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CONDITION MONITORING AND CONTROL OF ROLLING MILLS USING ELECTRO-HYDRAULIC SYSTEM

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ABSTRACT

Rolling mill automation systems gained prominence in the sixties when the servo system was introduced to the metal forming industry. With the rapid development of the measurement devices, computers and modeling techniques, rolling mill automation has made significant progress in recent years.

This paper introduces a new and simple electro-hydraulic system for controlling the rolling mill operation. The proposed system features less number of components, compactness, ease of installation and maintenance. It also exhibits the same rolling mill control performance as obtained when equipped with the conventional control systems. The results of the system simulation are experimentally validated, presented and discussed.

KEY WORDS

Rolling mills, pressure compensation, electro-hydraulic control, variable pumps

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NOMENCLATURE

Symbol	Description	Value	Unit
A_{cp}	Area of the control piston	8.1×10^{-4}	m^2
A_f	Compensator area factor	0.02	m
B	Effective oil bulk modulus	1×10^{9}	Pa
C_d	Coefficient of discharge	0.611	-
f_α	Equivalent angular viscous friction coefficient of swash plate	1.5	Nm/(rad/s)
F_r	Rolling force	-	N
f_v	Proportional valve viscous friction coefficient	90	N.s/m
f_x	Compensator viscous friction coefficient	48	N.s/m
I_e	Equivalent moment of inertia of the swash plate	0.0039	$kg.m^2$
I_s (A)(V)	Proportional valve control signal in Ampere / Voltage	-	(A)(V)
I_x(A)(V)	Compensator control signal in Ampere / Voltage	-	(A)(V)
k_α	Equivalent torsional spring stiffness of the swash plate	72	Nm/rad
k_i	Proportional solenoid force-current constant	2.5	N/A
k_p	Proportional gain	-	-
k_v	Proportional valve spring stiffness	20000	N/m
k_x	Compensator spring stiffness	4000	N/m
M_y	Moment acting on the swash plat around axis of rotation		N.m
m_v	Mass of the proportional valve spool	0.1	kg
m_x	Mass of the compensator spool	0.1	kg
p_c	Pressure difference across the control piston	-	Pa
$p_{c1,2}$	Pressure at the two sides of the control piston	-	Pa
P_D	Compensator downstream pressure	-	Pa
p_T	Tank line pressure	1×10^{5}	Pa
P_u / P_{ue} / P_{ur}	Compensator upstream pressure / error value / references value	-	Pa
p_v	Control pressure of the pump control unit	125×10^{5}	Pa
$Q_{a,b,c,d}$	Flow rates through proportional valve ports	-	m^3/s
Q_s/ Q_{se}/ Q_{sr}	Pump supply flow / error value / references value	-	m^3/s
R_L	Leakage resistance	1×10^{13}	$Pa/(m^3/s)$
R_s	Radius of swinging swash plate	.055	m
S/ S_e/ S_r	Valve spool displacement / error value / references value	-	m
$\dot{S}/\ddot{S}$	Proportional valve spool speed / acceleration	-	s^{-1}/ s^{-2}
$T_{d/i}$	Derivative/ Integral gain	-	-
$V_{c1,2}$	Control volume on the two sides of the control piston	-	m^3
V_{ci}	Initial control volume	13×10^{-6}	m^3
V_{ck}	Instantaneous cylinder volume of the k^{th} piston	-	m^3
V_o	Additional piston chamber volume	1×10^{-6}	m^3
V_r	Rolling speed	-	rpm
w	Proportional valve area factor	4.8×10^{-3}	m
X / X_e / X_r	Compensator displacement / error value / references value		m
$X_{cp(cpmax)}$	Control piston displacement (maximum value)	(0, 0.015)	m
$\dot{X}/\ddot{X}$	Compensator spool speed / acceleration	-	ms^{-1}/ms^{-2}
X_o	Compensator displacement overlap	0.001	m
$\alpha_{(sp)(e)}$	Swash plate angle of inclination (set point value) (error)	-	deg
$\dot{\alpha}/\ddot{\alpha}$	Swash plate angular velocity / angular acceleration	-	s^{-1}/ s^{-2}
ρ	Oil density	850	kg/m^3

1. INTRODUCTION

Electromechanical systems, as shown in Fig.1, were conventionally used for controlling the rolling mill operations. Due to the harsh environment and the intermittent high power demands of metal forming, rolling mills are highly dependent nowadays on fluid power control, particularly hydraulic systems.

In the intermediate rolling stage in which the rolled metal reaches close to its final controlled dimensions, both the rolling speed and the rolling load must be controlled in accordance with the independently varying driving torque that disturbs the control system.

Since the rolling speed and the rolling load are the key parameters for successful operation of the mill, several multivariable controllers have been applied to the control problem. However, this leads to complicated controllers, which are hard to tune on site.

Hydraulically driven rolling mills were always of interest to many of the control system designers. Some studies focus on the mill dynamic response using speed regulators, e.g. [1, 2] and some others focus on the response of the hydraulic position controller, [3, 4].

In order to control multivariable rolling mill operation, the industry approach is to employ several single loop controllers, [5, 6]. Rolling mill operation is a non-linear time-varying process with uncertain parameters in a very harsh environment.

In the present study, a simple hydraulic system has been developed in order to control both the rolling speed and the rolling load by controlling the supply flow and pressure, respectively, of a variable displacement pump. The system is composed of a servo-actuated variable displacement pump to supply the required controlled flow. An electro-hydraulic proportional throttle valve is used to control the downstream supply pressure of the variable displacement pump in accordance with the independently varying driving torque. Two separate control loops have been built separately. One of them is implemented to control the variable displacement pump supply flow while the other to control the supply pressure via controlling the proportional throttle valve, which works as pressure compensator.

2. DESCRIPTION OF THE HYDRAULIC CONTROL SYSTEM

The layout of the proposed hydraulic control system is shown schematically in Fig.2. Basically the system consists of the following four main components: a servo-actuated variable displacement pump “1”, a proportional pressure compensator “2”, a hydraulic single acting cylinder “3” and a hydraulic motor “4”. The pump “1” is used to control the rolling speed V_r by controlling its flow Q_s. The pressure compensator “2” is used to control the its upstream pressure P_u based on the change in its downstream pressure P_D. The hydraulic cylinder “3” and the hydraulic motor “4” are used to apply the required rolling load F_r and the driving torque T_r utilizing the pressures P_u and P_D, respectively. The most suitable pump for the proposed system is that which allows the flow rate to be electrically and remotely controlled via input signal. The pump “1” shown in Fig.2, is a servo-actuated axial piston swash plate pump in which the swash plate inclination angle is linearly proportional to the pump flow rate. The pump should be driven by an electric motor with adequate power. As shown in the figure, the pump swash plate is rigidly connected to a built-in symmetrical hydraulic cylinder, which drives the swash plate to change its inclination angle. Position of the piston of the symmetric hydraulic cylinder is controlled by means of a hydraulic proportional valve that is integral with the pump. A constant control pressure feeds the proportional valve from a secondary circuit.

Two LVDT position transducers that sense the swash plate position and the proportional valve spool displacement are used within the pump control scheme for controlling its supply flow. These transducers produce voltage signals proportional to the measured variables. The output signals are fed back to the electrical control system of the pump, which in turn feeds the proportional valve solenoid with the corresponding driving electrical control input signal I_s(A).

