An Energy Storage Flywheel Supported by Hybrid Bearings

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***Abstract*—Energy storage flywheels are important for energy recycling applications such as cranes, subway trains. In a petroleum field, a drilling platform runs with big load variation. A vertical flywheel energy storage system had been tested to stabilize the load fluctuation and proved its effectiveness. To improve bearing life and reliability, a new flywheel bearing system was designed. The key was the use of hybrid bearings including an axial permanent magnetic bearing (PMB), a lower end ball bearing and an upper end active magnetic bearing (AMB). The design effectively combined advantages of the different bearings. The original upper ball bearing was replaced with the AMB to avoid the use of a lubrication system for it and improve the damping ability for the rotor vibration modes. The vibration mode and the harmonic response were provided in the rotor dynamic analysis for the new flywheel rotor. The analysis results proved that the new rotor structure and the AMB design were appropriate.**

I. Introduction

In an oil drilling platform, the drill is a key instrument and its power is often provided by a diesel power system or a natural gas engine. When a drill operates, its power consume varies a lot with time and ongoing operation. Using diesel or natural engines to operate the drill would pose problems of low efficiency and large amount of emissions. It is attractive to replace such drilling energy sources with a new one which can make the entire drilling process economic and environmentally friendly [1], [2].

One of the advanced technologies is adding a flywheel subsystem to an original diesel engine system. When a drill operates in a low load state, the diesel engine could still operate in a full power mode and the redundant power from the diesel could be used to drive the flywheel rotor through a DC motor. When the rig operates in a high load station, energy shortage for a peak power requirement could be supplied by the flywheel system. The flywheel energy storage system would discharge and supply power to the rig through the DC motor.

A flywheel energy storage system (FESS) is one of options among available renewable energy resources. It has a high output power, a long life and a high response speed [3]. It is economically and practically feasible to modern oil and gas industry.

A large power flywheel (Flywheel I) had been successfully designed and tested in a drill system in Zhongyuan Petroleum Exploration Bureau of China, whose driving power was based on a diesel engine system. Its effects had been proved [1]. In the test rig system, the output time of the power system for a pulse drilling load of 300-500 kW was 2-20 s. During one operation period, the diesel engine had an abundant output time of 80-120 s at 100-150 kW power level. So in its operation mode, the flywheel worked as a motor at 90-130 kW power level for 120 s and then worked as a generator at 300-500 kW power level for 20 s. The flywheel operated as a power amplifier in a mode with lower charge power and higher discharge power.

The structure of the Flywheel I rotor was shown in Fig. 1. It was a vertical flywheel supported by an upper ball bearing, a lower ball bearing and a permanent magnetic bearing (PMB). The flywheel housing was filled with helium to decrease wind loss of the rotor in operation. The ball bearings were used to support radial load and partial axial load of the flywheel rotor whose maximum rotation speed was 3600 rpm. For each of the bearings, there was a plastic ring mounted outside to decrease its radial stiffness and provide necessary damping for the rotor bearing system. The PMB was used to counteract the most rotor weight. It effectively decreased the axial load of the lower ball bearing and was benefit for the bearing life. The lower bearing was immersed in oil and its life could be ensured by a good heat transfer and a good lubrication. For the upper bearing, it was lubricated by grease. The lubrication was not as good as the lower bearing and its life would be influenced.

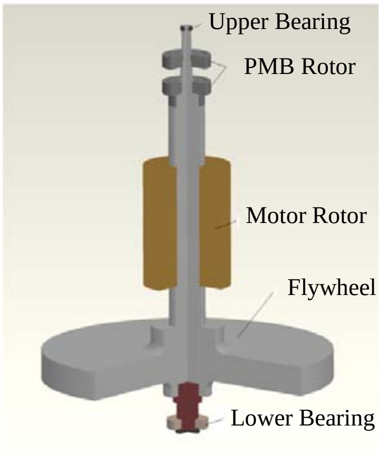

Figure 1. The structure of the Flywheel I rotor.

Active magnetic bearings are attractive for their special advantages compared with traditional ball bearings and fluid film bearings, such as contactless, no wear, no oil, low power consumption, low maintenance cost, and controllability of bearing dynamic characteristics and of rotor unbalance response [4].

