APPLICATION OF GIANT MAGNETOSTRICTIVE MATERIAL TO DISC BRAKE ACTUATOR

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ABSTRACT – For the next generation railway brake system, a disc brake which can be operated directly and electrically is strongly expected. This paper deals with newly developed disc brake actuator using giant magnetostrictive materials (GMM) which can be integrated with disc brake. Regarding the brake system performance, a better delay time was also attained which will contribute to shorten a stopping distance.

1. INTRODUCTION

Recently the requirements for railway brake system are increasing. If a brake itself can be operated and controlled directly and electrically, a very simple and multifunctional brake system can be realized.
This new system can provide any brake function not by equipping additional hardware but by adding only a software. This is the reason we are developing an electrically operated brake actuator which can be integrated with a disc brake. Giant magnetostrictive materials (GMM) have comparatively a large strain and a high output thrust as well as a short response time. They are expected to be applied to various actuators [1], [2], [3]. We have developed an electrically operated brake actuator for a railway disc brake using GMM which functions as a hydraulic pump actuator.

2. CONVENTIONAL AND GMM DISC BRAKE SYSTEM

2.1 Conventional Disc Brake System

Figure 1 shows the structure of the conventional railway disc brake system. A signal from the operating lever is transferred to the ECU (Electronic Control Unit) which activates the solenoid valves according to the input signal.
The air from the air tank is modulated and supplied to the air/hydraulic converter.
Air pressure is then converted to hydraulic pressure and sent to the brake cylinder through a hydraulic tube and acts on the brake piston.
The two brake pads are pushed on each side of the brake rotor which rotates together with the wheel of the train.

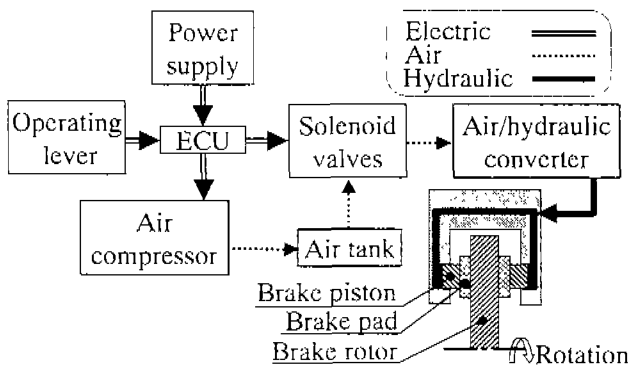

Fig. 1 Block diagram of Conventional brake system.

The braking torque to decelerate the train is calculated by the equation (1) in case of twin piston disc brake.

$$T = \pi D_B^{\,2} P R \mu \tag{1}$$

where,
T : braking torque per brake [Nm]
D_B : diameter of brake piston [m]
P : hydraulic pressure [Pa]
R : effective radius of brake rotor [m]
μ : friction coefficient between brake pad and brake rotor

As seen on Figure 1, the conventional brake system is very complex system requiring many devices.

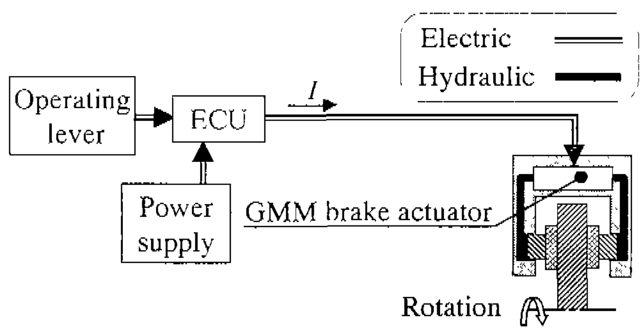

Fig. 2 Block diagram of GMM brake system.

2.2 GMM disc brake system

The concept of the GMM disc brake system is shown in Figure 2.

The system is greatly simplified as the GMM brake actuator is integrated with the disc brake and there are no devices related to the air pressure.

2.3 GMM brake actuator

Figure 3 shows the configuration of the GMM brake actuator. It consists of a GMM pump, a reservoir and a solenoid valve which is provided for releasing the brake.

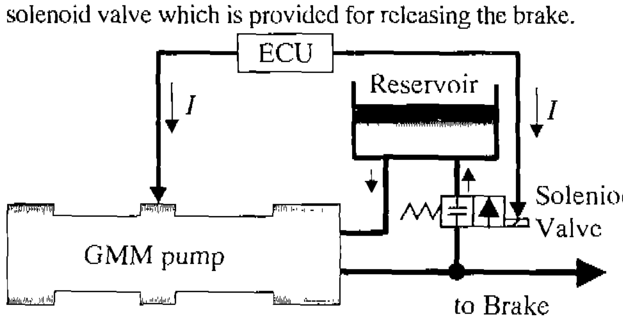

Fig. 3 GMM brake actuator.

