Application of Zero-Bias Current Active Magnetic Bearings to Flywheel Energy Storage Systems

S. GÄCHTER H. KAMENO

Flywheel energy storage systems are a target application for active magnetic bearings. Its competitiveness to roller bearings, however, depends strongly on the achievable bearing loss reduction. Conventional active magnetic bearings use linearizing bias currents superposed by control currents. These currents are the source of the bearing losses. Therefore, this paper discusses in the first part these losses and methods to reduce them. In the second part, a particular method is proposed, i.e. instead of bias currents, feedback linearization is applied. The simulation showed that control performance similar to that of bias current types could be achieved with these zero-bias current active magnetic bearings, if the feedback linearization is combined with a control force sign based commutation law.

Key Words: *bearing loss reduction, control force sign based switching, feedback linearization, flywheel energy storage system, zero-bias current active magnetic bearing*

1. Introduction

The commercialized flywheel energy storage systems provide a high power and a medium energy density per volume compared with other energy storage devices. The high power density, which can hardly be reached with the other storage devices[1], is the main advantage of the flywheel system. In contrast, the energy cycle efficiency, i.e. the charge-discharge efficiency considering the idling loss, is not competitive[2]. The application area of flywheel energy storage systems can only be widened, if the idling loss is decreased.

Therefore, the paper is organized as follows: In Section **2** and **3**, the flywheel energy storage system is introduced. The idling loss is discussed in detail and the bearing loss of either roller bearings or active magnetic bearings is identified as the major problem. It is argued, therefore, that the competitiveness of active magnetic bearings depends on the attainable bearing losses. In Section **4** and **5**, the active magnetic bearing losses are analyzed and different reduction schemes are proposed. In Section **6**, a particular reduction scheme is discussed, and in Section **7**, its performance is analyzed. The conclusion is given in Section **8**.

2. Flywheel Energy Storage Systems

2. 1 Storage Capacity

The high power density has been reached, because of the advent of composite flywheels. Since the maximum energy density per volume of the flywheel is directly proportional to the allowable material stress, high-strength composites led to high-speed flywheel energy storage systems.

Such systems have proved their usefulness, both as (i) regenerative energy storage systems, and as (ii) power stabilization systems. In both cases idling losses do not affect considerably the performance, because for (i) the charge-discharge cycles are of short duration, and because for (ii) the idling loss is compensated by the primary source. However, idling loss is still a crucial problem when using flywheel energy storage systems for (iii) load leveling or as (iv) independent energy source.

2. 2 Idling Loss

The idling loss is composed of bearing loss, windage loss, and loss of the electrical machine, cf. **Fig. 1**. Operating the system under vacuum and with an appropriate design for the electrical machine, the windage and the motor-generator losses are reduced so that the bearing loss becomes the decisive factor. This loss depends on the kind of bearings, which are either roller bearings or magnetic bearings. Both types operate under vacuum and at high rotational speed.

3. Comparison of Bearing Systems

Parallel to the advancement in material science the progress in control engineering led to active magnetic bearings. Such bearings, which are composed of electromechanical actuators, power amplifiers, controllers, and position sensors, provide contact-free suspension. Thus, friction is not present and lubrication is not needed. Therefore, the active magnetic bearing lifetime outperforms the mechanical bearing lifetime. These characteristics make active magnetic bearings suitable for flywheel energy storage systems[3]. However, in terms of losses, the active magnetic bearings do not necessarily perform better than roller bearings. This is especially true when compared with bias-current linearized active magnetic bearings. Furthermore, the high cost, the complexity, and the low stiffness have to be considered when designing a flywheel

