material will influence the required power to support the pump rotor. Static force reduction in comparison to a magnetic bearing with identical geometry but without a can has been elucidated in the previous paragraph. As a second important factor, the developed Eddy currents in the can should be contemplated. For canned asynchronous motor drives the losses caused by the can are reported as about 10–15 % of nominal power [Neum–94, Vetter–98]. To quantify the Eddy current losses experimentally it has been decided to design a handy test rig to perform measurements with different magnetic bearing configurations.

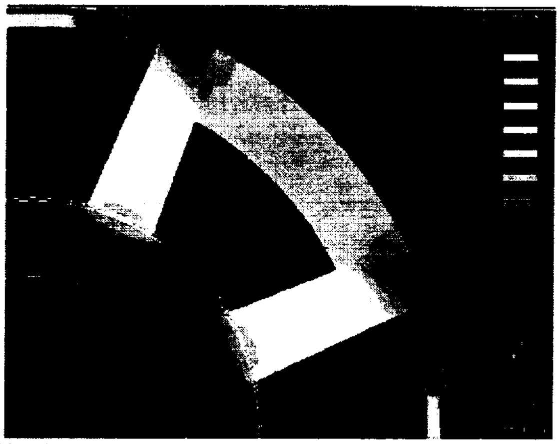


Fig. 4: Numerical results without considering the can

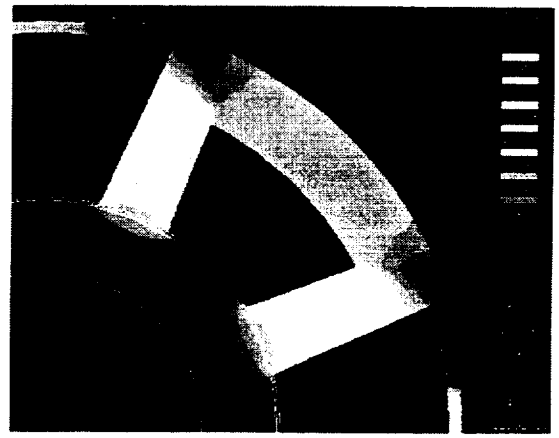


Fig. 5: Calculation considering the can

RUN DOWN TEST RIG

Measurements performed with magnet configurations with and without cans have to be compared to quantify the losses. Two different principle possibilities to measure the Eddy current losses have been conceivable. First, the torque loss could be determined by integrating a torque measurement device in a motor driven thread with an electromagnetic actuator. The selection of the correct measurement range for such a torque metering system would be difficult if the order of the loss moment is unknown, but the losses can be measured at different shaft speeds, if a speed controlled drive unit is available. Secondly, the losses may be derived by performing run down tests of a shaft with an electromagnetic actuator. Kasarda et al. [Kasarda–97] have compared measurements of rotor losses of homopolar with heteropolar bearing designs. Interesting results of these tests have been documented, and therefore it has been decided to design a test rig with a related functionality. To minimize the required effort, the test shaft should be supported by roller bearings. Very precise aerospace bearings are able to operate at high speeds with minimized friction losses, table 4.

bearing type	***B71907E.TPA.P4.UL***
inner/outer diameter	$35*10^{-3}/55*10^{-3}$ m
rotary speed limit, oil breathed	26 000 rpm
friction losses at maximum speed	$< 38*10^{-3}$ Nm

Table 4: data of exemplary roller bearings

Because the shaft is supported in roller bearings, no control loop is needed for the electromagnetic devices, only a current supply is required. The coil current has to be adjustable and recordable. The magnetic force will be measured by Hall sensors mounted on the pole shoes. The calibration of these Hall sensors has to be performed by external load cells to reach a sufficient accuracy, discussed in Knopf et al. [Knopf–98].
A test shaft of 0.3 m length will be accelerated to 12 000 rpm by a 2 kW AC–Servo drive. This drive unit is joint to the test shaft by a disconnectable friction coupling. When the targeted operational speed is reached, the shaft will be manually declutched and the run down curve will be registered. The test rig is presented in fig.6.

