



Mechanically Closed Loop Gearbox Test Rig Controller

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ARTICLE INFO	ABSTRACT
<p>Article History: Received 2 November 2019 Received in revised form 9 January 2020 Accepted 3 May 2020 Available online 3 May 2020</p>	<p>Most of the techniques used to monitor and diagnose faults from machines are usually based on additional measurements which require high setup costs and installation difficulties. This paper focuses on developing controller of a new method to monitor and detect problems of a gearbox transmission system based on the parameters acquired from control systems. The control data which are usually available in most of machines, including actuator current and valve flow, equivalent tensional stiffness, rotary inertia, load set point, speed demand, motor current and torque feedback have been explored based on gearbox test system[1]. Electro hydraulic actuators are widely applied in several engineering fields. However, their design is not a simple task since it is necessary to observe their behavior according to control theory. A nonlinear regression model is adopted to simulate the test condition and survey the results in the present of disturbances and any kind of interferences. The objective of this study was to obtain physical model of an electro- hydraulic actuator and provide the electro-hydraulic system which depends on the hydraulic actuators to meet the motion control system demands. However, their design is not a simple task since it is necessary to observe their behavior according to control theory. The core of the paper depends on comparing the results of two corresponding control methods (On-Off control and PID control) and the impact of the noise into the control input signal. The results confirm that the PID controller modifies the response even in presence of the noise in control input signal in accuracy, rise time, settling time, and speed.</p>
<p>Keywords: Test rig, Mechanically closed loop, Electro- hydraulic actuator, On-Off control, PID Control</p>	

1. INTRODUCTION

Test rigs as integral parts of industry are widely used in various industries. These systems shall be designed in a way that have the minimum energy loss and have lower capital and maintenance costs. Among these systems, mechanically closed-loop test system is the best [2]. One of the most important components of this system and its control section is torque inducer part. In some articles the method of the torque inducer control in mechanically close-loop are studied. In these systems torque is applied by apply twisting to the loop in different ways. Among these methods, using of hydraulic actuator to provide the necessary force for torque inducer system or an electric drive system with gearbox is more popular. [3, 4, 5, 6] There are many different ways for controlling these kinds of

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hydraulic actuators. Hydraulic actuators are known for their high power and have an important role in modern machine equipment. However, these behaviors are due to their complex nonlinear dynamics. This non-linear behavior can be caused by friction, compressibility of the fluid in the hydraulic actuator or other reasons. Using these actuators is possible and more comfortable by using modern digital control systems which are designed based on modern control theory, enables operators. The torque control is possible in different methods including actuator feedback controller design pressure, position control, speed of shaft movement, etc. Electronic proportional actuators and servo valves are used in the design of the controller drivers. Moreover, positional converters and force sensors are used as feedback for making closed loop control systems. The method of achieving good design in this study is based on study of past experiences and articles in closed loop test rigs with mechanically energy regeneration. In cases where several projects are technically optional, decision making is based on reliability and output quality. Moreover, the expense ratio index has also been considered.

In this paper planetary torque inducer is used in order to induce torque in the mechanically closed loop by rotating the planetary gearbox ring. For calculating the required amount of skin rotation the Torsional stiffness is required.

This paper is organized as follows. Mechanically closed loop is described in Section 2, which is followed by the system formulation in Section 3. In Section 4, controller is analyzed. To verify the PID performance and compare both controllers (on-off and PID) results simulations are performed in Section 5. The results are carried out to show the better control performance even if in the varying external disturbance as in real working conditions. Finally, some remarking conclusions are summarized in Section 6.