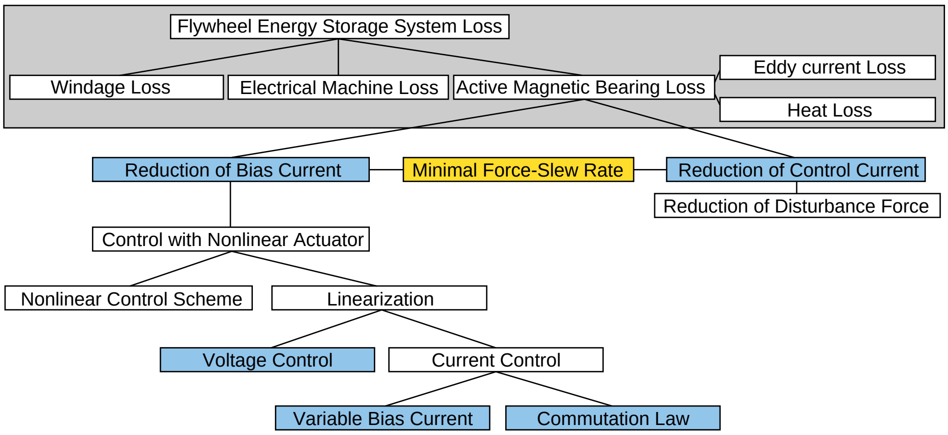

Fig. 1 Loss reduction methods for flywheel energy storage systems

energy storage system with magnetic bearings. Therefore, most of the commercialized flywheel systems are equipped with roller bearings.

Active magnetic bearings will be applied in flywheel systems only if the bearing loss is substantially reduced and if such flywheel systems gain a market in load leveling and energy storage applications.

How the active magnetic bearing losses can be reduced is discussed in the following sections.

4. Bias Current Active Magnetic Bearings

The flywheel system uses in general three active magnetic bearings to suspend the rotor, i.e. two bearings in radial direction and one in axial direction.

However, the axial bearing is not considered in the following unless otherwise stated.

The geometric setup of the rotor-bearing system including the bearing planes A and B is depicted in **Fig. 2**. The radial active magnetic bearings control the rotor position in x and y direction. For each coordinate x_A, x_B, y_A and y_B an actuator is necessary. Because of the considered symmetry, only 1DOF, the coordinate x_A, is treated in the following.

The actuator is composed of a pair of electromagnets: one is located at $x_A > 0$ and the other at $x_A < 0$. Each electromagnet exerts a tractive force. Therefore, with the present actuator configuration 1DOF can be fully controlled.

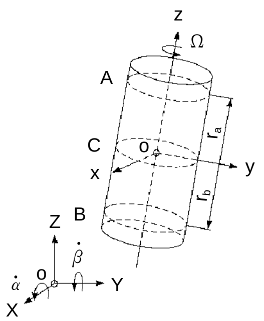

Fig. 2 Geometric setup of the rotor-bearing system

The bearing planes are A and B. The reference frame is OXYZ and the body-fixed frame is oxyz. The origin of the body-fixed frame is equal the center of mass C. The appendant angles are α and β. The rotational speed is Ω.

The resulting force F_{Ax} acting on the rotor is given by

$$F_{Ax} = k_A \left(\frac{i_1}{x_{0A} - x_A}\right)^2 - k_A \left(\frac{i_3}{x_{0A} + x_A}\right)^2 \tag{1}$$

with the condition $x_{0A} > |x_A|$.

The magnetic force coefficient is denoted by k_A, the actuator air gap is denoted by x_{0A}, the current of the actuator at $x_A > 0$ is denoted by i_1, and the current of the actuator at $x_A < 0$ is denoted by i_3. Obviously, the actuator characteristic is (i) nonlinear and (ii) has two control inputs. Therefore, conventional active magnetic bearings use linearizing bias currents with superposed control currents to obtain a linear SISO characteristic of the actuator. The currents in the above relation (1) are replaced by the bias current i_{0xA} superposed by the control current i_{xA} as follows: $i_1 = i_{0xA} + i_{xA}$ and $i_3 = i_{0xA} - i_{xA}$, cf. **Fig. 4**. Then the force is given by

$$F_{Ax} = k_A \left(\frac{i_{0xA} + i_{xA}}{x_{0A} - x_A}\right)^2 - k_A \left(\frac{i_{0xA} - i_{xA}}{x_{0A} + x_A}\right)^2 \tag{2}$$

with the condition $i_{0xA} > |i_{xA}|$.

This relation is linear for a sufficient small control current and rotor position compared with the bias current and actuator air gap respectively, cf. **Fig. 3**. Therefore, the actuator characteristic is linearizable, which results in a linear actuator model suitable for linear control design.