To further improve bearing life, especially for the upper bearing, a new hybrid bearing strategy was designed and it would be tested in a new flywheel system (Flywheel II). The hybrid bearings included an axial permanent magnetic bearing (PMB), a lower end ball bearing and an upper end active magnetic bearing (AMB). The key was to replace the original upper ball bearing with the new AMB.

Such a design combined advantages of the different bearings and could improve system reliability. The PMB was cheap and energy-saving. It removed most static load for the vertical rotor. So it was beneficial for improving the ball bearing life and decreasing the system power lost. The ball bearing was cheap and reliable but needed lubrication. Its static load was limited at high rotation speeds and was suitable to be used as the lower end bearing. For the upper end support, a lubrication system for a ball bearing was complex and difficult to maintain. The lubrication problem of the upper bearing could be avoided by using the AMB. The AMB with no need for lubrication was dynamic controllable and suitable for the high speed rotor with a small dynamic load. Though cost more, it could provide a strong ability of restraining the rotor vibration, effectively improve the bearing life and reduce the maintenance cost.

The new flywheel system structure with the new hybrid bearing system would be shown and the important feathers of the hybrid bearing system would be provided.

To prove the effectiveness of the AMB design, the rotor dynamic analysis for the new flywheel rotor would be done. The observability and controllability of the AMB for the vibration modes interested would be discussed, and the load capacity of the AMB would be compared with the harmonic response analysis results of the rotor system.

A. *Flywheel I rotor system*

The Flywheel I rotor was designed according to the FESS power and energy requirement. The overall energy efficiency of the system in one working period should be larger than 80%. Some important system parameters were listed in Table 1.

TABLE I. SYSTEM PARAMETERS OF THE FLYWHEEL I

Parameter	Value
Recycle speed range	1800-3600 rpm
Total kinetic energy at the highest rotation speed	16.3 MJ
Motor power	300-500 kW
Transverse moment of inertia	230 kg.m^2
Highest line speed of the rotor	226 m/s
Overall high of the rotor	1800 mm
Largest diameter of the rotor	1200 mm
Total weight of the rotor	1600 kgf

The design of the Flywheel I had been proven successful by field experiments. But for a long-term operation, the current lubrication of the upper bearing was not satisfied. It was hard to build a new lubrication system for the upper bearing to increase its reliability. So a new hybrid bearing strategy was designed and would be tested in the Flywheel II rotor system.

B. *Flywheel II rotor system*

The new flywheel system structure with the new hybrid bearing system was shown as Fig. 2.

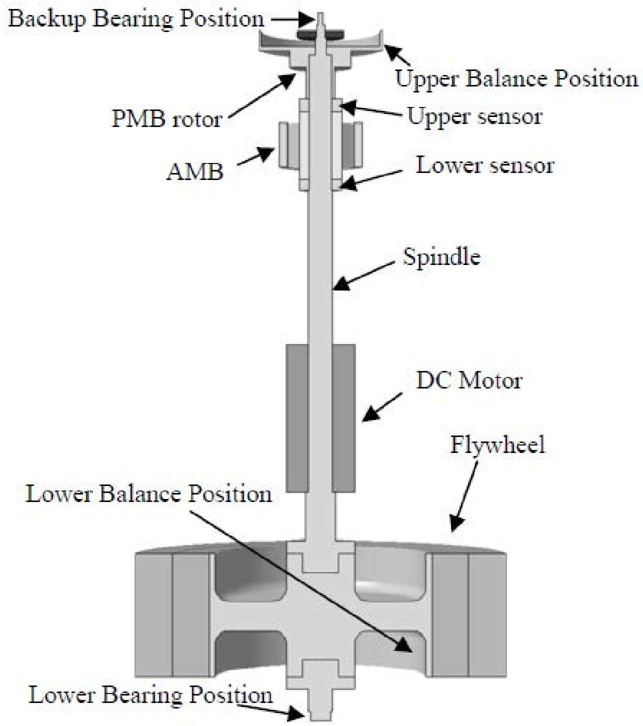

Figure 2. The structure of the Flywheel II rotor.

In the new design, an upper active magnetic bearing was added to the original bearing systems and it was used to replace the upper ball bearing to support the rotor at the upper end. A backup ball bearing was located in the position where the original ball bearing located. But there was a small gap of 0.25 mm between its inner ring and the flywheel rotor. Such a backup bearing was used to protect the AMB rotor from colliding with the AMB stator when the rotor was in a critical condition where the rotor vibrated violently, or when the AMB stopped working.