2. MECHANICALLY CLOSED LOOP SYSTEM

In the mechanically closed loop systems as shown in figure 1 energy do not return to the driver instead circulates inside the mechanical parts of the loop [7]. In mechanically closed loop systems a driver is considered for setting up the system which undertakes the task of compensating loss of component. As specified in the mechanically closed-loop method the consumed energy is much lower than the open looped one due to energy recovery and heat is the only consumed energy loss in different parts of the loop. The Mechanically closed loop system is low cost method due to the lower engine driver power and omission the energy conversion part. In contrast, the number of mechanical components is considerable in comparison with the other methods. [7]

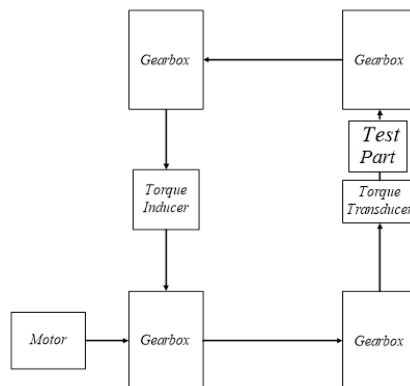


Fig. 1. Closed loop test rig with mechanical energy regeneration

Mechanically closed loop method or four squares is made of a chain of mechanical components that the beginning and the end of the chain since connect them to each other should have absolutely identical rotational speed. In order to produce circulating power in the closed loop we should create torque and rotational speed independently in the process. Since all components are engaged with each other in a closed loop for creating torque we should form locked torque proportional to angle difference and torsional stiffness of the closed loop component before all components are connected to each other. For creating the second factor for producing power (rotational speed) we could make motion from the arbitrary point in closed loop so all the components start moving. The mechanically closed loop made of three parts: Power generation, Power transmission, Torque induction.

2.1. Power generation section

If components in loop are used with full efficiency, primary motion and by overcoming to inertia then all the components in the closed loop could have been tested under the load and without spending energy. But because of the loss as shown in figure 2 the level of the power in the direction of the energy flow reduces. Therefore the total loss of loop component ($P_{fA1}+P_{fA2}$) should have been recovered by the input power.

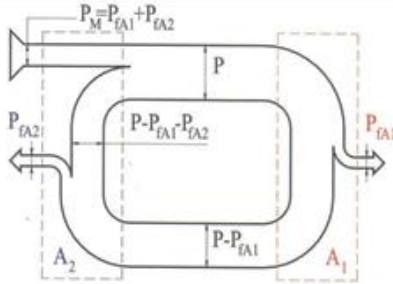


Fig. 2. Losses in mechanically closed loop

2.2. Power transmission

Mechanically closed loop chain is composed of a series of mechanical components which is used for power transmission tasks and they also could have been tested in the chain. [7] If we could test several similar pieces at the same time we should consider that the amount of power which is received by the components is reducing in the power flow direction. So, to achieve maximum throughput in all tested components some of them need to get more power which may causes some problem for them. Depending on the type, speed and power of the test piece the proper power transmission components should be used. So the losses will reduce by reducing the number of components and the system response will speed up. As shown in figure 3 the closed loop could be formed by two pairs of gearboxes or pulleys and their turn ratio is based on the test piece turn ratio.

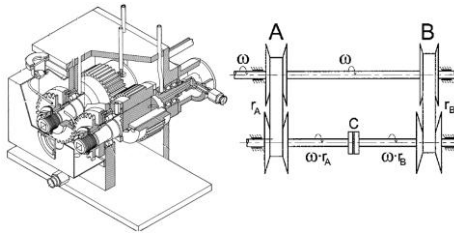


Fig. 3. Torque induction

As discussed, angle difference in mechanically closed loop for producing locked torque is necessary. The required mechanism by which locked torque is generated in mechanically closed loop is called torque induction system as shown in figure 4 and made of two parts: Static Dynamic.

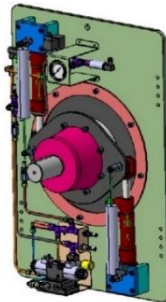


Fig. 4. Torque induction systems with planetary gearbox

In static method for making specified torque the closed loop should be in static position. Therefore, torque changing during process is impossible. The most simple torque induction mechanism is shown in figure 5 which is made of two opposing disk with wholes on their outsides.

The other disk is twisted for making specified torque induction. Two disks are connected to each other by wholes on their body.