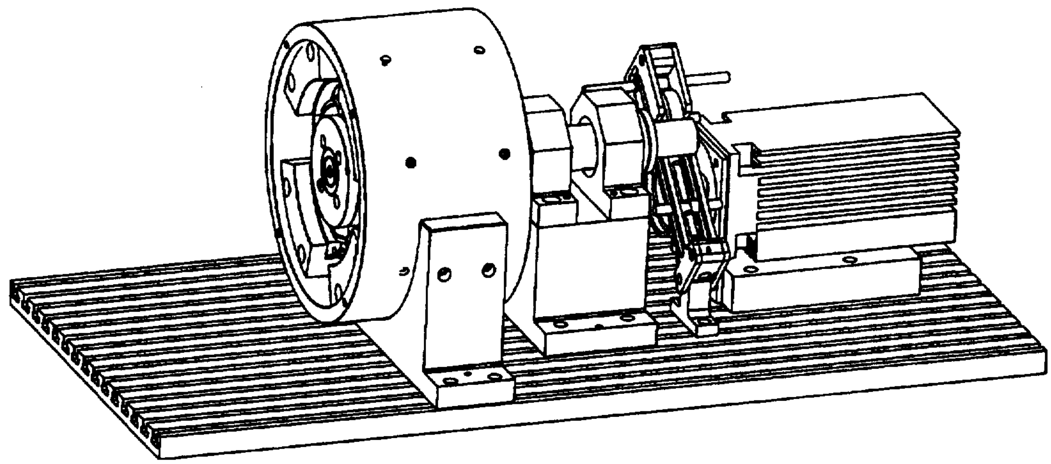


Fig. 6: Test rig for run down tests with different magnet designs

Run down time is depending on the rotors moments of inertia, on roller bearing and air friction and on the deceleration moments produced by the electromagnetic circuit. Due to comparison of different run down times realized with different magnetic circuit configurations, the losses caused by Eddy current and hysteresis can be quantified. The air drag is expected to be nearly constant in all configurations. The sum of mechanical losses may be estimated by subtraction of measurements with and without activated electromagnets. Scheduled measurement configurations are listed below in table 5.

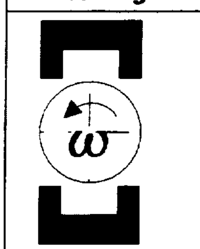
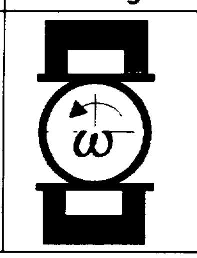
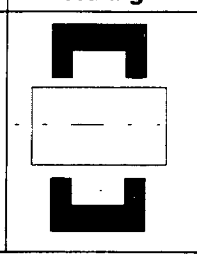
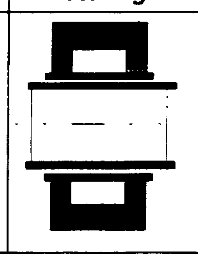
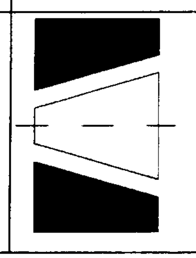
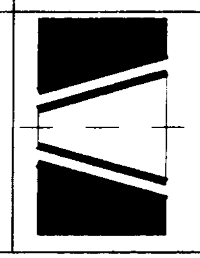
radial heteropolar bearing	***canned radial heteropolar bearing***	***radial homopolar bearing***	***canned radial homopolar bearing***	***conical shaped bearing***	***canned conical shaped bearing***
					

Table. 5: Different magnetic bearing configurations

Because of geometrical aspects,conical shaped bearings in homopolar configuration have been disregarded. The geometrical properties of the test magnets shown in fig.7 are listed in table 6 beside the assumed and calculated magnetic data.

The evaluation of the test results will hopefully lead to an optimal bearing design for canned–motor pumps by means of rotor power and magnetic suspension losses. In addition to the run down tests, flux density measurements by means of Hall sensors are planned. The Hall sensors will be placed in the airgap of the electromagnets or between the laminations and the can. Flux density measurement results should enable the validation of the numerically calculated magnetic fields.