3. CHARACTERISTICS OF GMM BRAKE ACTUATOR

3.1 GMM pump

The structure of the prototype GMM pump is shown in Figure 4. The GMM expands in the magnetic field generated by the coil and pushes the piston. The piston moves right in Figure 4 and discharge the fluid. The stroke of the piston is equal to the amount of expansion of the GMM. When the magnetic field is released the GMM shrinks and the piston returns by the spring force. The fluid is then sucked from the reservoir.

This pumping motion occurs cyclically by applying an alternating current to the coil. As the GMM can expand in both positive and negative magnetic field, the frequency of the piston oscillation is twice as much as the input current of the coil.

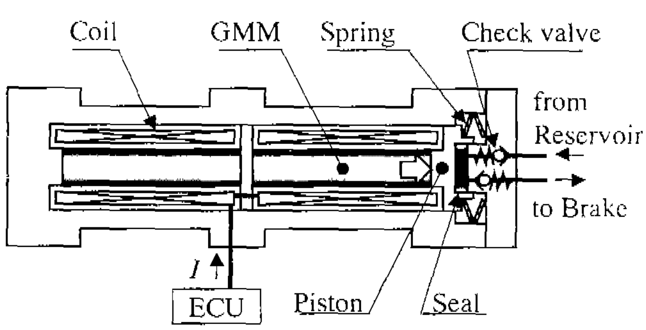

Fig. 4 Structure of GMM pump.

3.2 Calculation of GMM pump performance

The flow rate, Q of the GMM pump is given by the equation (2).

$$Q = \frac{\pi}{2} D^2 x f \eta \tag{2}$$

where,

Q : flow rate [L/s],
D : diameter of piston [m],
x : stroke of piston [m],
f : frequency of current [Hz],
η : mechanical efficiency.

The target flow rate at unloaded condition was set to be Q = 24 mL/s to conform the target delay time of the braking.

The total length of the GMM was decided as $2L_g$ = 200 mm from the packaging space.

The piston stroke therefore was estimated x = 0.28 mm at unloaded condition as the maximum magnetic strain of the GMM was expected to be 1,400 ppm.

Table 1 shows the design specification of GMM pump.

Table 1 Specification of GMM pump.

GMM	Diameter	D_g	[mm]	20
	Length / piece	L_g	[mm]	100
	Number of material			2
Coil	Diameter of wire	d	[mm]	1.3
	Number of turns	N		720
	Length	L	[mm]	200
	Resistance	R	[Ω]	0.88
Piston	Diameter	D	[mm]	24
	Stroke	x	[mm]	0.28

Then, the hydraulic pressure, P is expressed by the equation (3).

$$P = E_g \varepsilon_m \left(\frac{D_g}{D} \right)^2 \tag{3}$$

where,

P : hydraulic pressure [Pa],
E_g : Young's modulus of GMM [Pa],
ε_m : mechanical strain of GMM [ppm],
D_g : diameter of GMM [m].

From the equation (3), the mechanical strain, is expressed as follows:

$$\varepsilon_m = \frac{P}{E_g} \left(\frac{D}{D_g} \right)^2 \tag{4}$$

Considering that the required hydraulic pressure is P = 8.8 MPa and the Young's modulus of GMM is E_g = 30 GPa. The diameters of the piston and the GMM were designed as D = 20 mm and D_g = 24 mm respectively. In this condition the mechanical strain was calculated as ε_m = 422 ppm.

3.3 Drive circuit for GMM pump

To obtain the magnetic strain of 1,400 ppm, the coil should produce a magnetic field strength, H = 180 kA/m. From the coil specification it was estimated for the drive circuit that the current, I should be ±50 A and the frequency, f should be up to 100 Hz.

The drive circuit was designed according to above calculation and estimation. Figure 5 shows the drive circuit.

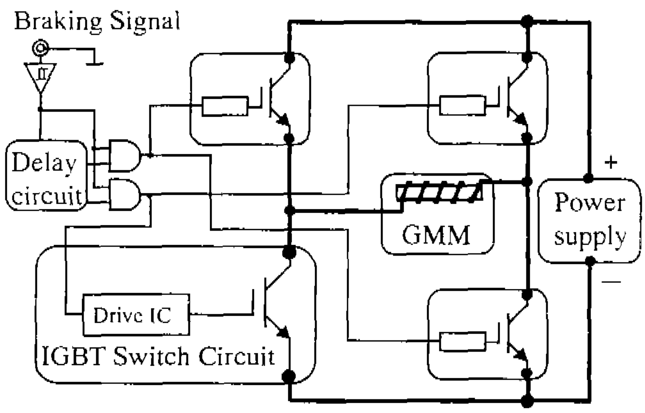

Fig. 5 Drive circuit for GMM pump.