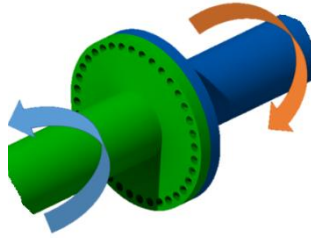


Fig. 5. Torque induction mechanism with perforated disks

In dynamic methods, static position for closed loop system is not required. The induction torque method in dynamic mechanism is made in different ways: helical gearbox, hydraulic mechanism, differential mechanism and planetary method.

As we mentioned before the planetary method is used in this article for torque induction operation. In this method as shown in figure 6 planetary gearboxes made of sun and planetary gearboxes and rings. In typical applications the planetary rings attached to its crust. If the gear ring spins around gearbox axis gearbox input and output axis rotates in their opposite side and cause angle difference proportional to their ratio. We can say this method is the most common way to create torque in closed loop.

$$\frac{1}{k_{eq}} = \frac{1}{k_1} + n^2 \left(\frac{1}{k_2} + N^2 \left(\frac{1}{k_3} + \frac{1}{N^2} \left(\frac{1}{k_4} + \frac{1}{k_5} + N^2 \left(\frac{1}{k_6} + \frac{1}{N^2} \left(\frac{1}{k_7} \right) \right) \right) \right) \right) \right) \quad (1)$$

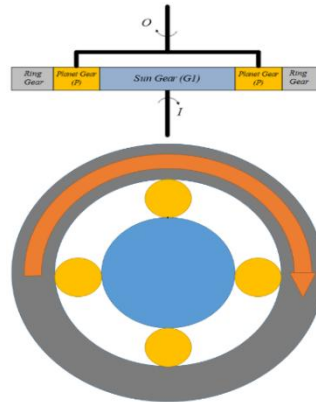


Fig. 6. Planetary mechanism

3. PROBLEM FORMULATION AND DYNAMIC MODELS

3.1. Mechanical system modeling

In order to ensure the overall efficiency of the mechanical parts of the system a model of system is created in the AMESIM software and system response is investigated for different inputs.[8] In this model torque induction parts including cylinder, jacks and hydraulic circuits are used for making angle in loop. Moreover, system shafts in the form of torsion spring with the same torsional stiffness modeled and gear’s rotational inertia is considered. All the values in the software are same as those in the actual system. The extracted model is shown in the figure 7.

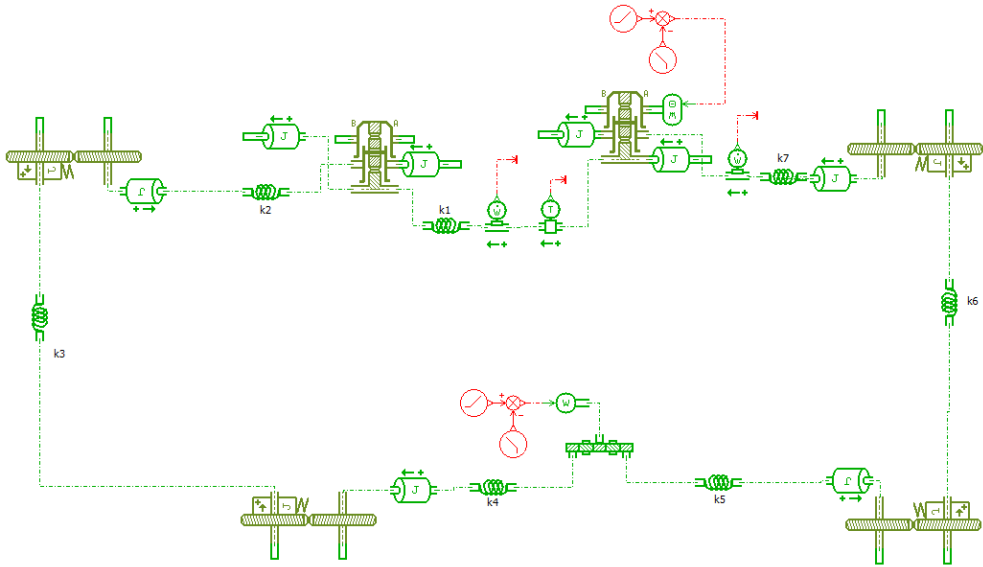


Fig. 7. Mechanically closed loop schematic

3.2. Model Parameters

For simplifying system modeling extraction we calculate critical mechanical parameters. Corresponding parameters like torsional inertia and torsional stiffness are calculable based on conservation rules.