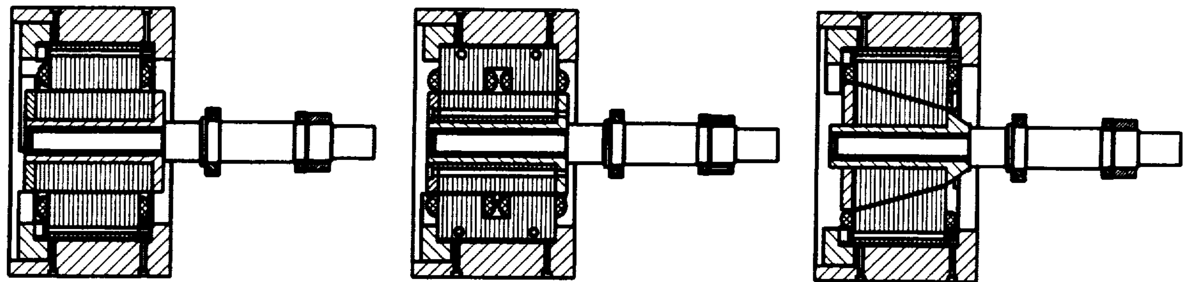


Fig. 7: Overview of different rotor configurations for the run–down test rig

	radial heteropolar magnet	*radial homopolar magnet*	*conical shaped magnet*
inner diameter	95 mm	95 mm	72 – 120 mm
outer diameter	180 mm	180 mm	180 mm
pole area	1290 mm^2	1290 mm^2	1290 mm^2
nominal air gap	1 mm	1 mm	1 mm
flux density B_{max}	1,4 T	1,4 T	1,4 T
coil area A_{coil}	310 mm^2		
conical angle			15 °
calculated radial force	2000 N	2000 N	1250 N
calculated axial force	–	–	380 N

Table 6: design properties of test magnets

POSITION MEASUREMENT

For the exact control of the rotor, its position has to be measured and fed back. Therefore sensors have to be integrated in a location where the collocation conditions are fulfilled. The target of a sensor selection process would be ideally a sensor which is placed in the center of the magnetic bearing.
The selection of possible sensor systems is one of the major tasks in the design process of a canned pump with magnetic bearings. Standard proximity probes are not qualified for this application because of the rough environmental conditions, temperature and aggressiveness of the process fluid. In principle there are three possibilities to meet this difficulty, see fig. 8.
A corrosion resistant sensor with a stainless steel housing may utilized. An additional sealing has to prevent any fluid leakage. The corrosion resistant sensor may be, e.g. an inductive or an Eddy current type. To use sensors with a conventional housing, an in the can integrated ceramic shielding would be helpful. An installation close to the magnetic bearings is possible if the ceramic ring diameter is identical to the stator. Another possibility is to install a ceramic plate, like a small bent disk. It can be used as a kind of window in the metallic can. Both the corrosion resistant sensor and the ceramic shielded sensor would fulfil the conditions for hermetically sealed pumps, only the static sealings are necessary.
If the measurement has to be performed through the can, Eddy current, inductive and Hall sensors represent three possible principles. Utilizing Eddy current or inductive sensors, the high signal attenuation has to be taken into account. Hall sensors enable the measurement of the flux density directly between the pole surface and the stator can.

In coherence with an additionally coinstantaneous available parameter, e.g. the measured current in the magnetic coils, the flux density could be utilized to calculate the air gap in the magnetic circuit and hence the rotor position.

