NEW APPROACHES FOR AXIAL MAGNETIC BEARINGS

***W.-L. LEE, W.-R. Canders, *W. Schumacher**
*Institute of Control Engineering, Institute of Electrical Machines, Traction and Drives
Technical University of Braunschweig, Germany
WL.LEE@TU-BS.DE

ABSTRACT
This paper describes the design, construction and tests of two new axial bearing concepts for rotating machines. They are mainly developed to reduce the total rotor length or the moment of inertia of the axial bearing segment . This leads to a stiffer rotor, and consequently a controller without considering the bending modes may be used although a higher speed of rotation is allowed.
In the first approach the axial thrust disk is integrated into one single bearing element including control windings for three directions. Hence an extra axial bearing segment is not necessary here.
In the second approach a tubular linear actuator is developed to serve as an active axial bearing. Instead of a thrust-disk, a rotor with a teeth contour is used here to provide axial force.

1. INTRODUCTION

Due to its lubricant and wear free operation with controllable stiffness and damping, magnetic bearings have been used to replace conventional bearings in some industrial applications. However, compared to conventional bearings with the same load capacity, the size of a magnetic bearing is larger. A thrust-disk-segment is normally added to the rotor to stabilize the axial motion (**FIGURE 1a**). This increases the total length of the rotor. In addition, the thrust-disk segment has a larger moment of inertia and its diameter is normally larger compared with the radial bearing, thus decreasing the bending frequency of the rotor.
As reported in MAG97' [1] , a teeth-contour **(FIGURE 1b**) is introduced into the force surface of the bearing stator and into rotor surface to avoid a thrust-disk. Due to the reluctance forces between the teeth, the rotor is inherently centered in the axial direction. Therefore it reduces the number of controllers and power amplifiers as well as position sensors. Consequently, a low cost implementation is achieved.
However, with this teeth-contour the load capacity in radial direction is also reduced. Furthermore, any axial deviation of the rotor changes the surface area of the air gap . As a result, the radial bearing forces are coupled with the axial deviation of the rotor. If the axial motion is not actively controlled and is not sensed, any axial impact may corrupt the stability of the rotor.
To overcome these problems, two types of active axial magnetic bearings have been developed to raise the bending eigenfrequencies of the rotor:

- A combined active bearing (**FIGURE 1c**)
- A tubular linear actuator (**FIGURE 1d**).

Both of them are using rare-earth permanent magnets to enhance their capacity density.

2. PRINCIPLES OF OPERATION

2.1 Combined active radial and axial bearing

FIGURE 2 shows the operation principle of the combined bearing. The thrust-disk and the radial bearings are combined into one single bearing element including control windings for three directions. The PMs are used to create a bias flux for both radial and axial bearings simultaneously. Since a permanent magnet has a low permeability (similar to an air gap), the electrically controlled fluxes should not be driven through it . For a radial bearing, the controlled fluxes flow on a radial plane to change the bias flux from axial direction (non-coplanar) for getting radial force (**FIGURE 2b**). Meanwhile a control-flux is introduced in the axial plane to create axial force. (**FIGURE 2a**)
Using this approach, a combined bearing unit with three actively controlled axes in a compact size is possible.

2.2 Tubular linear actuator

However a thrust-disk induces additional air friction losses due to its large diameter. Moreover, it is difficult to shrink it on the rotor and to achieve a planar motion even under rising temperatures due to eddy current losses.

That is why a tubular linear actuator with a small diameter of rotor has been developed. In this case not Maxwell's normal forces – as common in thrust disk type bearings – are used, but the lateral force is applied to drive the axial motion.

FIGURE 3 shows the field plot of this bearing. It describes the operational principle.

High performance permanent magnet-rings are placed opposite to each other on the stator side, creating a high concentrated flux on the rotor sided teeth-contour.