3.4 Measurement of current and piston stroke

Figure 6 shows the block diagram to measure the current, I and the piston stroke, x vs. time, t.

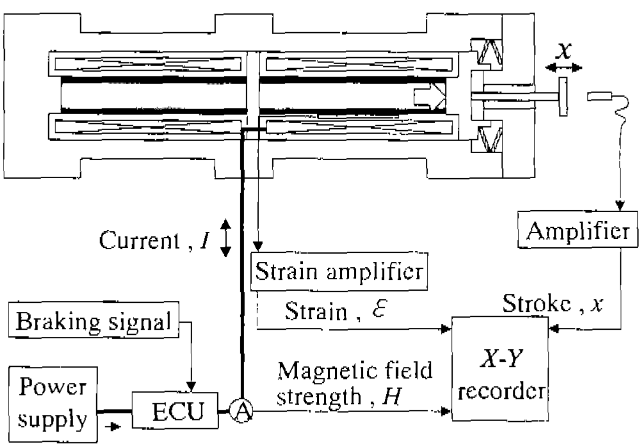

Fig. 6 Measuring block diagram for current and stroke.

Figure 7 shows the result. Both the current and the piston stroke satisfy the design specification.

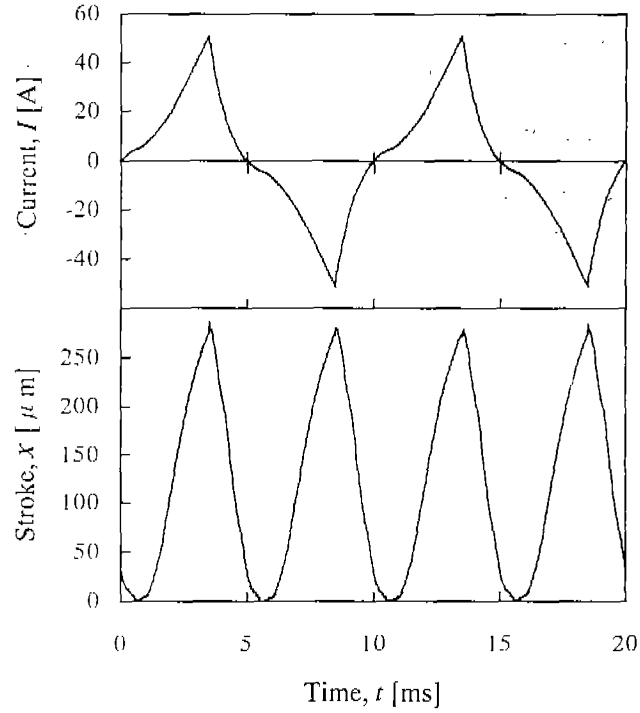

Fig. 7 I and x - t characteristics of GMM.

3.5 Pump performance

Figure 8 shows the block diagram to measure the flow rate of the pump.

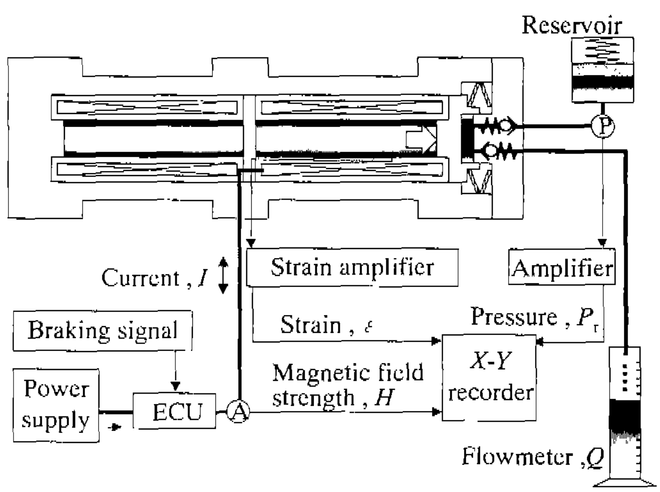

Fig. 8 Measuring block diagram for flow rate.

Figures 9 shows the result, the flow rate, Q vs. the frequency of the current , f at unloaded condition. The dotted line is calculated performance on the assumption that the mechanical efficiency η = 1. Target flow rate, Q = 24 mL/s is attained around the frequency of the current , f = 100 Hz.