The pressure compensator “2” could be a proportional directional or throttle valve. Three sensors serve the control scheme of this pressure compensator. Two of them sense the upstream and downstream pressures of the compensator. The third one senses the compensator spool position. These transducers produce voltage signals proportional to the measured variables. The output signals are fed back to the electrical control system of the compensator, which in turn feeds the compensator solenoid by the corresponding driving electrical control input signal I_x(A). The hydraulic system contains also some other accessories that are required for oil conditioning, working conditions monitoring and for the safety of the test. These accessories are not shown in the figure. Pressure relief valve is located on the pump supply line. Maximum supply pressure is adjusted at 15% to 20% above the maximum required value of P_u (100 bar). Testing will be at working pressure less than 100 bar so that the pump supply flow is guaranteed to fully flow through the compensator. Both the hydraulic cylinder “3” and the hydraulic motor “4” should be adequately sized in order to satisfy the demanded rolling force F_r and the rolling speed V_r. Hydraulic motor should also be capable of withstanding the rolling torque T_r.

3. PUMP MATHEMATICAL MODEL

A mathematical model was developed in [7] to describe the dynamics of the electrically controlled variable displacement swash plate axial piston pump that has conical cylinder block. It must be noted that, in the proposed hydraulic control system, any electrically controlled variable displacement vane or piston pump could be implemented. However, the electrically controlled swash plate variable pump mathematical model could be briefly described as follows. The pump consists of a group of reciprocating pistons fitted in a cylinder block. The model first calculates the pump kinematics from which each piston stroke and absolute acceleration are calculated. Each piston chamber pressure is then found by solving the continuity equation for the control volume inside each piston chamber. Moments acting on the swash plate due to the whole group of pistons are calculated based on the vector summation of the pistons chambers pressure forces and absolute accelerations. These moments should be overcome by the pump control unit.

Figure 3 shows schematically the layout of the pump control unit that is presented symbolically in Fig.2. It consists of a proportional valve drives a symmetric hydraulic control cylinder, which is mechanically attached to the swash plate. When the proportional valve solenoid receives a control signal I_s(A) above zero, an electromagnetic force proportional to the input signal acts on the valve spool and causes it to move against its return spring. Simple second order differential equations are used to describe the dynamics of the proportional valve spool

$$m_v\ddot{s} + f_v\dot{s} + k_v s = k_i I_s \qquad (1)$$

The four control gaps of the proportional valve are changed accordingly. Flow rate through the valve control gaps are given by

$$Q_a = C_d w(s_{max} - s)\sqrt{\frac{2|p_{c1} - p_T|}{}}\ \mathrm{sgn}(p_{c1} - p_T) \qquad (2)$$

$$Q_b = C_d ws\sqrt{\frac{2|p_v - p_{c1}|}{}}\ \mathrm{sgn}(p_v - p_{c1}) \qquad (3)$$

$$Q_c = C_d w(s_{max} - s)\sqrt{\frac{2|p_v - p_{c2}|}{}}\ \mathrm{sgn}(p_v - p_{c2}) \qquad (4)$$

$$Q_d = C_d ws\sqrt{\frac{2|p_{c1} - p_T|}{}}\ \mathrm{sgn}(p_{c2} - p_T) \qquad (5)$$

Consequently, the control pressure that feeds the valve is redistributed across the piston of the symmetric hydraulic cylinder. Applying the continuity equation to the side chambers of the control piston resulted in

$$p_{c1} = \frac{B}{V_{c1}} \int (Q_b - Q_a - A_{cp}\, \dot{x}_{cp} - p_c / R_L)\, dt, \tag{6}$$

$$p_{c2} = \frac{B}{V_{c2}} \int (Q_c - Q_d + A_{cp}\, \dot{x}_{cp} + p_c / R_L)\, dt \tag{7}$$

where $V_{c1} = V_{ci} + A_{cp}x_{cp}$ and $V_{c2} = V_{ci} - A_{cp}x_{cp}$

Piston of the control cylinder is attached mechanically to the pump swash plate. The pressure difference on the two sides of the control piston drives the swash plate to a new equilibrium position. Considering the moment acting on the swash plate due to piston group My, and the swash plate moment of inertia I_e, swiveling of the swash plate is described by the following second ordered differential equation.

$$I_e\ddot{\ }=(p_{c1}-p_{c2})A_{cp}r_s+M_y-f\,\dot{\alpha}-k\,(\alpha+0.09) \tag{8}$$

4. PRESSURE COMPENSATOR MODELING

As shown in Fig.2, an electro-hydraulic pressure compensator is located on the delivery line of the pump. The compensator is shown schematically in Fig.4 and is modeled as follows. When the compensator solenoid receives a control signal I_x(A) above zero, an electromagnetic force equals $I_x k_i$ proportional to the input signal acts on the valve spool and causes it to move against its return spring. Compensator displacement is found by solving the following second order differential equation.

$$m_x\ddot{X} + f_x\dot{X} + k_x X = I_x k_i \tag{9}$$

Assuming constant discharge coefficient, zero leakage and negligible oil compressibility effect, the pressure difference across the compensator is proportional to the square of the flow rate through it. Compensator upstream pressure is then given by:

$$P_u = P_D + \frac{\rho}{2}\left[\frac{Q_s}{C_d A_f (X - X_o)}\right]^2 \tag{10}$$

5. DESCRIPTION OF THE ELECTRICAL CONTROL SYSTEM

As shown in Fig.5, the overall electrical control system is composed of two interacting control schemes. First one controls the pump flow rate Q_s and the other controls the compensator upstream pressure P_u. Each control scheme consists of double negative feedback control loops.

The inner feedback control loop in the pump control scheme is used for the accurate positioning of the proportional valve spool using displacement sensor and PID controller. In this loop, a displacement sensor measures the actual spool position S of the proportional valve and feeds back the corresponding value in voltage. A signal conditioner reconverts this signal into a corresponding percentage of the spool position to be compared with a reference spool position S_r. PID controller receives the error in the spool position S_e and provides a corresponding control signal I_s(V) in voltage.

An amplifier is then used to provide a current signal I_s(A) powerful enough to drive the valve solenoid for positioning the valve spool in the required position. Similarly the outer loop works to control the swash plate inclination angle and hence the pump flow rate. The only difference is that the controller is of PD type because the outer loop represents a system type 1 that has zero steady state error due to the presence of the symmetric hydraulic cylinder as a physical integrator. Under steady state conditions, error values for both the pump and the valve are zero and the valve spool is positioned at the center in order to hold the swash plate at the current position.

Construction of the compensator control scheme is quite similar to that used to control the pump flow rate. The inner feedback loop is used to control the compensator spool position, while the outer loop takes care of the compensator upstream pressure. PID controllers are used in both loops. If the downstream pressure increases, the compensator opens more widely so that its resistance to the pump flow reduces, and vise versa. The compensator works to compensate the effect of the independent change in the downstream pressure so as to keep the upstream pressure constant.

By controlling the flow rate and the compensator upstream pressure the rolling speed and the rolling load, respectively, are consequently controlled. The pump and the compensator control systems interact with each other. Both control schemes are working interactively. The change of the compensator upstream pressure P_u causes a corresponding change in the moments acting on the pump swash plate that should be overcome by the pump control system. The change in the pump flow rate causes a corresponding change in the pressure P_u. Rolling speed and rolling load as final controlled parameters are still controlled in open loops. However, this is acceptable in the intermediate stages of the rolling process.

6. SYSTEM SIMULATION AND EXPERIMENTAL TESTING

Software based on Matlab-Simulink is developed in order to simulate the performance of the rolling mill electro-hydraulic control system. As shown in Fig. 5, the simulation software consists of subsystems constructed and integrated to simulate the entire electro-hydraulic control system. The pump used in this study is of 40 cc/rev geometric volume. Design parameters of the whole electro-hydraulic system are presented in the Nomenclature.