A control-coil is added to control the axial motion. When there is no control current applied, the high flux density on the both edges of a tooth is symmetrical and keeps the bearing inherently stable (**FIGURE 3a**). Compared with an active magnetic bearing usually showing negative stiffness, this one has a positive stiffness. If the control-coil is engaged, the control-flux will change the flux distribution to an unsymmetrical one thus producing an axial propulsive force (**FIGURE 3b**). Using this approach, an inherently stable active axial bearing is achieved and it yields a controllable axial degree of freedom. Additionally the outer diameter of the rotor becomes smaller than the inner diameter of the motor-stator. This relieves some difficulties at the mounting of the axial bearing and will significantly reduce the windage losses.

3. ROTORDYNAMIC ANALYSIS

In this section the rotordynamics of the proposed system is examined. Finite element models are used to predict the first bending modes and the eigenfrequencies for different rotors (free rotor, zero speed).

It is well known from literature and experimental measurements [2], that with rotors possessing a small moment of inertia, the forward and the backward whirling modes are nearly coincident. If a thrust disk is introduced, the eigenvalues are split up significantly by the gyroscopic effects. As a magnetic bearing typically excites the forward and the backward whirling mode as well, the first critical speed of a rotor with a thrust disk will be lower than without it.

FIGURE 1 shows the different system designs with the conventional approach (Rotor 1- **FIGURE 1a**) as the reference for comparison. It is levitated by two non-coplanar PM biased radial bearings. However the axial bearing consists of only a pair of electrical magnets and a thrust disk.

TABLE 1 summarizes the comparison of rotor-dynamics, load capacities and rotor length between the different systems. All quantities are normalized to the reference system.

It is clear to see that with the new concepts, the first bending frequency is raised to a range from 120% to 176% compared to the system with an extra thrust-disk segment. This allows a higher operational speed with a rather simple controller which has only to care for the rigid body motion. Besides, higher axial capacities are achieved in the new designs.

TABLE 1: System comparison

ROTOR	1 (Ref.)	2	3	4
Load Cap. (Rad.)	**1 (800N)**	**0.60**	**1.00**	**1.0**
Load Cap. (Axial)	**1 (1000N)**	**0.50**	**2.50**	**1.5**
Length of rotor	**1 (340mm)**	**0.76**	**0.82**	**1.0**
First Bending	**1 (2057Hz)**	**2.07**	**1.76**	**1.2**

4. TEST RIGS AND MEASUREMENT

Two prototypes of bearings are built to verify the new design concepts. The axial static characteristics of these were investigated. They are namely the bearing stiffness k_x and force-current factor k_i. Normally, a magnetic bearing force can be linearized as:

$$F = k_x \cdot x + k_i \cdot i \qquad (1)$$

where x is the rotor displacement and i is control current.

4.1 Test rig for combined bearing

FIGURE 4a shows the realized combined bearing unit and its rotor. In order to investigate the characteristic values in the axial direction, the rotor is fixed in both radial degree of freedoms. **FIGURE 4b** is the simulated plot of the force-mmf-displacement dependency. (mmf: magneto motive force). It shows that a high axial force capacity can be also achieved without using an extra thrust-disk segment.

Due to the intensive bias flux on the axial bearing, a small mmf is able to create a huge axial force, certainly it is also easier to fall into the saturation area.

We are preparing for the experiment to verify this simulation result.

4.2 Test rig tubular linear actuator

FIGURE 5a shows the test rig for tubular linear actuator. **FIGURE 5b** is the measured force-mmf-displacement dependency.

This bearing has a positive stiffness, therefore it is inherently stable without any control flux.

It shows a very high axial force capacity without using a thrust-disk. In our measurements only 40% of the possible mmf (ampere-turns) were used to avoid overheating by a long time applied DC current during the static tests. The peak force can be still enhanced with a higher current linkage.

The measurement reveals that this bearing possesses an excellent linearity in a wide range of mmf. This eases the controller design for the axial bearing.

5. CONCLUSION

Two new types of axial bearings are proposed. They fulfil the requirements of raising the bending eigenfrequencies of the rotor and of providing a high force capacity.

However there are also some drawbacks existing with those systems.

Related to the combined bearing, there is a coupling between radial and axial bearing. Compared with the thickness of the bias PM, the air gaps of the axial bearing are small. Therefore the control flux of the axial bearing passing through the air gaps can be neglected.