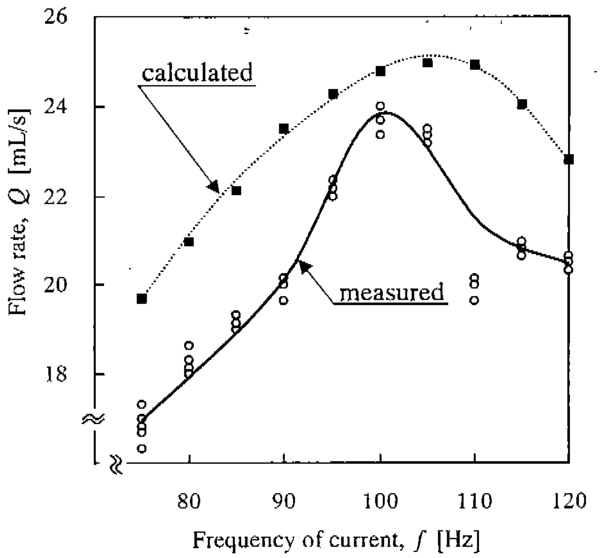

Fig. 9 Q - f characteristics of GMM.

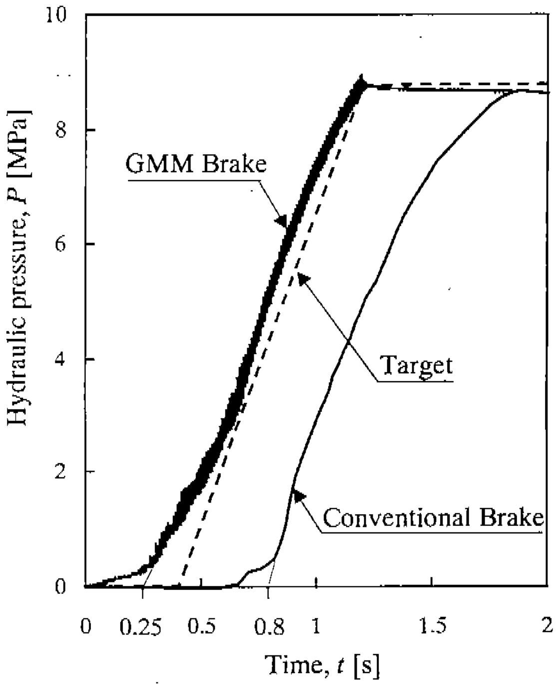

Fig. 11 $P-t$ characteristics.

4. PERFORMANCE AS GMM DISC BRAKE SYSTEM

Finally a test was conducted by connecting the prototype GMM pump with a conventional disc brake for evaluation. Figure 10 shows the testing block diagram. The hydraulic pressure, P vs. time, t were measured. The result is shown in Figure 11. The curves of the target and the conventional system are also expressed. The prototype GMM brake actuator could satisfy overall performance. Specifically, the delay time was greatly improved from 800 ms to 250 ms. This will result in shorter stopping distance.

On the other hand, the targeted buildup rate of the hydraulic pressure stays almost the same level as that of the conventional system. This is because the excessive buildup rate is not good for the comfort of the passengers.

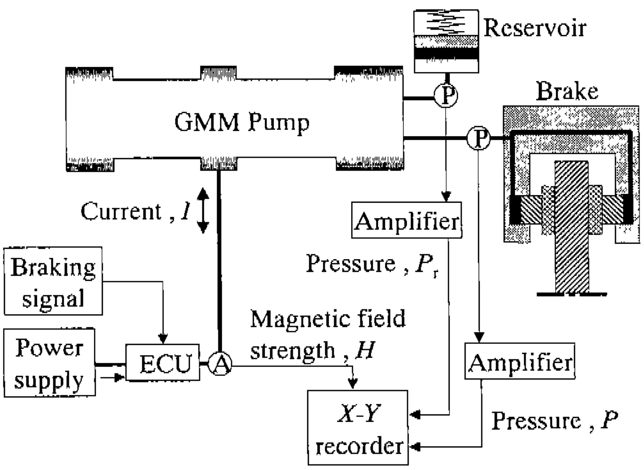

Fig. 10 Testing block diagram for evaluation.

5. CONCLUSION

The following were found through designing and evaluating the prototype GMM brake actuator:

(1) An electrically operated brake actuator has been realized in a simple structure by using the GMM as a hydraulic pump actuator which can be fitted integrally with a disc brake.

(2) Delay time of the prototype GMM disc brake system was 250 ms which is much better than 800 ms of the conventional system. This will contribute to shorten a stopping distance.

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