Research of the Bus Secondary Hydraulic Transmission System under Variable Pressure Rail

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ABSTRACT

Hydrostatic transmission with secondary regulation can regenerate inertial and gravitational energy of load. Based on deeply analyzing the working principles and energy-saving theory of the secondary regulating transmission system under variable pressure rail, regenerating the transmission system's inertial energy by controlling power was put forward. Considering large changes of the parameters of the transmission system and its non-linearity, a fuzzy-neural network control was adopted, and the mathematical model of the system was established, then the simulations of the performance of the transmission system has been conducted. The conclusion was made that the inertial energy can be reclaimed and reused, and the self adaptive ability and controlling performance of the secondary hydraulic transmission system is improved.

Keywords**:** *Secondary Hydraulic Transmission, Fuzzy-Neural Network Control, Constant-Power Control, Performance Research.*

1. INTRODUCTION

Secondary regulating static-liquid transmission technique is regulating for the secondary element that inter-converts hydraulic energy and mechanical energy. By regulating displacement of the secondary element, rotary speed or torque of load can be regulated. By changing oil-flowing direction of the secondary element (passing zero point), the secondary element has working states of both "hydraulic motor" and "hydraulic pump" [1]. When working in the motor state, the secondary element outputs power to the load. When working in the pump state, the secondary element recycles braking energy of the system. When used for the transmission system of the locomotive machine which works in periodicity, in particular the bus that continually starts and stops, the secondary regulating static-liquid transmission technique can greatly advance efficiency of the system, save energy and reduce environmental pollution.

2. ENERGY-SAVING MECHANISM of the TRANSMISSION SYSTEM

The bus transmission structure showed in Fig. 1. The secondary regulating transmission system consists of the engine(1), clutch(2,5), gear-box(3), gearing(4), accumulator(6), secondary element(7), rear bridge(8). It drives the bus in the form of that mechanical energy of the engine and the secondary element work together to drive the bus[2]. It includes two power-driving systems. The first system is that the power of the engine is transmitted to the driving system by the clutch, which is the same as that of the bus. Another system is the hydraulic driving system that transmits energy by the hydraulic pump, accumulator, secondary element and transmission shaft. These two systems can be used jointly or respectively. When climbing slope or accelerating, the bus is driven by two driving systems jointly with assistance of the hydraulic system. When running commonly, the bus is driven by the engine directly. When the bus is braked, the secondary element works in hydraulic pump state that recycles braking energy of the bus and save it into the hydraulic accumulator in the form of hydraulic energy. In the process of start and accelerating, the hydraulic energy in the hydraulic accumulator can drive the bus by the secondary element that works in hydraulic motor state.

3. STRUCTURE AND WORKING PRINCIPLE OF DOUBLE-ACTING VANE-TYPE SECONDARY ELEMENT

The double-acting vane-type secondary element mainly consists of the rotor (4), stator (2), vane (3), variable cylinder (10), shell (1) and distributing tray (not showed in the Fig. 2) showed in Fig. 2. The variable cylinder (10) consists of the variable rod (6), variable piston (8) and body (9). The ball-shape head of the variable rod is connected with the groove of the variable piston. Another side of the variable rod is fixed at central line of long radius arc of the stator. Another side of the variable rod is fixed on exterior surface at central line of long radius arc of the stator.