$$\begin{aligned}
 J_{eq} &= 2J_s + J_1 + \left(\frac{1}{n}\right)^2 (J_p + J_{g1} + J_2 + \left(\frac{1}{N}\right)^2 (J_{g2} + J_2 + J_{g2} + \\
 &(N)^2(J_{g1} + J_4 + J_{g1} + \left(\frac{1}{N}\right)^2 (J_{g2} + J_2 + J_{g2} + \\
 &(N)^2(J_{g1} + J_2 + J_{pc})))
 \end{aligned} \tag{2}$$

In these formulas we modeled system components by springs and put their value in the equations.

Since, planetary gearbox is the kernel of our study we consider planetary gearbox with $n=3.4$ ratio and we have the corresponding value as mentioned in equation3.

$$n = 3$$

$$N=1.7$$

$$k_{eq} = 623.9709 \text{ N/m}$$

(3)

3.3. Extracted model verification

For verifying the mathematical extracted model and with respect to its complexity we could make a similar condition as shown in figure 8 and giving different inputs and making angle in closed loop. This verification could be simpler.

Moreover, by changing the motor cycle rate in system and measuring the torque and acceleration we could verify the authenticity of the extracted model.

Table 1. Model Parameters

Parameters	Description
J_{g1}	Small 90 degree gearbox inertia
J_{g2}	Big 90 degree gearbox inertia
J_p	Planetary wheel carrier inertia in son gear
J_{pc}	Inertia of planetary gear and ring in son gearbox
J_s	Inertia of son gearbox in planetary gearbox
J_1	Inertia of spring k_1
J_2	Inertia of spring k_2
J_3	Inertia of the spring k_3 and its connected shafts
J_4	Inertia of spring k_4 and pulleys k_5
n	Planetary gearbox ratio
N	Test gearbox ratio
k_1	k_1 torsional stiffness
k_2	k_2 torsional stiffness
k_3	k_3 torsional stiffness
k_4	k_4 torsional stiffness
k_5	k_5 torsional stiffness
k_6	k_6 torsional stiffness
k_7	k_7 torsional stiffness
k_{eq}	Equivalent hardness

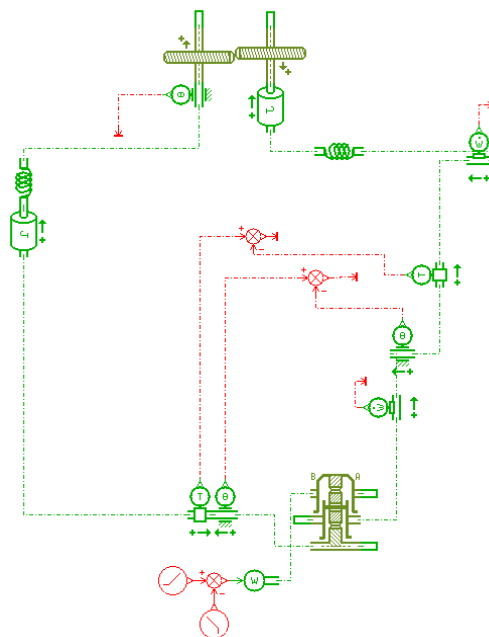


Fig. 8. Model for equivalent hardness validation

3.4. System transfer function

Torque control in presence of any kind of distraction in the closed loop is our finale goal. In the best procedure we should change the speed of drive motor. Here we consider that changing the motor speed causes some distraction in torque value. Therefore we should eliminate this distraction by changing the angle in gear ring. For controlling the value of the torque in the mechanically closed loop systems we need system transfer function for controlling it with the proper controller. Our transfer function consists of four parts. They are valve, controller, hydraulic jacks and planetary gearbox. In torque induction system two hydraulic actuators are used for rotating the planetary gear ring. As is shown in figure 9 motor cycles are equal to first shaft cycle. So, first should reach to proportional relation between point angles to basis point one