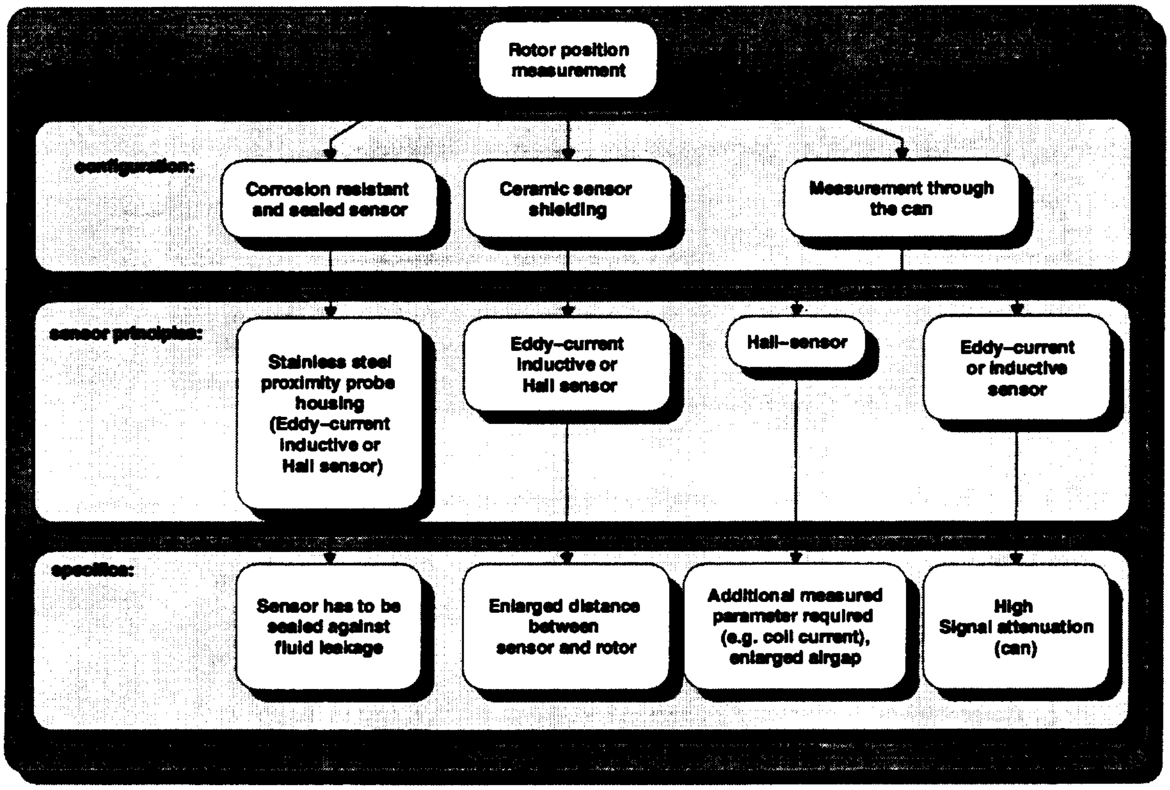


Fig. 8: Rotor position measurement in canned pumps

The design of a future test pump should enable the integration and testing of different sensor principles. At the moment, Hall sensors seem to be the most elegant way for the position measurement. Their advantage is the possibility of a position measurement directly where the force is acting on the rotor. Their first disadvantage is the additional effort to calculate the position out of flux density and, e.g. the coil current. The second disadvantage is the required enlargement of the airgap which leads to a significant reduction of the maximum bearing force. The signal processing must be performed fast enough to satisfy the dynamic characteristics of the entire system.

PUMP PROTOTYPE

In the near future a widespread test program will be executed. Measurements on different magnet configurations are scheduled to get information about their individual behaviour. There will be the examination of force development and force losses by means of Eddy currents and hysteresis effects. At least conventional radial bearings in combination with two single–acting axial bearings (see fig. 9) and conical shaped bearings (see fig. 10) will be tested and compared with each other. At this moment these

two alternatives seem to be the most attractive configurations to reach a satisfiable system performance with acceptable effort. In parallel the numerical calculations will be validated and updated.
The two pump configurations will be installed in the laboratories of the University of Darmstadt and run in a lab testloop under nearly real conditions. The prototypes will be equipped with sensor systems to record all interesting mechanical, electrical, thermal and process data.