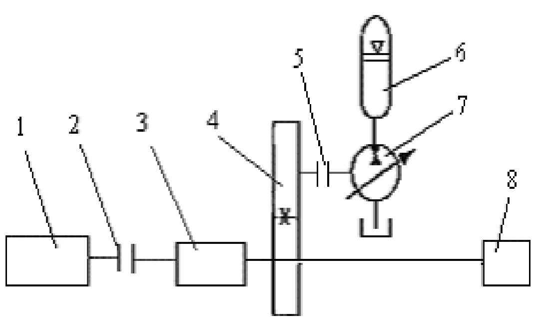

Fig.1 Diagram of the Transmission System

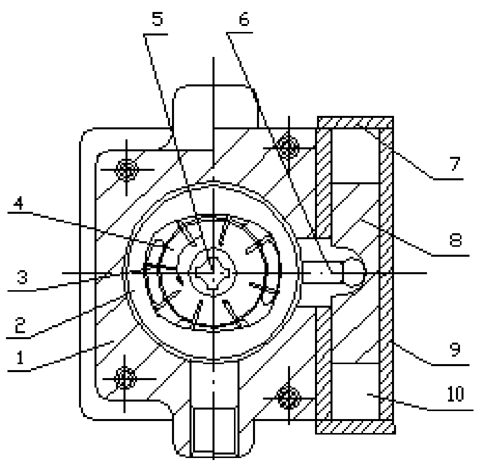

Fig. 2 Structural Diagram of the Element

The oil discharge port and oil entry port of the secondary element are the same in size. Vanes are placed radially on the rotor (placing angle is zero). The stator may rotate around its center by driving of the variable piston. When the secondary element does not work, the central line of the long radius arc of the stator superposes central lines of a group of oil-distributing windows. The rotor is in initial position (zero point) and the displacement of the secondary element is zero. When the stator rotates clockwise or counter-clockwise, the rotary angle increases and the displacement of the secondary element also increases. When the rotary angle reaches to 45º, the placement of the secondary element is maximal. When the rotary angle is negative 45º, the negative displacement of the secondary element is maximal. As the stator rotates within the range between negative 45º and positive 45º, the displacement of the secondary element changes continuously. Position relation between the distributing trays and the stator can be regulated by the variable cylinder rotating the rotor of the secondary element. Accordingly, sizes and positions of windows for oil discharge and entry are changed to achieve changes of flux and oil flowing direction. In working, the position sensor checks the rotary angle of the stator and feeds back it to the controller. The controller gives instructions to the electro-hydraulic valve to regulate the displacement of the piston.

3. MATHEMATIC MODELS OF THE SYSTEM

According to structural form of the secondary regulating transmission system for the bus, we get its closed loop pane diagram of system basing on mathematical models of the electro-hydraulic servo valve and all parts of the secondary regulating transmission system.

3.1 Dynamic Mathematic Model for the Electro-Hydraulic Servo Valve

In this system, the natural frequency of the electro-hydraulic valve is high that is much higher than that of the variable cylinder of the secondary element. So, the system can be looked as a proportional process[3]. We have:

$$\frac{Q_{sf}(s)}{I(s)} = K_v \tag{1}$$

where, K_v is flow gain of the servo valve, Q_s is output flow of the servo valve, I is input current of the servo valve.

3.2 Mathematic Models for the Double-Acting Vane-Type Secondary Element

3.2.1 Flux continuity equation for the variable cylinder

$$q = A_g \frac{dx_g}{dt} + C_t p + \frac{V_t}{4\beta_e}\frac{dp}{dt} \tag{2}$$

where, q is flux that enters into the high-pressure cavity of the variable cylinder, A_g is effective area of the piston of the variable cylinder, x_g is the displacement of piston of the variable cylinder, C_t is the total leaking coefficient of the variable cylinder, p is the pressure difference between the high pressure cavity and the low-pressure cavity, V_t is the total volume of two cavities of the variable cylinder, β_e is volume modulus of elasticity.

3.2.2 Force equilibrium equation for the variable cylinder and load

$$A_g p = m_1 \frac{d^2 x_g}{dt^2} + B_1 \frac{dx_g}{dt} + k_1 x_g + F_f \tag{3}$$

where, m_1 is the total mass of load, piston rod and piston components, B_1 is viscous damping coefficient, k_1 is the spring stiffness of the centering spring in the variable cylinder, F_f is the exterior resistance of the piston of the variable cylinder.

3.2.3 Displacement of the secondary element

$$D_2 = \frac{D_{2\max}}{X_{g\max}} x_g \tag{4}$$

where, D_2 is volume, $D_{2\max}$ is the maximum displacement of the element, $X_{g\max}$ is the maximum displacement of the variable cylinder.