For the AMB, there were two steel rings fixed on the upper and the lower sides of the AMB rotor. They were both used as detection targets of eddy current sensors. The displacement signal of the AMB center was calculated from the signals of the two sensors.

The flywheel was designed to store a total energy of 2.47 kWh at 18,000 rpm and deliver 40 kW for more than 160 seconds. Some important system parameters were listed in Table II.

The flywheel fitted on a steel hub was composed by two kinds of composite materials. The steel hub was first wrapped with glass fiber composites and then carbon fiber composites were wrapped outside the inner glass fiber.

TABLE II. SYSTEM PARAMETERS OF THE FLYWHEEL II

Parameter	Value
Recycle speed range	9000-18000 rpm
Total kinetic energy at the highest rotation speed	8.9 MJ
Motor power	20-40 kW
Transverse moment of inertia	5 kg.m^2
Highest line speed of the rotor	565 m/s
Overall high of the rotor	1150 mm
Largest diameter of the rotor	600 mm
Total weight of the rotor	170 kgf

C. Active magnetic bearing

The AMB was placed above the motor and it was used to replace the original upper radial ball bearing. When it worked, it would levitate the rotor from the upper backup bearing (a ball bearing), provide stiffness and damping for the rotor and help the rotor running stably across several critical speeds in a high speed operation. Because the distance from the AMB center to the mass center was much larger compared with the distance from the lower bearing and the rotor was vertically oriented, the static load for the AMB was not large. But its dynamic load capacity should be large enough to restrain the rotor's synchronous critical vibration when the rotor crossed its critical speeds.

The basic parameters of the AMB were shown in Table III.

TABLE III. AMB PARAMETERS OF THE FLYWHEEL II

Parameter	Value
Stator length	70 mm
Bearing load	1000 N
Diameter of the bearing rotor	70 mm
Gap between the bearing rotor and the stator	0.5 mm
Stator pole number	8

D. Rotor vibration modes

For the Flywheel II, an important consideration in the rotor design was to assure the effectiveness of the AMB in the rotor vibration control. The key was that the observability and controllability of the AMB should be suitable. So an accurate rotor vibration mode analysis should be carried out.

To obtain accurate dynamic behavior of a rotor, finite element methods are usually used. Many commercial software packages, such as NX Nastran, have provided the ability to construct a model for a rotor with complex meridian cross sections. In such a model, gyroscopic effects and centrifugal softening effects can be effectively taken into account.

For simplicity, 2D Fourier elements were used to model the flywheel rotor. With the 2D Fourier elements, axisymmetrical rotors can be modeled using 2D axisymmetrical shell or volume finite elements whose displacement field is developed in Fourier series [5]. The corresponding model was shown in Fig. 3. There were two springs connecting the ground with the rotor at the positions corresponding to the AMB and the lower bearing. The spring stiffness was set as 1000 N/mm and 10000 N/mm for the two bearings respectively. The damping was set the same as 500 N.s/m for them. The first 5 vibration modes of the flywheel rotor at standstill were shown as Fig. 4. The modes included two rigid modes and 3 flexible modes. Because the eigenfrequencies of the other modes were much higher than the maximum rotation frequency of the rotor, the analysis for them was ignored here.

Figure 3. 2D FEM model of the flywheel rotor.

Figure 4. Vibration modes of the flywheel rotor.

When the rotor runs from 0 to 18000 rpm, the vibration of the 5 modes must be well restrained. So their observability and controllability condition at the AMB position should be assured. The FEM calculated results for them were listed in Table IV. The values in the table showed the vibration amplitude of the rotor relative to the maximum amplitude values of the corresponding vibration modes. For the AMB, the mode vibration amplitude of the 5 modes was acceptable and it assured that the AMB force could effectively control the mode vibration. But at the position of the upper sensor, the vibration amplitude of the 4^{th} mode was relatively small, namely, that the observability of the sensor for the 4^{th} mode was not good. A similar case happened at the lower sensor position for the 3^{rd} mode. If the two sensors were combined to provide an average value, the AMB center displacement measurement accuracy for all the modes could be acceptable.