The empirical-analytical method "Ultimate Sensitivity" introduced by Ziegler–Nichols [8], is used to parameterize the PID controllers of the pump and the compensator. In this approach, the proportional gain is increased until the system becomes marginally stable and continuous oscillations appear. The corresponding proportional gain is defined as the ultimate gain and the oscillation period is defined as the ultimate period. Then the controller parameters are selected as follows. The control signal that is coming out of the controller equals $K_p[1+1/T_iS+T_dS]$, where K_p=0.6 of the ultimate gain, T_i=0.5 of the ultimate period and T_d=0.125 of the ultimate period. Using such an approach, as reported in [7], the parameters of the PID proportional valve controller are found to be K_p=1, T_i= 0.01and T_d= 0.001. The parameters of the PD pump controller are K_p=1and T_d=0.02. Same method is applied for the PID controllers of the compensator control scheme. Parameters of the compensator spool position controller are K_p=1, T_i=0.007 and T_d=0.005. Parameters of the compensator upstream pressure controller are K_p=20, T_i= 0.2 and T_d= $5x10^{-5}$.

As shown in the Fig.6, the test setup consists of the hydraulic control system interfaced with real time control and data acquisition system. Function generator has been implemented in the system to behave as a virtual mill produces rolling torque disturbs the control system. Real time control software has been built to replace the physical electrical control system in order to facilitate the tuning and prototyping of the controllers of the proposed control schemes. Figure 6 shows also separate hydraulic circuit is used to supply the control pressure required for the proportional valve integrated with the pump. It consists of an electric motor of suitable output power coupled with the control pressure pump that can afford pressure up to 15 MPa. The control pressure pump draws the fluid from the main reservoir via a100 μm mesh size strainer and discharges it through a pressure line filter of a proper flow capacity and a 5 μm mesh size. Control pressure supply line is connected in parallel to an accumulator in order to absorb the possible variation of the control pressure. The control pressure pump is protected against overloading by a pilot operated pressure relief valve integrated with the unloading valve electrically actuated by a push button in order to remotely apply or release the control pressure. Oil is kept clean by using online return filter in the control pressure return line. Water-type oil cooler is connected in parallel to the main return line, after the return filter, in order to partially cool the return oil and keep its temperature within 55 to 60oC as recommended. The cooler is connected to the cooling water supply and return via a pair of shutoff valves. The oil tank is equipped with a thermometer, an air breather and an oil level indicator.

In the reality of the rolling mills operation, when the hydraulic motor encounters varying resistance from the mill, rolling torque varies accordingly. Assuming hydraulic motor of constant efficiency, the pressure P_D varies linearly with the driving torque. The proportionality factor is function of the hydraulic motor volumetric size. In our simulation and testing process, normalized value of the rolling torque was shown in the investigation based on a value of 10 MPa maximum working pressure. The generated actual applied rolling torque could be calculated by selecting the suitable motor size. For simulation purposes, see Fig.5, P_D is supplied by a function generator as a part of the simulation system. Calculation of P_u then follows equation 10. For experimental purposes, see Fig.6, a physical function generator replaces the actual pressure sensor that senses the P_D from the rolling mill. Because the actual P_D in the experimental setup is the reservoir pressure, the actual sensed value of P_u represents the second term of the right hand side of equation 10. Therefore the generated value of P_D should be added to the actual sensed value of P_u. the resulted value is that should be compared with the reference value P_{ur}.

Rolling speed V_r is linearly proportional to the pump supply flow rate. The proportionality factor is function of the hydraulic motor volumetric size. Measurement of the pump supply flow then represents the rolling speed. Pump supply flow is linearly proportional to the swash plate inclination angle that is measured using displacement sensor. Normalized value of the rolling speed was shown in the investigation based on a value of 60 l/min maximum flow rate. The required actual rolling speed could be easily calculated by selecting the suitable hydraulic motor size.

Rolling force F_r is linearly proportional to the controlled pressure P_u. The proportionality factor is function of the hydraulic cylinder area. Measurement of the controlled pressure P_u then represents the rolling force. Pressure P_u is measured using a pressure sensor. Normalized value of rolling force was shown in the investigation based on a value of 10 MPa maximum working pressure. The required actual rolling force could be calculated by selecting the suitable hydraulic cylinder dimensions.

Electro-hydraulic compensator spool displacement is measured by a displacement sensor. Normalized value of compensator spool displacement is shown in the investigation based on spool full stroke displacement of 4 mm.
Computer runs were carried out to simulate the performance of the system. Different rolling torque signals are used in the simulation. Results are presented in Figs 7 to 9. Simulation results are compared with the experimental measurements and are presented in Figs 10 to 12.

7. SYSTEM PERFORMANCE INVESTIGATION

Normalized values are used in order to generalize the results of the investigation. Reference rolling speed and rolling force are assumed arbitrarily to be at 40% and 60% of their nominal values.

Static characteristics of the hydraulic control system are first investigated. Rolling torque is assumed to change gradually from zero to 100%. Simulation results presented in Figs 7 and 10 show that the pump control scheme keeps the rolling speed constant despite the disturbance torque. Experimental results show that the speed fluctuates by nearly 1% around the steady state value due to the effect of the hydraulic lines. Results show also that whenever the pressure difference across the compensator is more than 4% of the upstream pressure, the later is kept nearly constant as the reference value fed to the control system. During this period, the compensator control system drives the compensator to compensate the increase in the downstream pressure from zero to 56%. Once the pressure difference across the compensator becomes less than 4% of the upstream pressure, the later is increased accordingly while the compensator is saturated in fully opned position.

As presented in Figs. 8 and 11, the rolling speed is shown unaffected by the stepwise change in the rolling torque. In reality, rolling speed changes marginally due to the change in the volumetric efficiencies of both the pump and the hydraulic motor. This change is negligible at this stage of the rolling process. Load shocks at the transient periods could be minimized by using an adequately sized accumulator on the line of the upstream pressure.

A sinusoidal signal, of 2.5% amplitude and frequency of 0.25 Hz., represents harmonic change of the rolling torque. Theoretical and experimental system responses are presented in Figs.9 and 12, respectively. Results show that, the rolling speed is not affected. The compensator responds in sinusoidal movements in order to keep the rolling load constant. More investigations are recommended to study the system frequency response over a wide range of frequencies.

The relative agreement between the simulation and the experimental results fairly validates the model

8. CONCLUSIONS

Metal forming and rolling mills are nowadays highly dependent on the hydraulic power. In the intermediate stages of the rolling mills, both rolling speed and rolling load must be controlled.

In this paper, an electro-hydraulic control system has been proposed to control the rolling speed and rolling force. The hydraulic system basically consists of a variable displacement pump and a pressure compensator, both controlled electrically. The electrical control system is composed of two interacting control schemes. First one controls the rolling speed and the second controls the rolling force. Each of them consists of double negative feedback control loops. The proposed system has been simulated, realized and experimentally tested. Design of the control system revealed the following characteristic features of the rolling mill.

With the assumption of negligible internal leakage of the pump and the hydraulic motor, the pump control system keeps the rolling speed constant despite the changes in the rolling torque. Statically, rolling load is kept constant whenever the pressure difference across the compensator is more than 4% of the upstream pressure. Dynamically, rolling load experiences shocks at the transient periods of the stepwise change in the rolling torque. Results show also that the harmonic change of the rolling torque is compensated and lead to a constant rolling load as the reference value.