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3.2.4 Torque equilibrium equation for the secondary element and load

$$p_s D_2 = J_2 \frac{d^2\theta}{dt^2} + B_2 \frac{d\theta}{dt} + M_L \tag{5}$$

where, p_s is the pressure of the system, J_2 is moment of inertia of rotary parts of the element, θ is the rotary angle of the rotor of the secondary element, B_2 is the viscosity damped coefficient of the secondary element, M_L is the exterior load torque of the secondary element.

3.3 Mathematical Model for the Transmission System

3.3.1 Equilibrium equation for driving force of the bus

If aerial resistance is neglected, the equilibrium equation for driving force is showed below.

$$F_1 + F_2 = m\frac{d^2y}{dt^2} + +mgf\ (\cos\beta + \sin\beta) \tag{6}$$

where, F_1 is the driving force of the engine, F_2 is the driving force of the secondary element, m is the total quality of bus and load, g is the acceleration of gravity, y is the displacement of the bus, f is the rolling friction coefficient, β is the angle of the slope.

3.3.2 Load torque equation for mixing power bus transmission system

$$M_L = \frac{r}{i_1 i_2} F_1 + \frac{r}{i_1 i_3} F_2 \tag{7}$$

Here, i_1 is the transmission ratio of the rear axle of the bus, i_2 is the speed ratio of the gearbox, i_3 is the transmission ratio of the transmission device, r is the radius of the tyre.

3.3.3 Displacement equation for the bus

The displacement equation for the bus is available from relation between the displacement of the bus and the rotary angle of the rotor of the secondary element.

$$y = \frac{r}{i_1 i_3}\theta \tag{8}$$

3.4 Closed Loop Model of Power Feedback for the System

After Laplace transform, simplified process and combined process, the pane diagram of closed loop control for constant power feedback is available[4]. It is showed in Fig. 3.

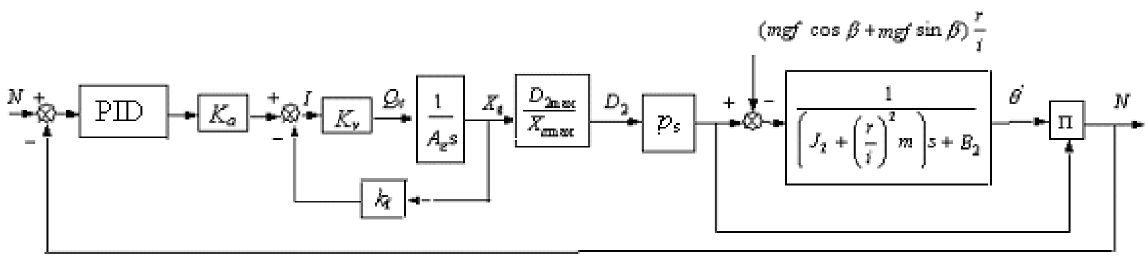

Fig. 3 Closed Loop Pane Diagram of Double Feedback Systems

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4. PERFORMANCE SIMULATIONS

We take one type bus as the simulation object. Main parameters of the double-acting vane-type secondary element are as below: the nominal pressure is 16MPa; the max pressure is 20MPa; the max displacement is 35×10^{-4} m^3/r; the max rotary speed is 1250 r/min; and the max rotary angle of stator is ±45º. Control rule adopts fuzzy-neural network[5] control arithmetic as Fig. 4.