However, with a very large control flux, e.g. during the levitating-up phase, it will influence the bias flux of the radial bearing. As a result, a bending moment is applied on the rotor. This effect should be considered while designing the controller.

Considering the tubular linear actuator, it is an inherently stable bearing without applying any control current. Any deviation from center position will induce a restoring force to center the rotor automatically. An active control current can be also applied to keep the rotor in center position even though any axial load exists.

However, a very strong radial disturbing force will arise if the rotor is not kept in the center of the bearing. This is especially critical during the “taking off” phase. The radial bearings in this system have to provide enough overload capacity to conquer the large “locked on“ force from the direct contact of rotor and PM.

REFERENCES

1. W.-R. Canders, W.-L. Lee, Integrated radial and axial low cost magnetic bearing, Proc. of MAG ’97, Virginia, USA, 1997.

2. W.-R. Canders, R. Larsonneur, N. Ueffing, U. Schrader – Hausmann: MTG 400- a magnetically levitated 400 kW turbogenerator system for natural gas expansion , Proc. 4th ISMB, Aug. 94, ETH Zurich

ACKNOWLEDGEMENTS

This work is based upon research supported by the Deutsche Forschungsgemeinschaft under grant SCHU-977/4-1.

The authors are grateful to B. Amlang, A. Rusniok, H.-J. Bonney and S. Schütze for their technical assistance.