The good agreement between the simulation and experimental results validates the system model and the analytical findings.

Recommendations for future investigations are as follows. Use of an adequately sized accumulator on the upstream pressure line in order to reduce the transient period and to minimize the rolling load shocks due to the stepwise change in the rolling torque. Study of the system response to harmonic change in the rolling torque over a wide range of frequencies and amplitudes are also recommended. In the present study, both rolling speed and rolling load reference values are assumed constants. In future investigations, different combinations of the working conditions may be assumed in order to study the capacity of the proposed control system to cover various situations.

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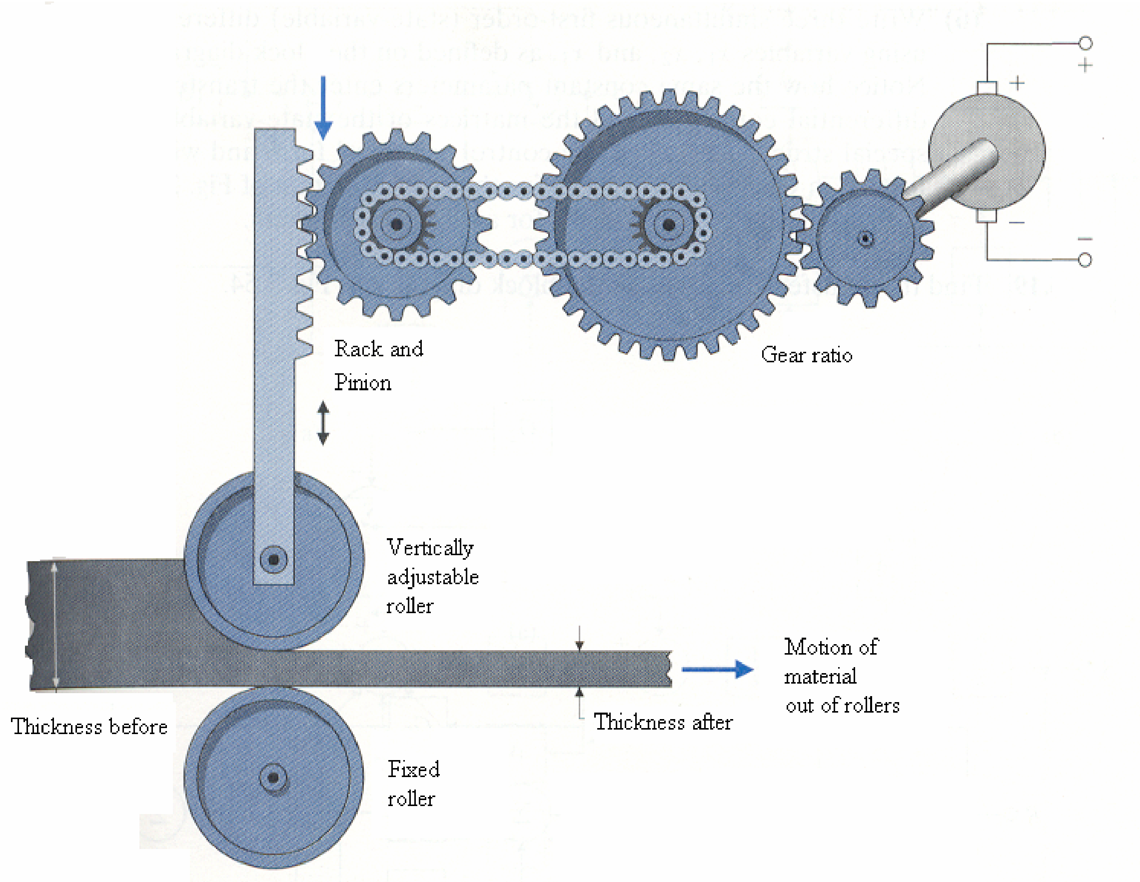