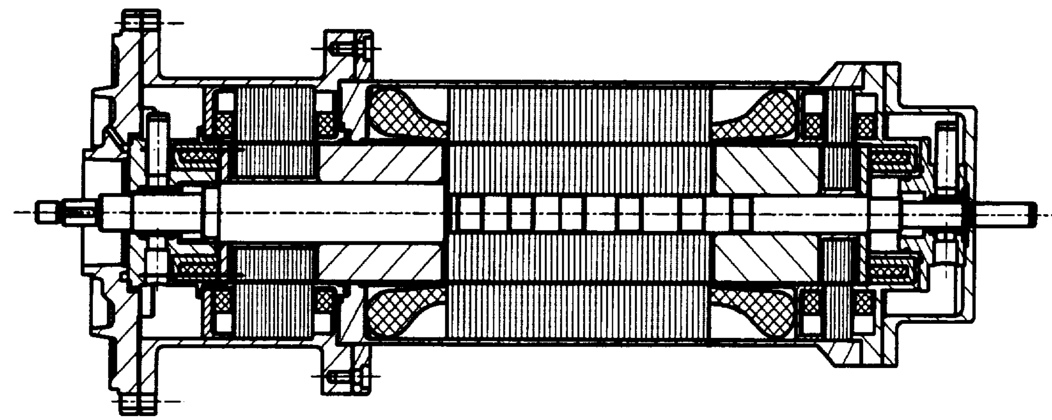


Fig. 9: Canned motor pump with two radial and two single–acting axial bearings

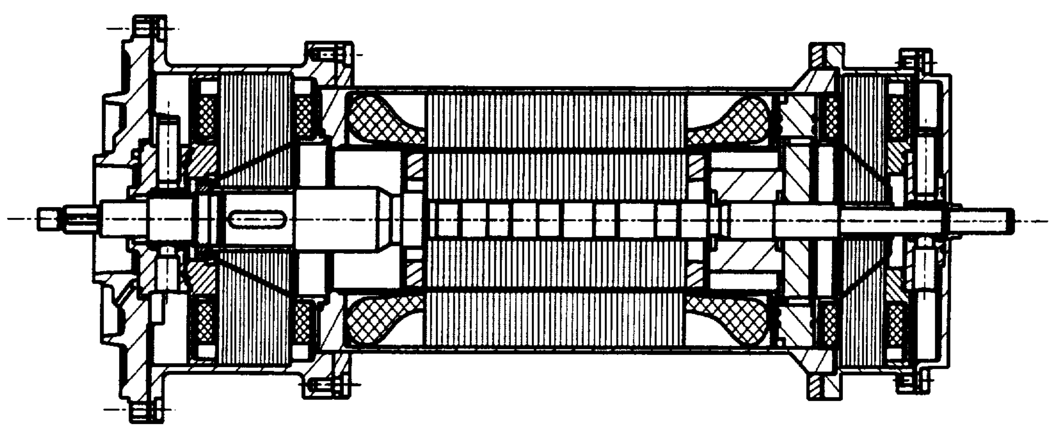


Fig. 10: Canned pump with two conical shaped bearings

The design of the test pumps enables the integration of different sensor concepts. Especially the ceramic sensor shielding and the integration of Hall sensors are scheduled. The ceramic shielding will be realized as a concentric ring next to the magnetic bearing. An even better solution for the future would be the entire substitution of the metallic can through a ceramic one (e.g. ZrO_2). Such ceramics are already used to build shieldings for permanent magnetic couplings in mag–drive pumps, the second design alternative for a hermetically sealed pump. By the time they are available with an acceptable thickness they will reduce can losses to nearly zero. They would enable the use of a wide range of different sensor principles Also they provide interesting characteristics as backup bearing materials.

SIMULATION OF THE CLOSED–LOOP SYSTEM

Before the realization of the prototype pump, a nonlinear time domain based simulation model of the pump with two different bearing configurations has been developed. The first one has two radial bearings and a double acting axial bearing, the second consists of two conical shaped bearings. Fig. 11 shows the MatrixX simulation model of the canned pump. The block *External Forces* describes one possible set of hydraulic forces acting on the rotor. The overall hydraulic force in a pump consists of a static part which is variable in direction and value depending on the operation conditions and a dynamic part which depends on the rotational speed, number of impeller blades, etc..