4. 1 Jacobian Linearization

The Jacobian linearization of (2) at the operating point $x_A = x_{A0}$ and $i_{xA} = i_{xA0}$ results in

$$\Delta F_{Ax} = \left.\frac{\partial F_{Ax}}{\partial x_A}\right|_{\substack{x_{A0} \\ i_{xA0}}} \Delta x_A + \left.\frac{\partial F_{Ax}}{\partial i_{xA}}\right|_{\substack{x_{A0} \\ i_{xA0}}} \Delta i_{xA} \tag{3}$$

The current-displacement-force relation at the operating point $x_{A0} = 0$ and $i_{xA0} = 0$ is

$$F_{Ax} = \underbrace{4k_A \frac{i_{0xA}^2}{x_{0A}^3}}_{k_{pxA}} x_A + \underbrace{4k_A \frac{i_{0xA}}{x_{0A}^2}}_{k_{ixA}} i_{xA} \tag{4}$$

where k_{pxA} is the position stiffness and k_{ixA} the current stiffness.

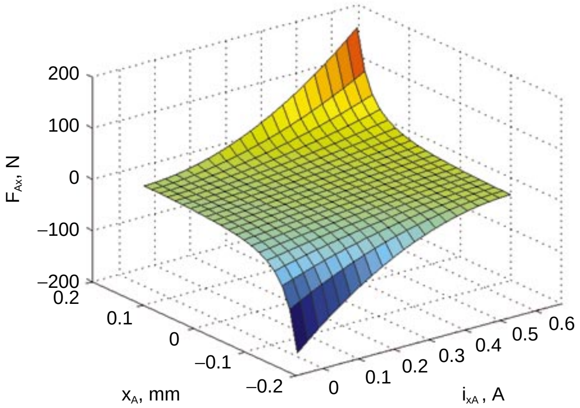

Fig. 3 Actuator characteristic in x_A direction

Depicted is the graph of the force F_{Ax} in function of position x_A and actuator current i_{Ax}. The graph is computed for a bias current of i_{Ax} = 0.3A, a magnetic force coefficient of k_A = 1.34 10-6A^2 / Nm^2 and an actuator air gap of x_{0A} = 0.2 mm.

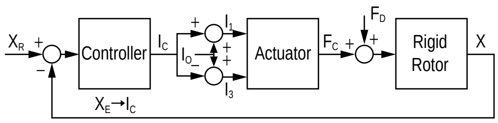

Fig. 4 Block diagram of bias current active magnetic bearing

5. Loss Reduction Methods

The active magnetic bearing losses consist of heat and eddy current losses, which are inherent in electromechanical systems with rotating parts. The eddy current loss is an electromagnetic drag loss and affects therefore directly the idling loss, whereas the heat loss affects indirectly the idling loss. Both kinds of loss are due to the actuator currents, cf. **Fig. 1**. As discussed, conventional active magnetic bearings use bias currents to obtain a linear actuator characteristic. The total actuator current for one coordinate is constant and is equal to two times the bias current. Thus, the bearing loss is independent of the control current and the bias currents become the main source of the bearing losses. Therefore, considerable efforts have been undertaken to reduce the bias currents.

5. 1 Reduction of Bias Current

The reduction of the bias current constrains the validity of relation (3) and, therewith, the stability region for an appropriate linear controller[4]. Thus, it is necessary to consider the nonlinearity of the actuator, either in the control design by using an appropriate control approach such as gain scheduling[5] and sliding mode, or in the plant model by applying an appropriate linearization method, cf. **Fig. 1**. The latter approach will be discussed in Section **6** for zero-bias current active magnetic bearings.

However, together with the bias currents the control currents have also to be reduced to increase the flywheel system efficiency.

5. 2 Reduction of Control Current

The suspension of the flywheel assembly by active magnetic bearings is a regulation process, i.e. the rotor has to be maintained at the same position in presence of disturbances. Therefore, reducing the control current requires the reduction of the disturbance forces, cf. **Fig. 5**. These forces can be classified as static and dynamic loads. In case of the flywheel system, the static loads are due to the gravitational force, and the dynamic loads are due to the unbalance, the sensor-ring eccentricity, the forces necessary to pass bending critical speeds, the acceleration forces such as shock and vibration, the gyroscopic action forces, and the electrical machine forces. The diversity of the disturbance force nature makes it necessary to treat each case separately. The static loads due to gravitational force are canceled by using permanent magnetic bearings in axial direction. The dynamic loads due to rotor unbalance and sensor-ring eccentricity are reduced by cancellation of the synchronous disturbance signals in the control loop. Thus, the rotor spins around its inertial axis and not around its geometrical axis. The reduction of the dynamic loads due to acceleration forces is not straightforward. An indirect approach is to stiffen the system by increasing the rotational speed. The remaining dynamic loads depend strongly on the application of the flywheel energy storage system. Therefore, disturbances forces cannot be reduced completely and the actuators must guarantee a minimal force slew-rate $\Delta F/\Delta t$.