Fig.1 Conventional system for controlling rolling mills operations

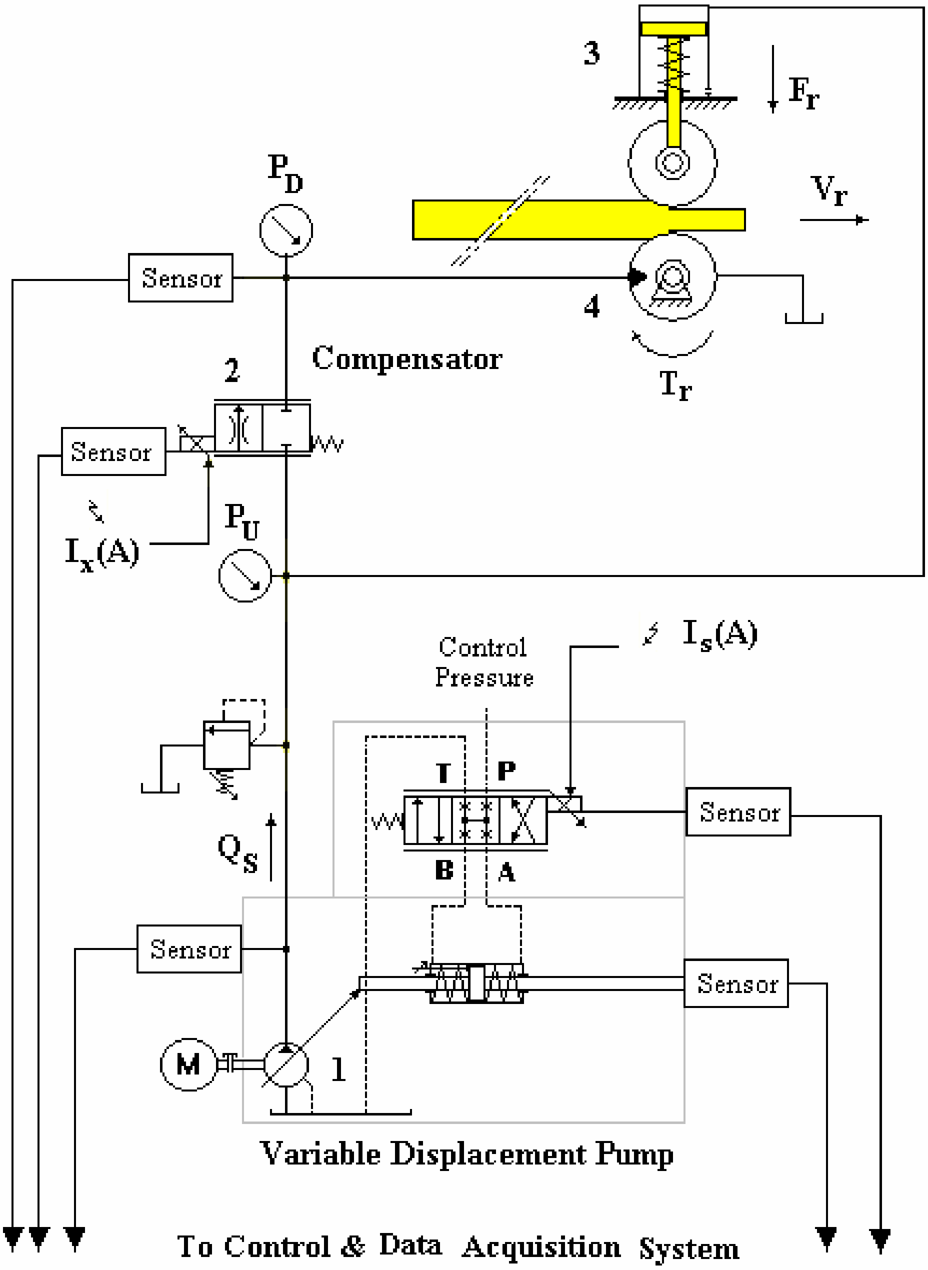

Fig. 2 Symbolic representation of the hydraulic control system

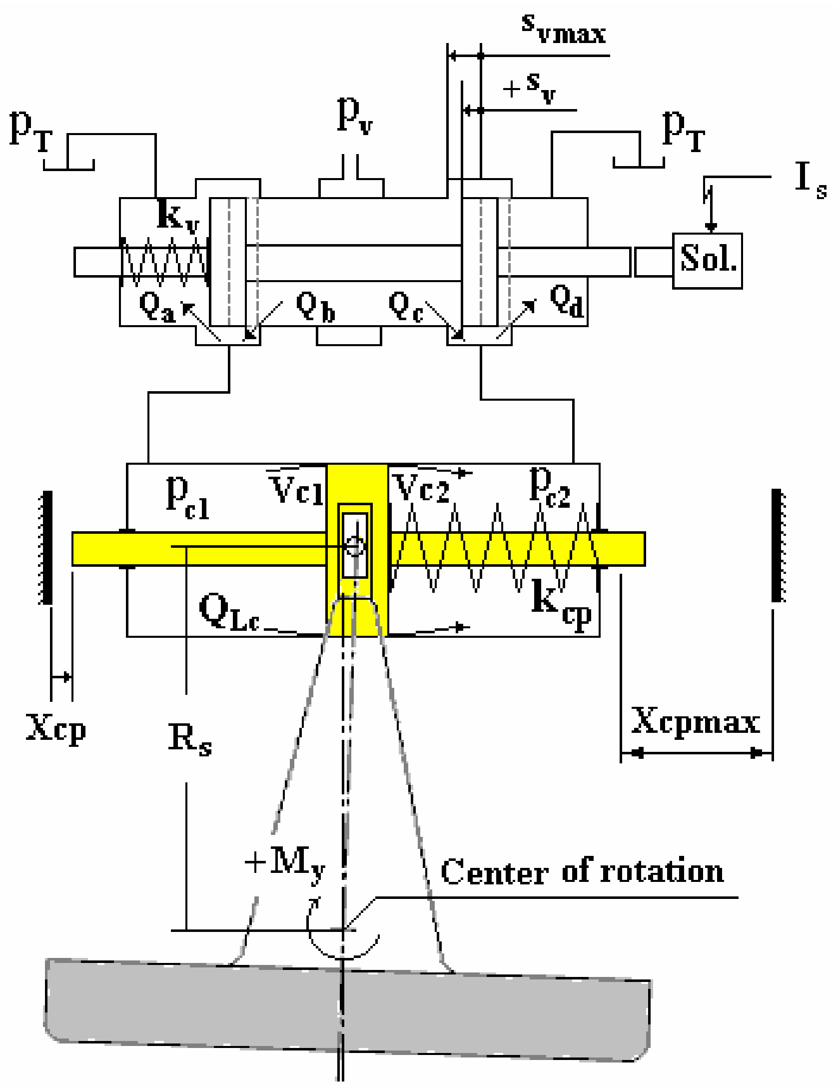

Fig. 3 Schematic representation of the pump swash plate attached to the pump control unit