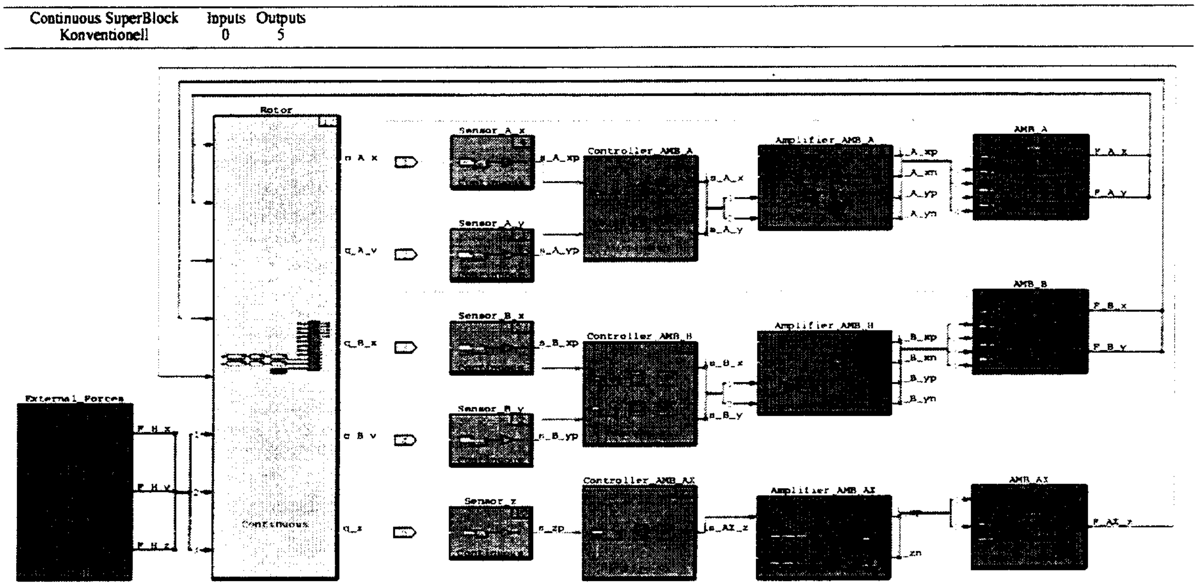


Fig. 11: Model of a AMB system with two radial and one double–acting axial bearing

In a first step the sum of static and dynamic hydraulic forces have been assumed as one harmonic force rotating with shaft speed. The rotor is modelled as a rigid body system, an acceptable simplification because its first bending mode eigenfrequency (approx. 275 Hz) is clearly above the frequencies of the dynamic excitations (1. assumption: 50 Hz). The displacements are measured by sensors, which have at this state of the simulation a P characteristic, in five degrees of freedom to be fed forward to the controller. The PID–Controllers set the required currents for the amplifiers which have a PT1 performance and are limited to 70 VDC and 10 A. The absolute magnetic bearing force depends on this current and the actual position of the rotor. The closed loop system allows the prediction of the system performance especially an estimation of the rotor displacements in the magnetic bearings considering the mechanical and process forces. The external hydraulic and mechanical forces and the resulting magnetic bearing forces acting on the rotor are shown in fig. 12. Fig. 13 displays the simulated rotor displacements at the bearing locations.
This simulation model helps to select the optimum hardware to reach a satisfactory system performance for the given loads and operation conditions.

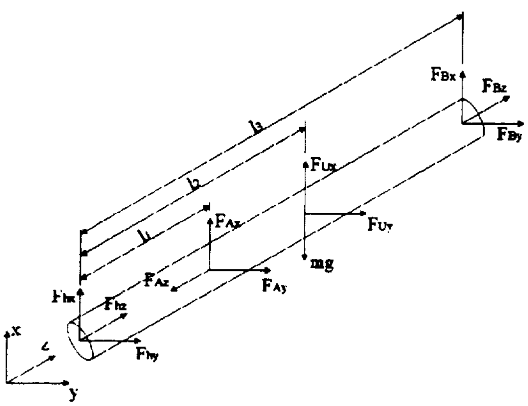


Fig. 12:.The on the pump rotor acting forces

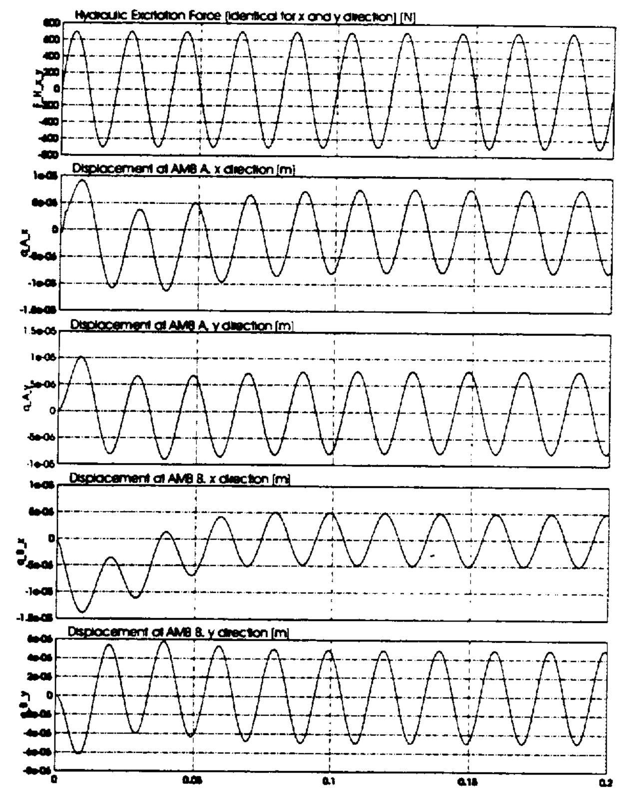


Fig. 13: Simulation results; displacements in the AMB's

SUMMARY

Magnetic bearings for hermetically sealed pumps have been introduced years ago. Despite increasing demands for these machines the magnetic bearing technology has not yet been leveraged in chemical pumps. The relatively high costs and the additional effort to integrate magnetic bearings have decelerated adequate developments.