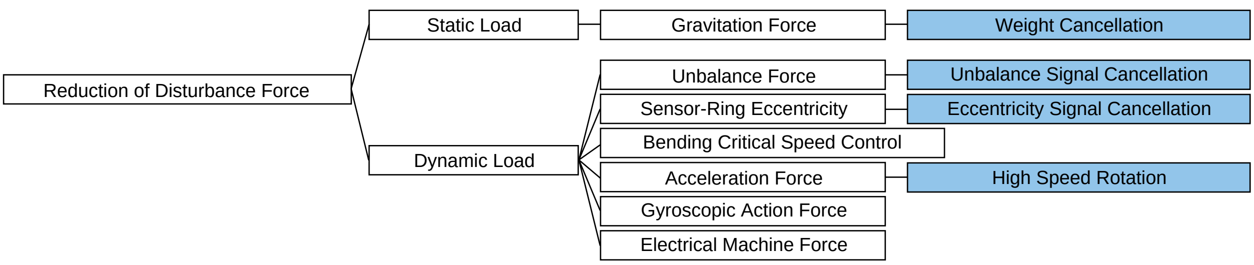

Fig. 5 Reduction method of control current

6. Zero-Bias Current Active Magnetic Bearings

In case of zero-bias active magnetic bearings, the force-slew rate depends on the control current. If the control current is zero, then the slew-rate is zero too. Therefore, a sufficient slew-rate for small control current is guaranteed only if the actuator characteristic (1) can be linearized. Because the common relation (4) is no longer valid for zero-bias current, another linearization method has to be found.

6. 1 Feedback Linearization

Feedback linearization is the transformation of nonlinear system dynamics into, at least partially, linear ones. However, input-output feedback linearization using the systematic Lie algebra approach is not applicable for zero-bias current active magnetic bearings, because of the zero force point. This requires another method for linearizing the actuator characteristic, cf. **Fig. 1**.

One approach is to use active magnetic bearings configured in voltage control. Then input-output linearization becomes possible[6], but this approach creates potentially unbounded voltage levels[5].

Another feedback linearization approach is to set the actuator currents equal the inverse of the current-displacement-force relation given by equation (1). However, the actuator has two control inputs and the problem arises how the currents have to be chosen, such that the resulting force is well defined. Possible solutions are either variable bias current that decreases when approaching the working point, or switching properly the control inputs.

6. 2 Commutation Law

In analogy with the bias current actuator, the electromagnets are switched according the sign of the control force F_{CAx}, cf. **Fig. 6**.

The combination of the feedback linearization with the control force signed based switching result in the following relation for the actuator currents i_1 and i_3:

$$\begin{cases} i_1 = \dfrac{1}{\sqrt{\eta_1(x_A)}}\sqrt{F_{CAx}}\,, & F_{CAx} \geq 0 \\ i_3 = \dfrac{1}{\sqrt{\eta_3(x_A)}}\sqrt{-F_{CAx}}\,, & F_{CAx} < 0 \end{cases} \tag{5}$$

where

$$\eta_1(x_A) = \frac{k_A}{(x_{0A} - x_A)^2}$$
$$\eta_3(x_A) = \frac{k_A}{(x_{0A} + x_A)^2} \tag{6}$$

Equation (1) has been used for the derivation.

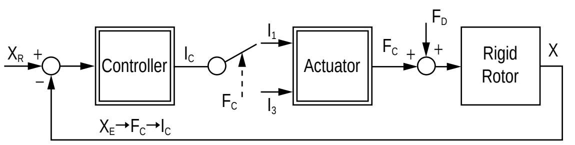

Fig. 6 Block diagram of zero bias current active magnetic bearing

7. Performance Comparison by Simulation

Different approaches have been proposed to reduce the bearing losses. Here, a particular case is discussed more in detail, i.e. zero-bias current active magnetic bearings, and compared with bias current active magnetic bearings. The simplest case is treated: two bearings support a rigid rotor, cf. **Fig. 2**.