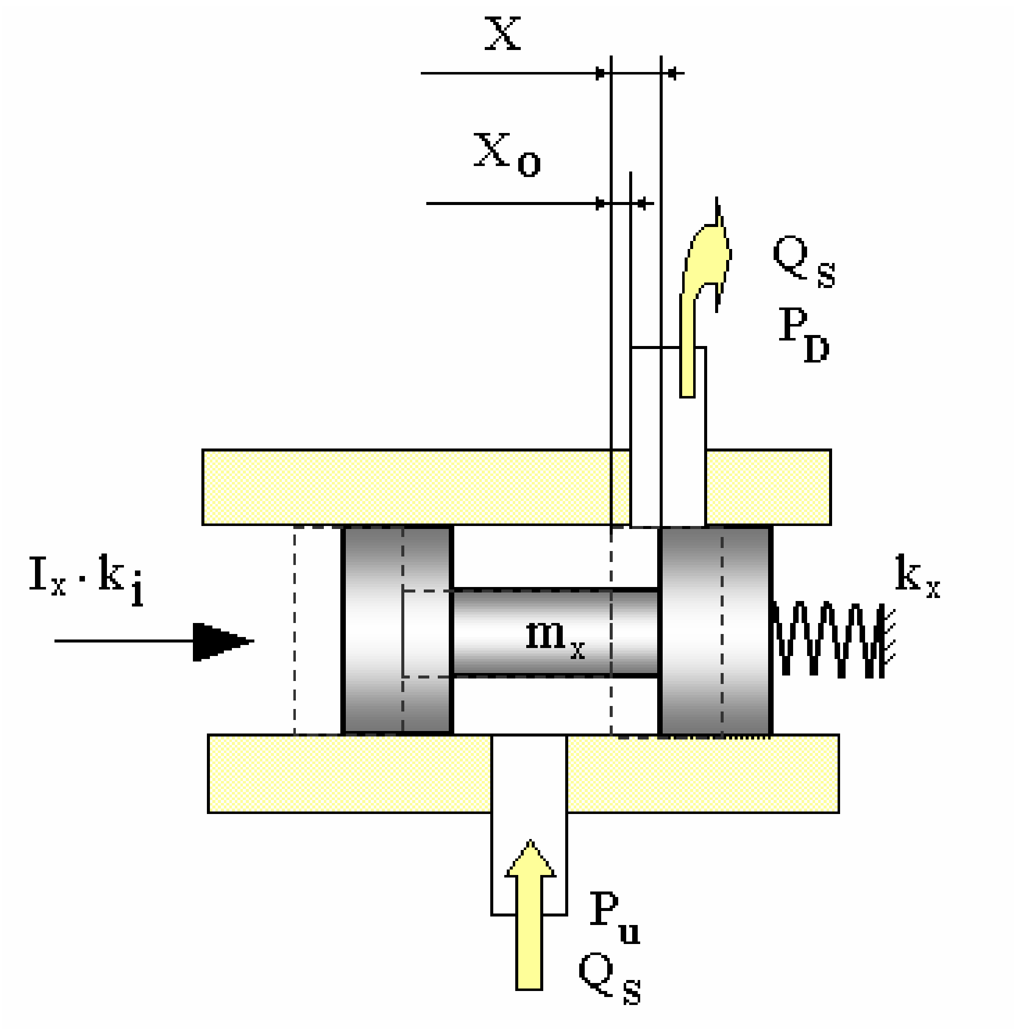

Fig 4 Schematic of the pressure compensator

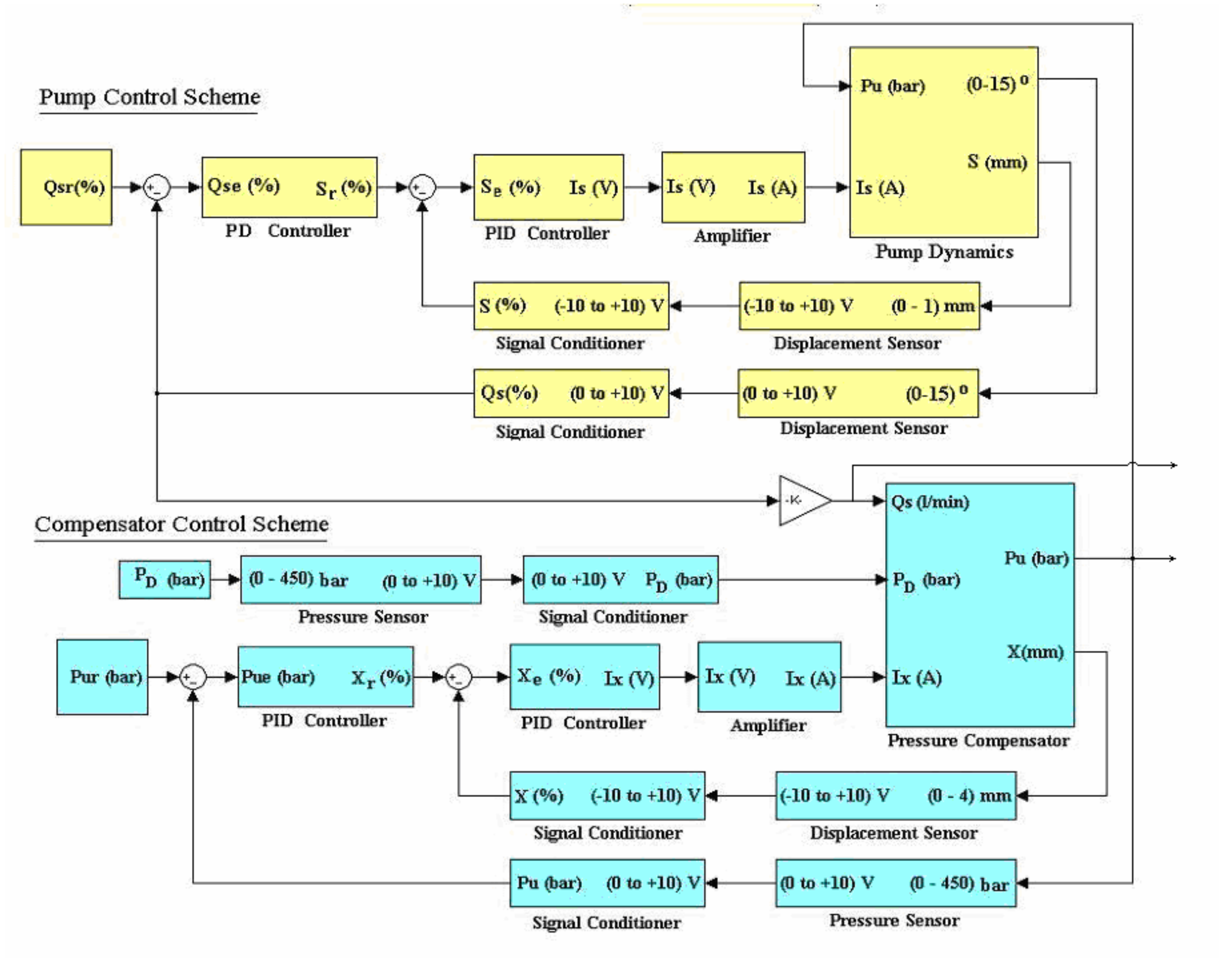

Fig. 5 Block diagram of the control schemes and simulation subsystems

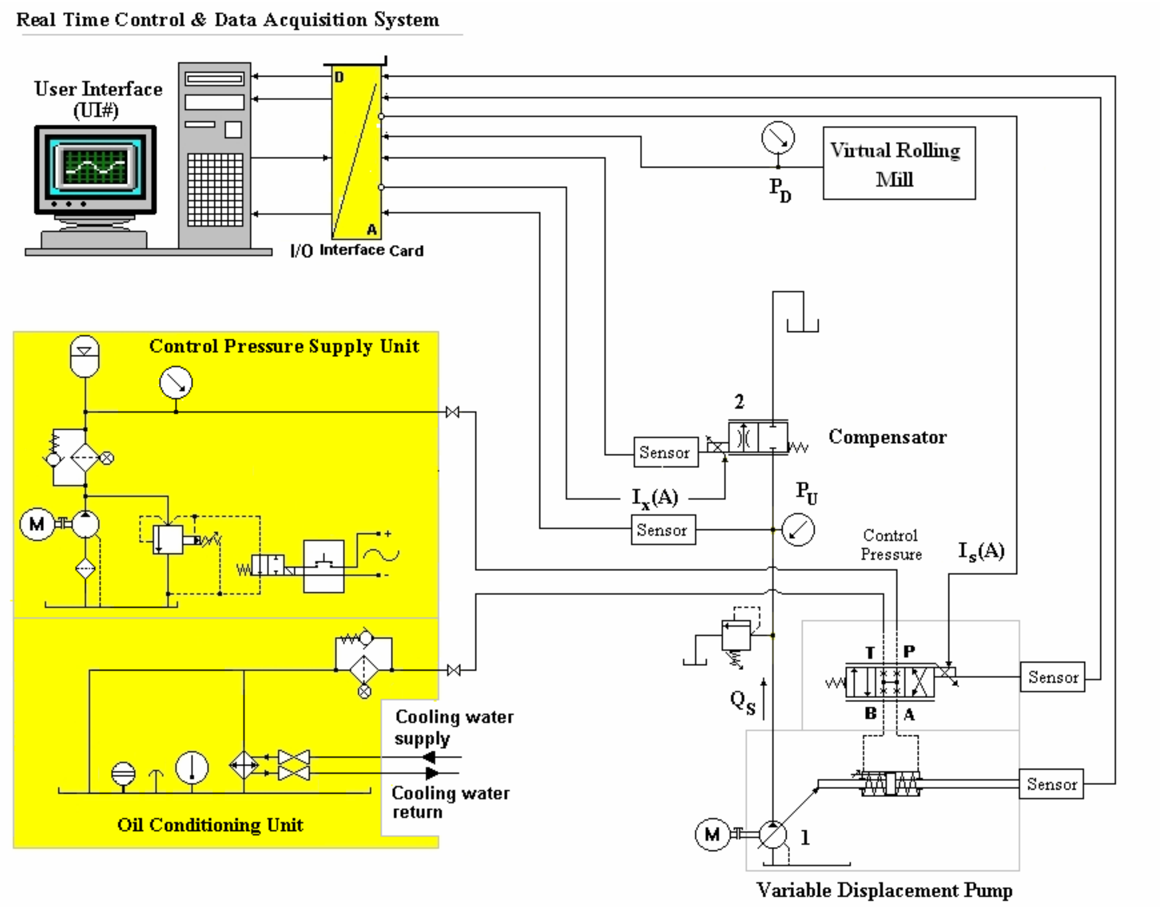

Fig. 6 Schematic diagram of the test setup

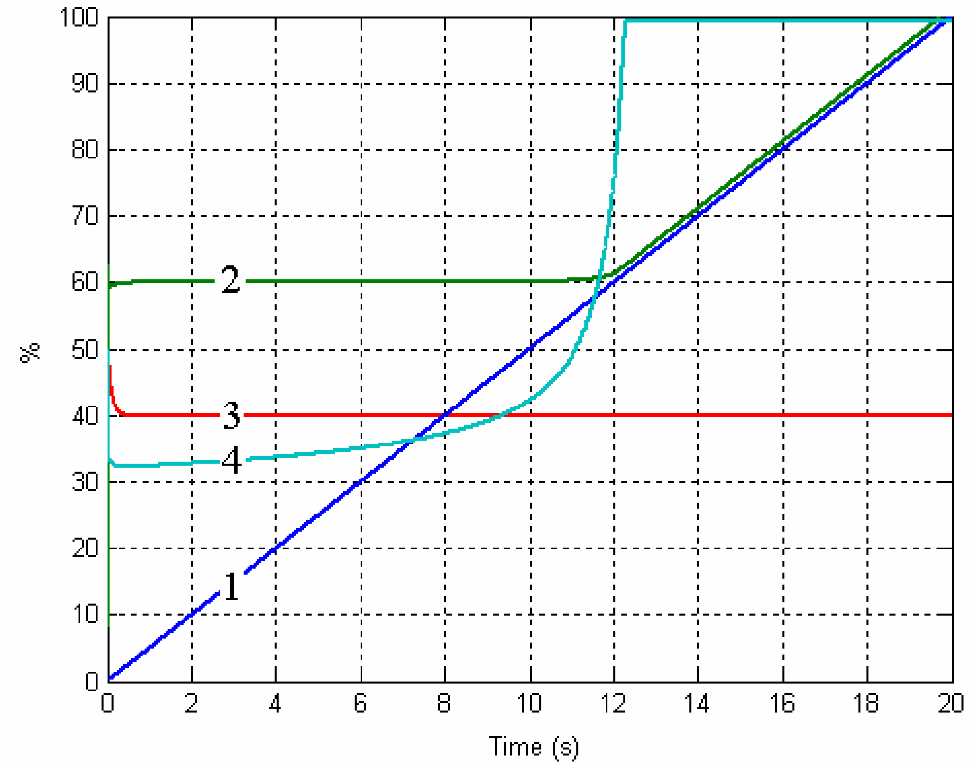

Fig. 7 Theoretical static characteristics