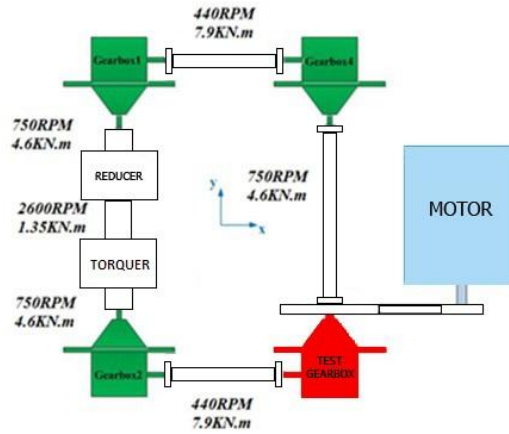


Fig. 9. System component schematic

According to proportional ratio of gearboxes between these two points we have:

$$\theta_m = \theta_1 = \theta_3$$

$$\theta_4 = 3.4 \times \theta_3$$

(4)

$$\ddot{\theta}_m = 3.4 \times \ddot{\theta}_m$$

Where $\ddot{\theta}_m$ basis point speed is changing rate and θ_m is motor speed change. In this test at the beginning torque is considered equal to the set point. Then torque is start changing by motor speed changes and with respect to drive resistant to motor sudden speed changes and keeps it in the specific rate. These changes in torque values should be controlled by controller. Resistance torque with respect to rotational speed change rate is reachable by having the entire system rotary inertia proportional to basis point. This inertia according to motor speed change is equal to $J_{eq} \cdot \ddot{\theta}_m$. The current control valve transfer function according to MOOG company control valve catalogue is equals to:

$$\frac{Q}{I}(s) = \frac{k_{SV}}{1 + 2\xi\left(\frac{s}{\omega_n}\right) + \left(\frac{s}{\omega_n}\right)^2} \quad (5)$$

Where k , ω_n , ξ are valve static gain, natural frequency and damping factor.

The required torque by torque induction system should be equal to torque generated by the motor speed changes rate. Moreover, maximum stroke length of jacks is very important and should not be exceeded. Induced torque is corresponds with the amount of jacks stroke.

Actually, input is the current control valve flow and output is the induced torque from torque induction system. What's more, the torque caused by motor cycle change rate could be considered as set point. System transfer function which is the proportional between inputs and outputs (flow, torque) could be reached as follows [9]:

$$V = \int Q dt \tag{6}$$

$$\Delta T_{torquer} = 2 \int i(k_{sv} \frac{\omega_n}{\sqrt{(1-\xi^2)}}) e^{-\zeta \omega_n t}.$$

$$\sin((\sqrt{1-\xi^2}) \omega_n t) dt) \cdot ((n-1)k_{eq}/AD) =$$

$$2 \cdot i(k_{sv} \frac{\omega_n}{\sqrt{(1-\xi^2)}})$$

$$\left(\frac{e^{-\zeta \omega_n t} (-\xi \omega_n \sin((\sqrt{1-\xi^2}) \omega_n t) - (\sqrt{1-\xi^2}) \omega_n \cos((\sqrt{1-\xi^2}) \omega_n t))}{(\xi \omega_n)^2 + (1-\xi^2) \omega_n^2} \right)$$

$$\cdot ((n-1)k_{eq}/AD) =$$

$$\frac{T}{I} = 2 \cdot k_{sv} \cdot \omega_n \cdot e^{-\zeta \omega_n t}$$

$$\left(\frac{e^{-\zeta \omega_n t} (-\xi \omega_n \sin((\sqrt{1-\xi^2}) \omega_n t) - (\sqrt{1-\xi^2}) \omega_n \cos((\sqrt{1-\xi^2}) \omega_n t))}{(\xi \omega_n)^2 + (1-\xi^2) \omega_n^2} \right)$$

$$\cdot ((n-1)k_{eq}/AD)$$

(7)

Where Q_p is the pump flow rate.