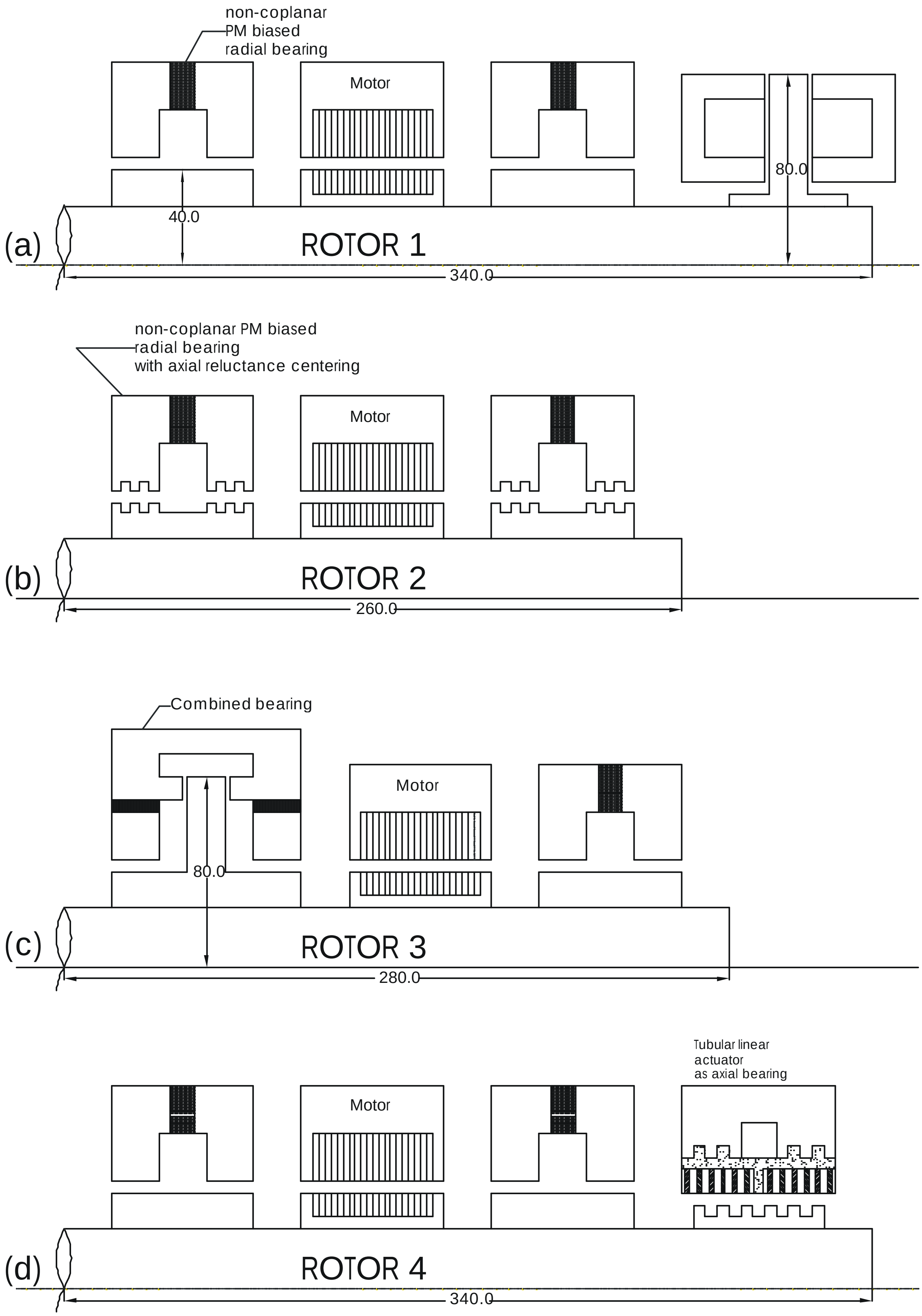

non-coplanar
PM biased
radial bearing
Motor
80.0
40.0
(a)
ROTOR 1
340.0
non-coplanar PM biased
radial bearing
with axial reluctance centering
Motor
(b)
ROTOR 2
260.0