(1-Rolling torque, 2- Rolling Force, 3- Rolling speed and 4- Compensator displacement)

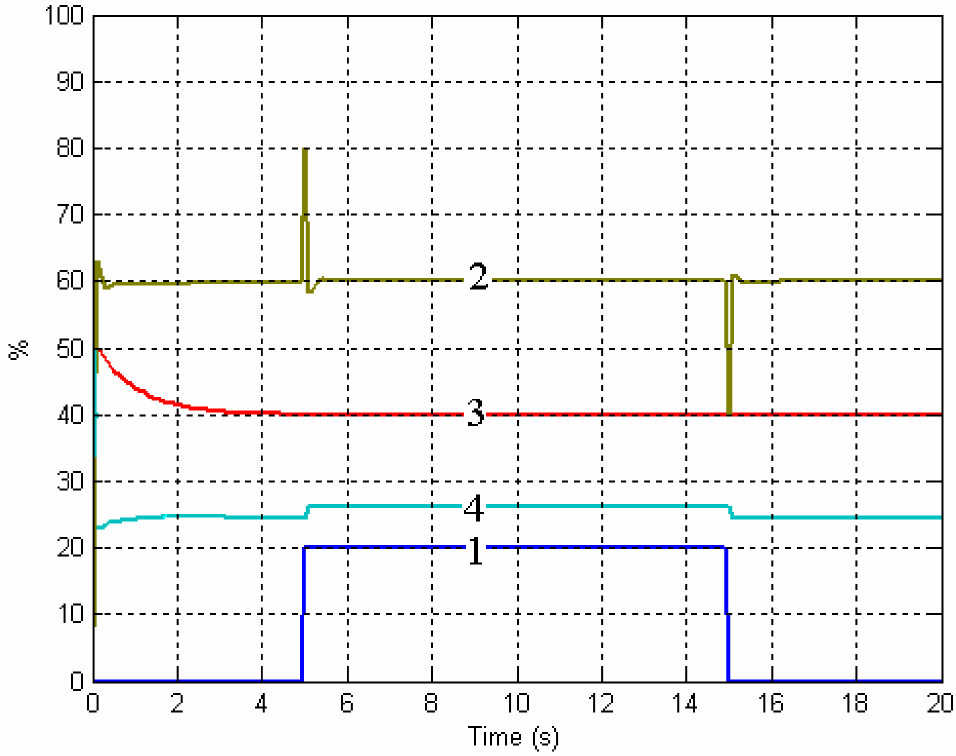

Fig. 8 Theoretical step response

(1-Rolling torque, 2- Rolling Force, 3- Rolling speed and 4- Compensator displacement)

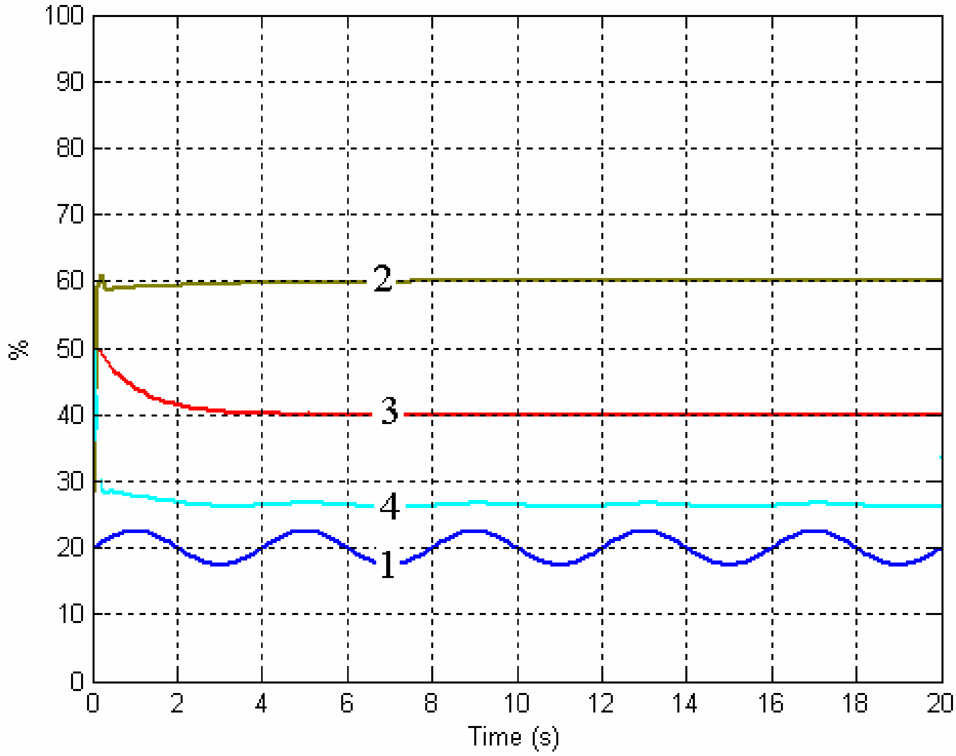

Fig. 9 Theoretical response to harmonic input
(1-Rolling torque, 2- Rolling Force, 3- Rolling speed and 4- Compensator displacement)