The equation of motion of the rigid rotor for the coordinate x_A is

$$\ddot{x}_A = \underbrace{\left[\frac{r_A^2}{j_r} + \frac{1}{m}\right]}_{\sigma_A} F_{Ax} \tag{7}$$

where m is the mass, j_r is the radial mass moment of inertia, and r_A is the distance between the center of mass and the bearing plane A. The coupling between the bearing planes A and B, and between the coordinates x and y, are neglected.

In the bias current case, substituting equation (4) into the equation of motion (7), and solving for the control force $F_{CAx} = k_{ixA}\, i_{CxA}$ results in the linear plant model

$$G(s) = \frac{X_A(s)}{F_{CAx}(s)} = \frac{\sigma_A}{s^2 - \sigma_A k_{pxA}} \tag{8}$$

In the zero-bias current case, the assumed exact feedback linearization combined with the force sign based switching results as well in a linear plant model. Substituting equation (5) into equation (1) and (7), and solving for the control force F_{CAx} results in

$$H(s) = \frac{X_A(s)}{F_{CAx}(s)} = \frac{\sigma_A}{s^2} \tag{9}$$

7. 1 Control Model Comparison

Two control models are given for the rotor-bearing system, one for bias current active magnetic bearings by equation (8), and one for zero-bias current active magnetic bearings by equation (9). They are both of order two, but with a different pole placement on the real axis. The Jacobian linearized system $G(s)$ is unstable, where the feedback linearized system $H(s)$ is marginally stable.

Furthermore, with increasing frequency the systems have the same dynamic as can been seen in the Bode diagrams depicted in **Fig. 7**. Therefore, for a sufficient big control gain a similar controller dynamic can be chosen for both cases.

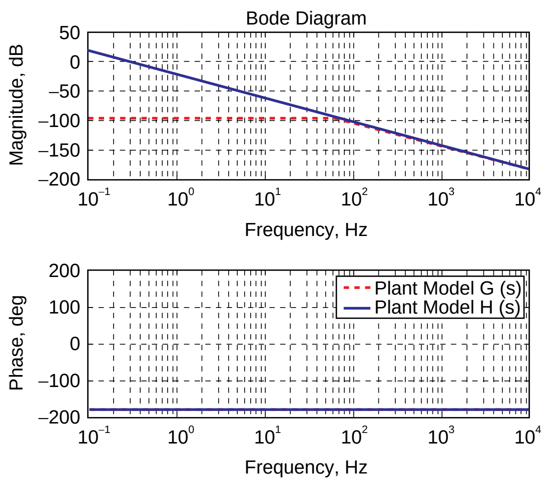

Fig. 7 Bode diagram of plant models G(s) and H(s)

The used parameters are the coupling factor σ_A = 0.0885kg^{-1} and the position stiffness k_{pxA} = 6.03 104 N/m.

7. 2 Regulation

At least a PD controller is necessary to stabilize the plant.

$$C(s) = \frac{F_{CAx}(s)}{X_A(s)} = K_{pxA}(1 + T_{dxA}\, s) \tag{10}$$

The proportional gain is denoted by K_{pxA}, and the derivative time constant by T_{dxA}.

The control goal is regulation of the rotor position. However, for the given PD controller, static disturbances cannot be corrected and a steady-state error remains. Because the steady-state behavior is compared in the following, no integral action is introduced.

Then the steady-state error for a step disturbance force F_{DAx}, cf. **Fig. 9**, is in the bias current case

$$\lim_{t\to\infty} x_A(t) = \frac{1}{K_{pxA} - k_{pxA}} F_{DAx}(0) \tag{11}$$

and is in the zero-bias current case

$$\lim_{t\to\infty} x_A(t) = \frac{1}{K_{pxA}} F_{DAx}(0) \tag{12}$$

Increasing the proportional gain of the bias current case controller by the position stiffness, $K^{*}_{pxA} = K_{pxA} + k_{pxA}$, results in the same steady-state performance as for the zero-bias current case. This is illustrated in **Fig. 8**, where the steady-state error is depicted in function of the disturbance force step level. The difference at high disturbance levels is, because the Jacobian linearization is only valid for small disturbances.