First steps to a systematic approach to an enhanced design of a canned motor pump with active magnetic bearings have been introduced. Different bearing configurations and applicable sensor principles have been compared. In completion of analytical and numerical calculations, the design of a run down test rig to quantify Eddy current and hysteresis losses has been quantified. A conventional design with two radial and two single-acting axial bearings and a design with conical shape bearings have been presented as the most attractive solutions under consideration of pump specific and manufacturing aspects.

Two prototype pumps are scheduled by the Institute of Mechatronic Systems for the next time. The aim of this study is to introduce new ideas for the development of hermetic sealed pumps with magnetic bearings.

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REFERENCES

Voll– 93 Vollmüller, H., Fischer, K.: Canned Motor and Magnetically Coupled Pumps Applications, Operations and Maintenance in a Chemical Plant, Pump Symposium, Texas A&M University, Texas, USA, 1993

Neum–94 Neumaier, R.: Hermetische Pumpen, Verlag Faragallah, Sulzbach, Germany, 1994

Hergt–99 Hergt, P.: Pump Research and Development: Past, Present and Future, Journal of Fluids Engineering, June 1999

Allair–89 Allaire, P. E., McDonald, J. P.: Design, Construction and Test of Magnetic Bearings in an Industrial Canned Motor Pump, World Pumps, Sep. 1989

Marscher–91 Marscher, W. D., Jen, C.: Development of an Active Magnetic Bearing for a 50 HP Canned Motor Pump, ROMAC '91 Magnetic Bearings and Dry Gas Seals, International Conference and Exhibition, University of Virginia, USA, 1991

Hanson–92 Hanson, L., Imlach, J.: Development of a Magnetic Bearing API Process Pump with a Canned Motor, 9th International Pump Symposium, Texas A&M University, USA, 1992

Gempp–96 Gempp, T., Schöb, R.: Design of a Beringless Canned Motor Pump, 5th International Symposium on Magnetic Bearings, Kanazawa, Japan, 1996

Lähteen–98 Lähteenmäki, J. K., Lantto, E. J. : Optimization of the Conical Angle of Cone–shaped Active Magnetic Bearings, Proceedings of the Sixth Int. Symp. on Magnetic Bearings, MIT, Cambridge, USA, 1998

Canders–97 Canders, W.–R., Lee, W. L. : Integrated Radial and Axial Low Cost Magnetic Bearings, Proceedings of MAG '97, Industrial Conf. and Exhib. on Magnetic Bearings, Alexandria, USA, 1997

FLUX2D–98 FLUX2D: CAD Package for Electromagnetic and Thermal Analysis using Finite Elements, CEDRAT, Meylan Cedex, France, cedrat@cedrat–grenoble.fr

Vetter–98 Vetter, G. (Editor): Leckfreie Pumpen, Verdichter und Vakuumpumpen, Vulkan Verlag, Essen, Deutschland, 1998

SchTr–93 Schweitzer, G.; Traxler, A.; Bleuler, H.: Magnetlager, Grundlagen, Eigenschaften und Anwendungen berührungsfreier, elektromagnetischer Lager, Springer–Verlag, Berlin, Germany, 1993

Vetter–98 Vetter, G. (editor): Leckfreie Pumpen, Verdichter und Vakuumpumpen, Vulkan–Verlag, Essen, Germany, 1998

Kasarda–97 Kasarda, M. E. F., Allaire, P. E., et al.: Comparison of Measured Rotor Power Losses in Homopolar and Heteropolar Magnetic Bearings, Proceedings of MAG ´97, Industrial Conf. and Exhib. on Magnetic Bearings, Alexandria, USA, 1997

Knopf–98 Knopf, E., Nordmann, R.: Active Magnetic Bearings for the Identification of Dynamic Characteristics of Fluid Bearings, 6th Int. Symp. on Magnetic Bearings, Massachussetts Institute of Technology, Cambridge, USA, 1998