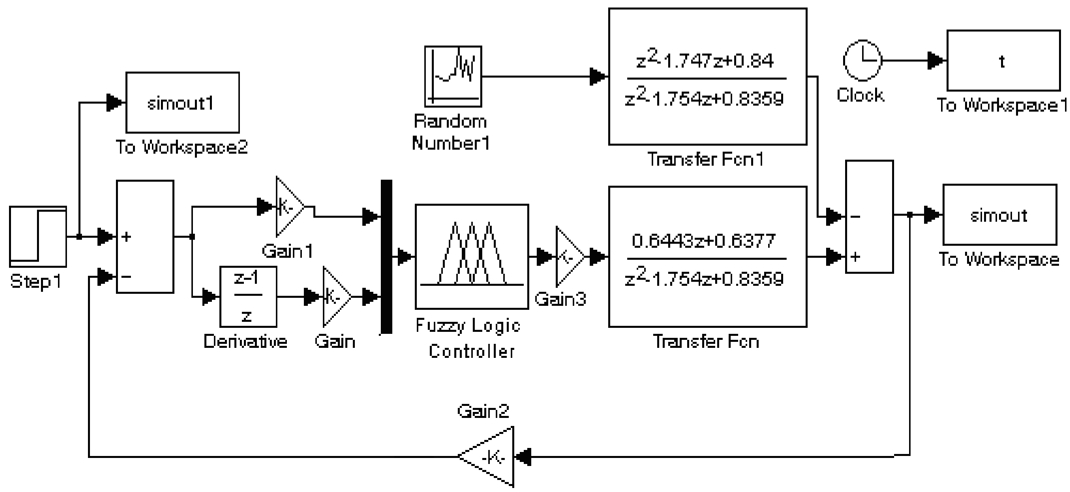

Fig.4 Fuzzy Neural Network Control System Simulation Diagram

We can acquire the simulation curve of velocity, torsion and power of energy-conservation brake of constant power control[6] as Fig. 5, Fig. 6 and Fig. 7 respectively shows. From simulation curves of speed, torque and power, we can find some features of the constant power control energy-saving braking. The double-acting vane-type secondary element can reclaim the braking kinetic energy. Because the constant-power controlled energy-saving braking indirectly controls the braking torque, acceleration in initial braking phase can be controlled by choosing the value of power. In the initial phase and most time of the constant-power braking, the braking torque continuously changes with slow change speed. So, when the constant-power energy-saving braking is used, the braking at high speed is stable and there is not acceleration siltation in most braking time. Torque change is great at evening phase. Particularly, when the chosen braking power is big, the impact braking torque appears. In the constant-power energy-saving braking, the recycle efficiency of energy can be increased by choosing big braking power.

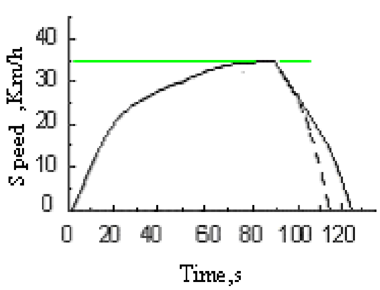

Fig. 5 Curve of Speed

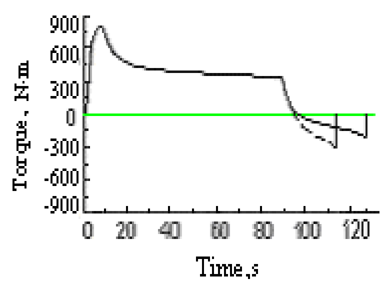

Fig. 6 Curve of Torque

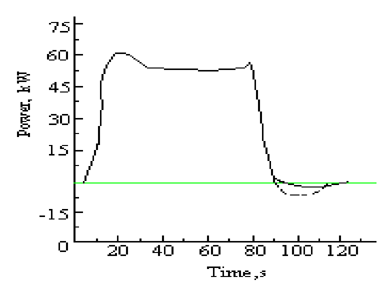

Fig. 7 Curve of Power

5. CONCLUSION

When the braking power is big, the braking torque of the secondary element is big. And, braking time is little and braking distance is short. So, energy consumed by road resistance is little and recycle efficiency of the energy is high. Contrarily, the recycle efficiency of the energy is low. If the constant-power control form is used to recycle braking energy of the bus, at high speed, outputting braking torque of the secondary element and braking force of the bus are little, in the process of braking, the braking torque changes smoothly, which can ensure safety at high-speed braking and upgrade seating comfort, and it not only recycles inertial energy of the bus and decreases design power of the system, but also reduces mechanical wear of the bus and environmental pollution.

6. ACKNOWLEDGEMENT

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