Combined bearing
Motor
80.0
(c)
ROTOR 3
280.0
Tubular linear
actuator
as axial bearing
Motor
(d)
ROTOR 4
340.0

FIGURE 1

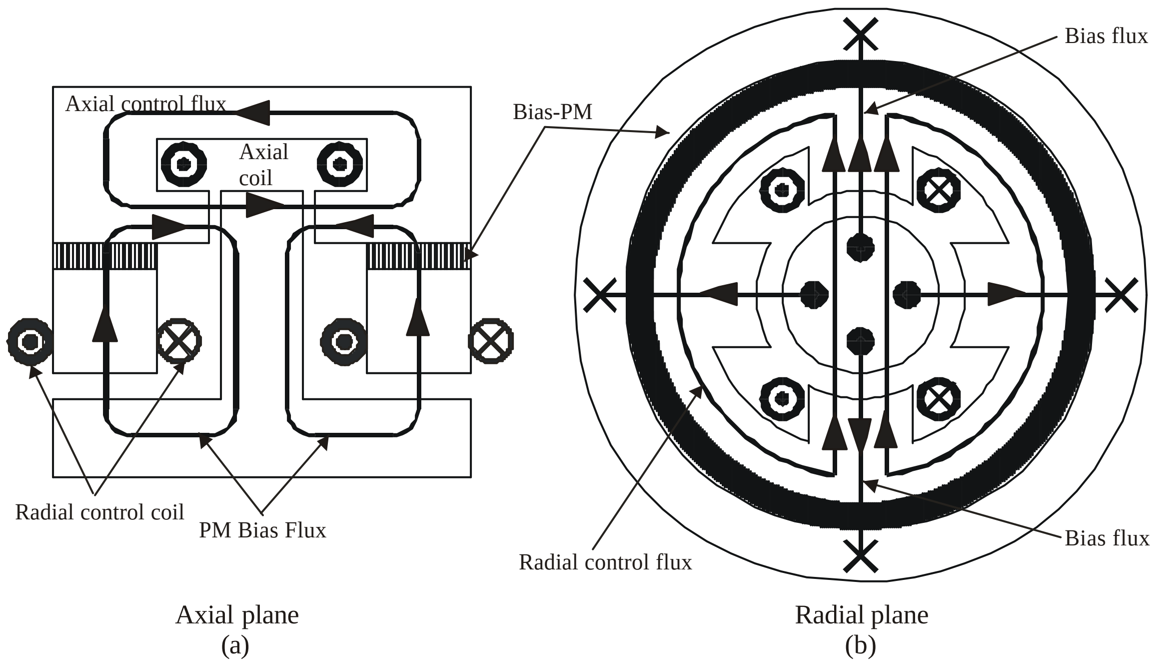
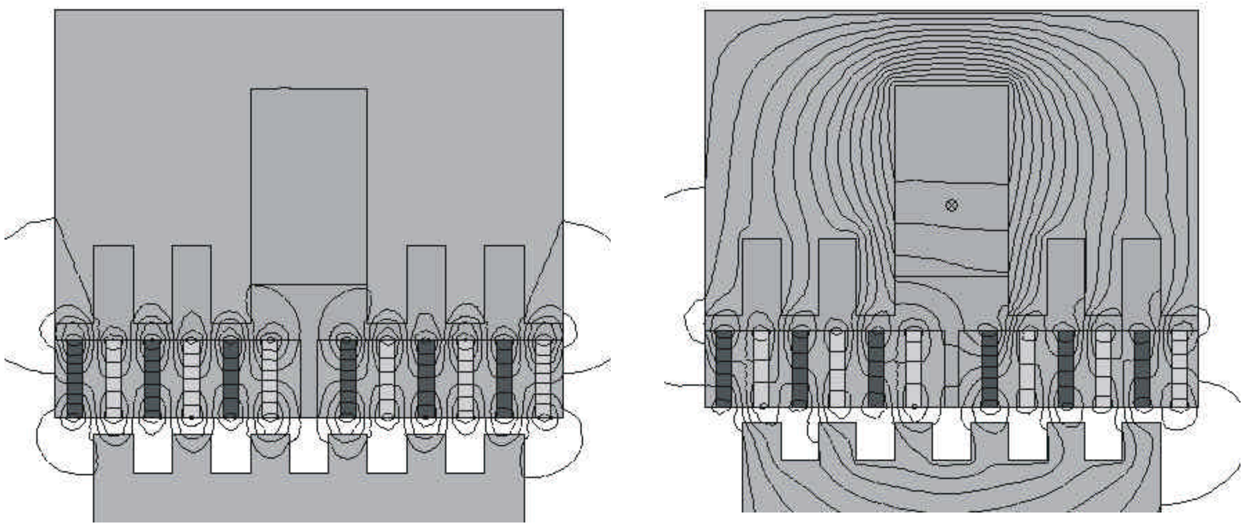