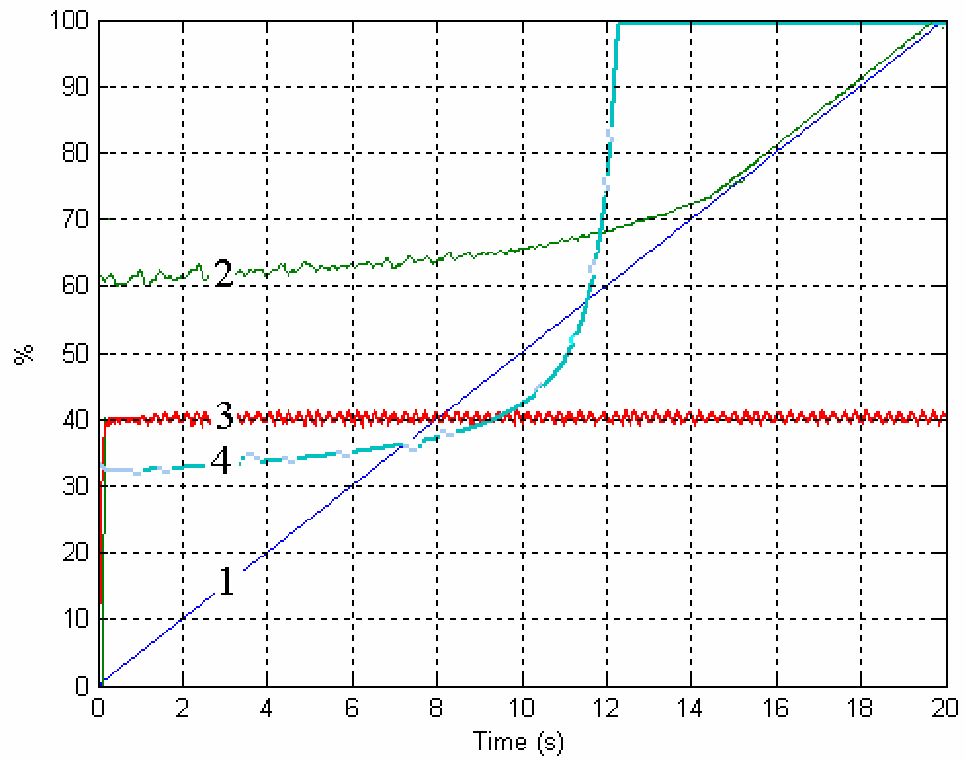

Fig. 10 Experimental static characteristics
(1-Rolling torque, 2- Rolling Force, 3- Rolling speed and 4- Compensator displacement)

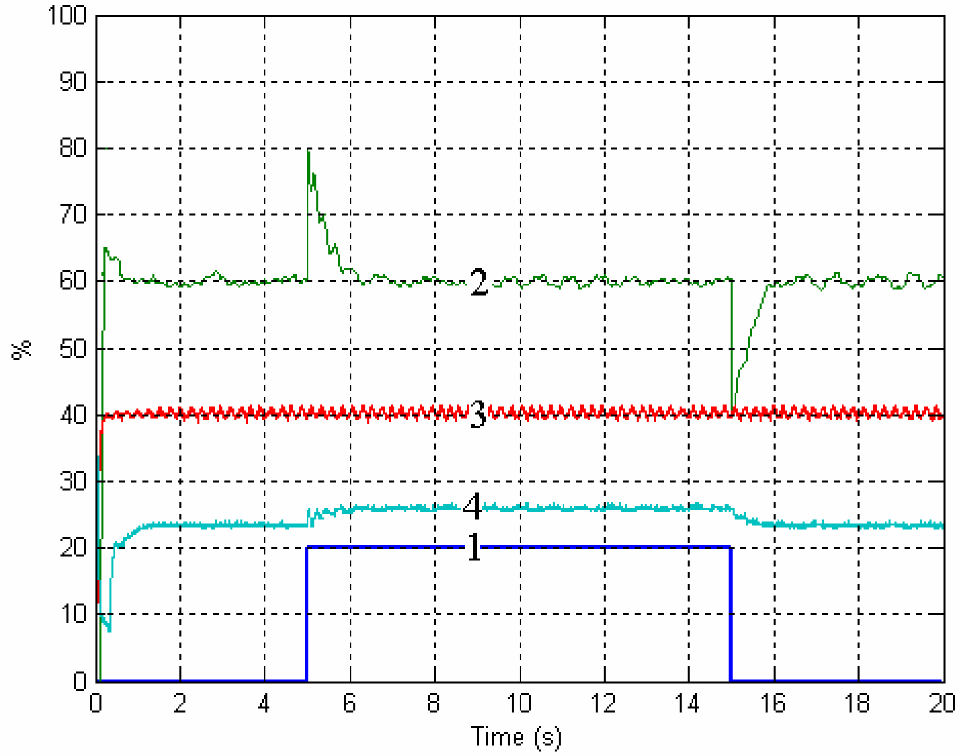

Fig. 11 Experimental step response

(1-Rolling torque, 2- Rolling Force, 3- Rolling speed and 4- Compensator displacement)

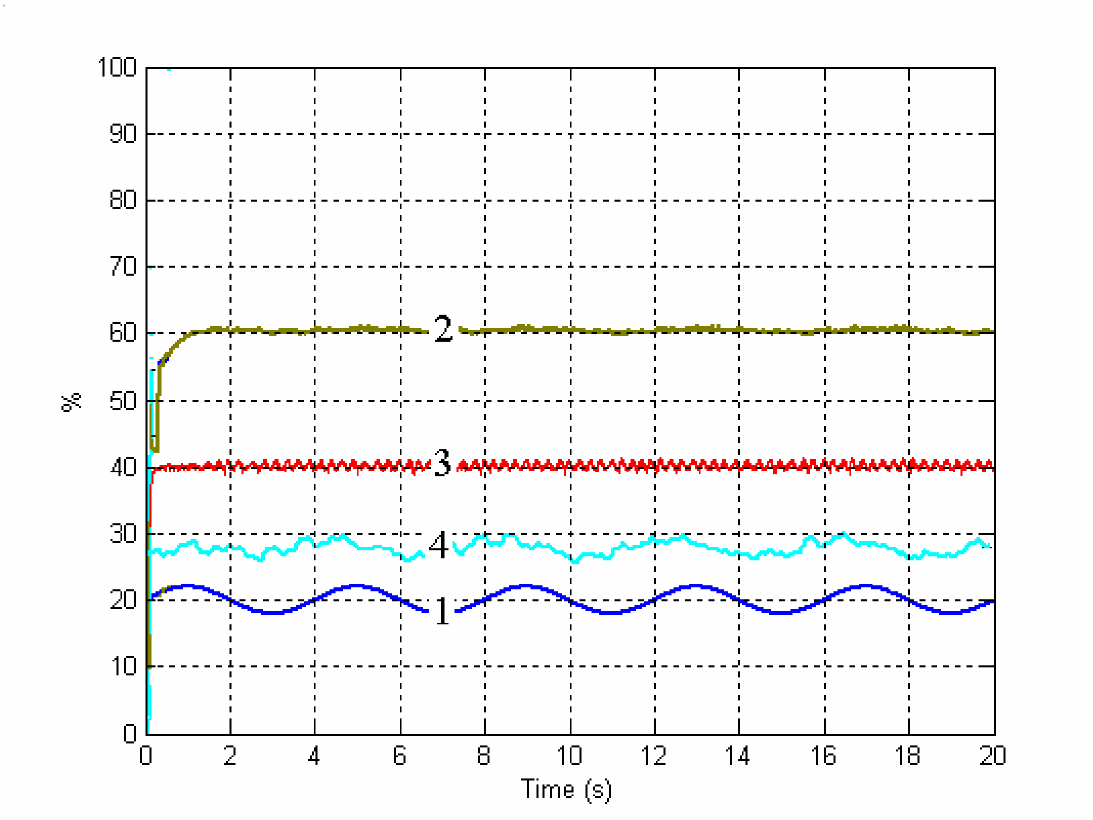

Fig. 12 Experimental response to harmonic input
(1-Rolling torque, 2- Rolling Force, 3- Rolling speed and 4- Compensator displacement)