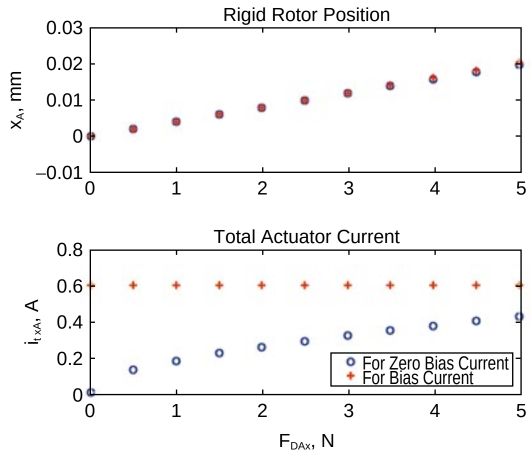

Fig. 8 Position steady-state error and total actuator current

The used parameters for the plant models are the same as in **Fig. 7**. The controller parameters are the proportional gain K_{pxA} = 2.5 105 N/m and K^{*}_{pxA} = 3.103 105 N/m, the derivative time constant T_{dxA} = 0.002s, and the sampling time h = 0.2ms. The bias current is i_{0xA} = 0.3A.

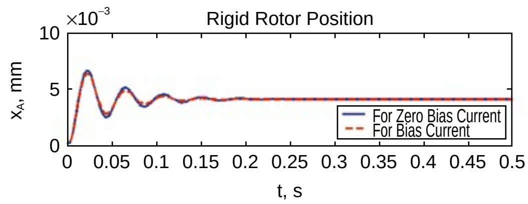

Fig. 9 Disturbance force step response

The disturbance step level is 1N. The used parameters for the plant models and the controller are the same as in **Fig. 8**.

7. 3 Total Control Current

The total actuator current for the bias current case is equal two times the bias current. In the zero-bias current case, the total current depends on the disturbance level, as can be seen in **Fig. 8**. Depicted is the sum of the steady-state control currents corresponding to the steady-state error.

8. Conclusion

The paper is concluded as follows,

(i) the bearing loss of zero-bias active magnetic bearings depends strongly on the nature of the disturbances,

(ii) for zero-bias current active magnetic bearings feedback linearization is possible that together with control force sign based switching leads to a linear system.

(iii) This system can be regulated using the same controller as for its bias current counterpart.

(iv) Furthermore, correcting the proportional gain by the position stiffness yields the same steady-state performance.

However, the investigations have only been done by simulation. It is therefore necessary to prove by experiment the feasibility of the approach, and, if necessary, to consider the bending and gyroscopic characteristic of the rotor and the inhomogeneity of the bearings due to switching.

Acknowledgement

We would like to thank Prof. Kenzo Nonami of the University of Chiba, Japan, and Dr. Selim Sivrioglu of the Gebze Institute of Technology, Turkey, for the many discussions we had, which greatly helped us to understand the problems of zero-bias current active magnetic bearings.

References

1) Thoolen F. J. M., Development of an Advanced High Speed Flywheel Energy Storage System, Ph. D. Thesis Eindhoven University of Technology, The Netherlands, 1993.
2) Kondoh J. et al., Electrical Energy Storage Systems for Energy Networks, Energy Conversion and Management, **41**, 17 (2000) 1863-1874.
3) Kasarda M. E. F., An Overview of Active Magnetic Bearing Technology and Applications, The Shock and Vibration Digest, **32**, 2 (March 2000) 91-99.
4) Hu T. et al., Stabilization and Performance of a Balance Beam Suspended on Magnetic Bearings, Proceedings of the 2002 American Control Conference, ACC 2002, Anchorage, Alaska, USA, **6** (May 8-10, 2002) 4662-4667.
5) Knospe C. and Yang C., Gain-Scheduled Control of a Magnetic Bearing with Low Bias Flux, Proceedings of the 36th IEEE Conference on Decision and Control, CDC 1997, San Diego, California, USA, **1** (December 10-12, 1997) 418-423.
6) Charara A., De Miras J., and Caron B., Nonlinear Control of a Magnetic Levitation System Without Premagnetization, IEEE Transactions on Control Systems Technology, **4**, 5 (September 1996) 513-523.


S. GÄCHTER* H. KAMENO*

* *Mechatronic Systems Research & Development Department, Research & Development Center*