FIGURE 2: Operational principle of combined bearing



(a) Control flux=0 (b) Control flux applied

FIGURE 3: Field plot of the tubular linear actuator

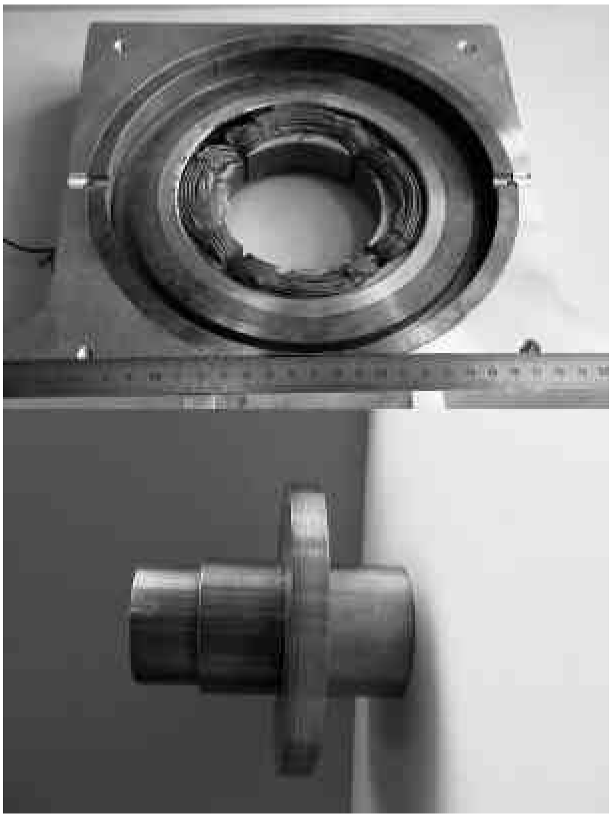


FIGURE 4a: Combined bearing , ½ stator and rotor

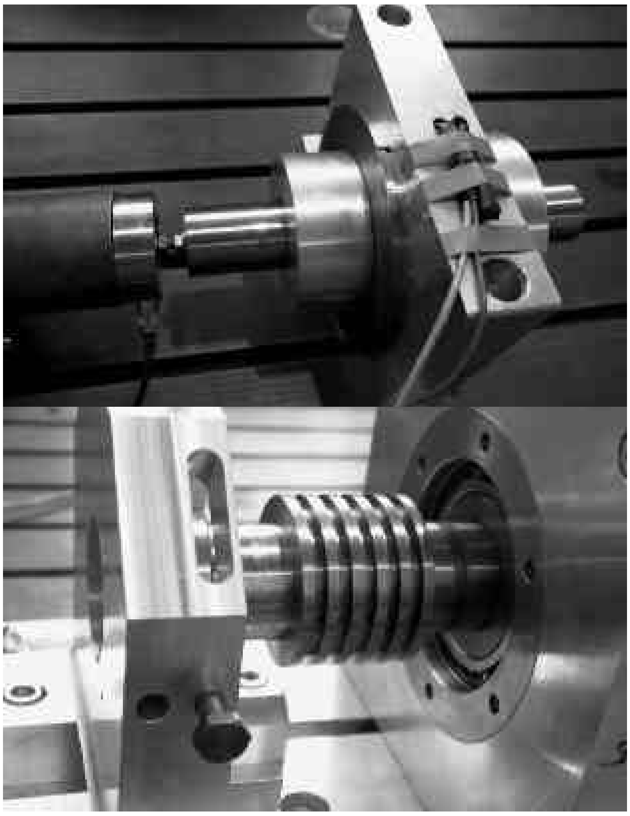


FIGURE 5a: Tubular linear actuator stator and rotor

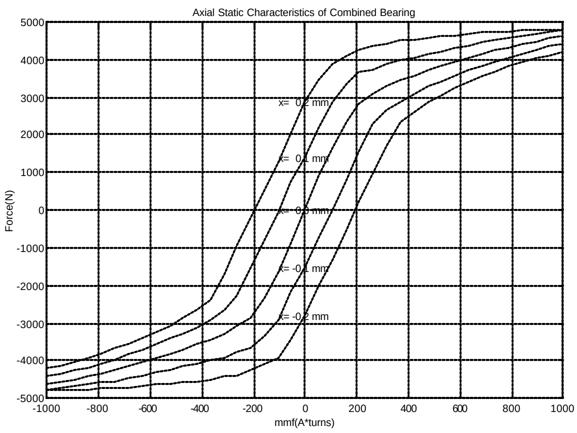


FIGURE 4b: Axial characteristics of combined bearing, simulated Force-mmf-Displacement dependency plotted on F-mmf coordinated with parameter x (displacement)

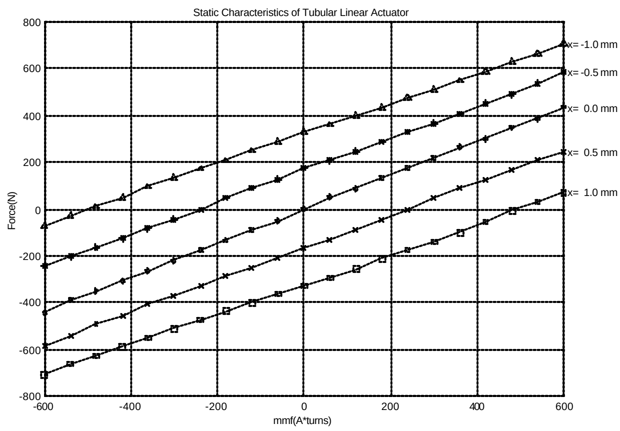


FIGURE 5b: Axial characteristics of tubular linear actuator, measured Force-mmf-Displacement dependency plotted on F-mmf coordinated with parameter x (displacement)