TABLE IV. OBSERVABILITY AND CONTROLLABILITY CONDITION OF THE FIRST 5 MODES

Position	Mode number				
	1st	2nd	3rd	4th	5th
Upper sensor center	-0.86	0.68	0.50	-0.24	-0.50
AMB center	-0.90	0.55	0.29	-0.50	-0.82
Lower sensor center	-0.94	0.43	0.12	-0.80	-0.88

E. *Gyroscopic effects*

Because of the flywheel body on the rotor, its inertia was high and the gyroscopic effects were strong. The gyroscopic effects greatly influenced the rotor dynamic behavior of the rotor at a high rotation speed. The eigenfrequencies at different rotation speeds were obtained from the FEM analysis and the Campbell diagram for the rotor was shown in Fig. 5.

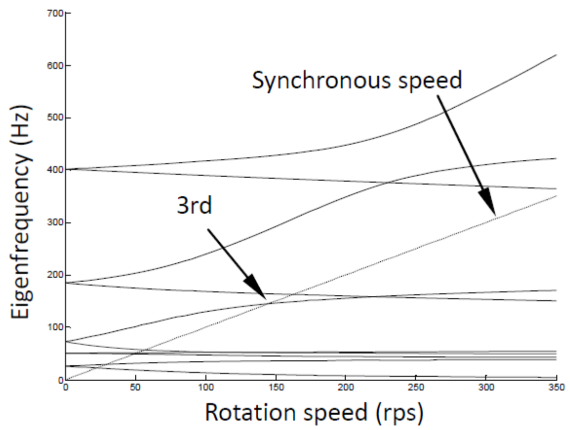

Figure 5. Campbell diagram for the rotor.

From Fig. 5., it could be seen that the rotor's rigid body modes and bending modes were both influenced by the gyroscopic effects obviously. For a full speed operation, the rotor had to pass 3 critical speeds totally. Because the 3rd critical speeds was much higher than the other critical speeds, as shown in the figure, it was the most difficult to be passed. The AMB should have a large enough dynamic load capacity to provide a force to counteract the excitation force from the rotor unbalance at this critical speed. Furthermore, the rotor should be properly balanced before a high speed operation to decrease the load of the bearings.

F. *Frequency response analysis*

To verify that the AMB could effectively restrain the synchronous vibration of the rotor at the 3rd critical speeds, a frequency response analysis for the rotor was necessary.

After a good dynamic balance, if the balance quality grade of the rotor could reach G6.3, the residual unbalance of the rotor would be 536 g.mm. When the residual unbalance was placed at the rotor mass centre and the rotor was suspended by the two bearings as shown in Fig. 3, the harmonic response of the rotor at the upper and the lower bearings was shown in Fig. 6. It could be seen that the reaction force of the upper bearing was much less than its load capacity within the rotation speed range when a damping of 500 N.s/m was provided (such a damping was reasonable for a flywheel AMB [6]). So for an operation above the 1^{st} flexible critical speed, the AMB load capacity would be sufficient.

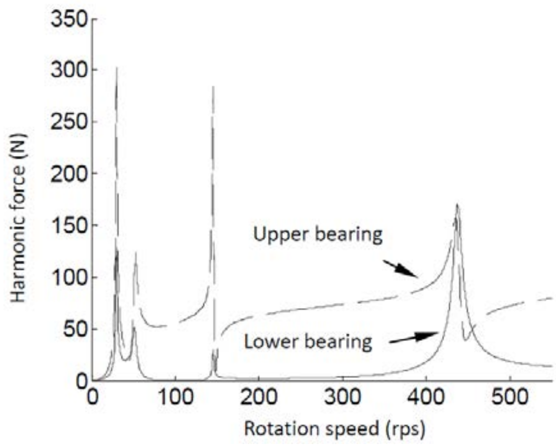

Figure 6. Harmonic response of the rotor.

G. *Conclusion*

The new FESS was introduced, especially its new hybrid bearings. The new bearings including the axial PMB, the lower end ball bearing and the upper end AMB could combine advantages of the different bearings to improve the bearing life and the system reliability.

With the rotor dynamic analysis for the new flywheel rotor, it was seen that the rotor had to run across its first flexible critical speed to reach its maximum rotation speed. The observability and controllability of the AMB for the vibration modes interested were suitable. The harmonic response analysis results showed that the bearing load was enough for such a supercritical operation.

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