Hence, we could reach to the system transfer function with proportional valve for using PID controller.

Moreover, and as we mentioned, here we want to compare the on-off and PID controller behavior on the gearbox test rig for the torque induction part of the system. So for the on-off controller the on-off valve is required. Therefore transfer function is equals to [9]:

$$G = \frac{T/t}{V/Q_p} = 2 \cdot (n - 1) k_{eq} \cdot \frac{Q_p}{AD} e^{-\zeta \omega_n t} \tag{8}$$

Where Q_p is the pump flow rate.

4. CONTROLLER DESIGN

Hydraulic systems are extremely complicated, making modeling and design of feed-back controllers difficult. Nonlinearities are created primarily by flow-pressure characteristics, orifice area openings, and so on.

4.1. On-Off

The most basic type of control device is an on/off controller. There is no intermediate state in the device's output; it is either on or off. Only when the output's value exceeds the setpoint will an on-off controller switch the output. For control, when the output is below the setpoint, it is on, and when it is above setpoint, it is off. The process will continuously cycle from below setpoint to above setpoint and back below because the output crosses the setpoint to

alter the output state. In situations where this cycling happens quickly, an on-off differential, or "hysteresis," is introduced to the controller operations to prevent damage to contactors and valves. With this differential, the output won't switch on or off until the values are a specific amount above the setpoint. If the cycling above and below the setpoint happens very quickly, on-off differential prevents the output from "chattering," or making quick, continuous shifts. When precise control is not required, on-off control is typically employed.

4.2. PID

A PID-controller has been mentioned in numerous studies as having been utilized in hydraulic position servo systems [10–13]. It is the most popular controller in many industrial applications. This controller type offers control that is proportional, integral, and derivative. With the help of two extra adjustments and proportional control, this controller enables the device to automatically account for system changes. The proportional, integral, and derivative terms must each be individually modified or "tuned" to a specific system through trial and error. These modifications, integral and derivative, are stated in time-based units.

Of the three controller kinds, it offers the most precise and steady control. It works best in systems with a relatively low mass and those that respond quickly to changes in the energy added to the process. It is advised in systems where the controller is anticipated to automatically adjust for frequent changes in the setpoint, the quantity of energy available, or the mass to be managed. In these systems, the load changes often and these changes are caused by the load.

It is a highly natural solution for a type 0 system's controller (velocity control, temperature control, pressure control). A position servo system, on the other hand, is a type 1 system, which means that an integrator is already present in the forward loop (actuator, cylinder, or motor). The nonlinearities of valves, load, and friction forces are the most problematic. Nonlinearities in position servo systems, according to linear theory of hydraulic servo systems, create position error and are accountable for performance constraints. Without a doubt, the plant as a whole presents a very difficult dynamic to regulate.

In the recent decades linear control is used for hydraulic cylinder control. Moreover, new nonlinear methods for hydraulic cylinder control are investigated. In single rod hydraulic drive we cannot combine the differential pressure equation in two chambers inside correspondence equation with pressure [15]. Therefore, it makes some complexity in control tasks in comparison with double rod cylinders.

In two rod cylinders due to pressure associated with power and torque, this parameter can be used to control torque. Represented controller with this method should be designed in order to controls uncertain parameters and nonlinear behavior of the system. In closed loop torque induction systems to enhance productivity demand for high speed performance is very high. In many cases the result of high speed is achievable by using servo valve with hydraulic drives in pressure control. In such systems the main aim is designing a control which could cope with nonlinear behavior and sudden parameter changes and produce the quick response with low overshoot and error with acceptable boundary value error.

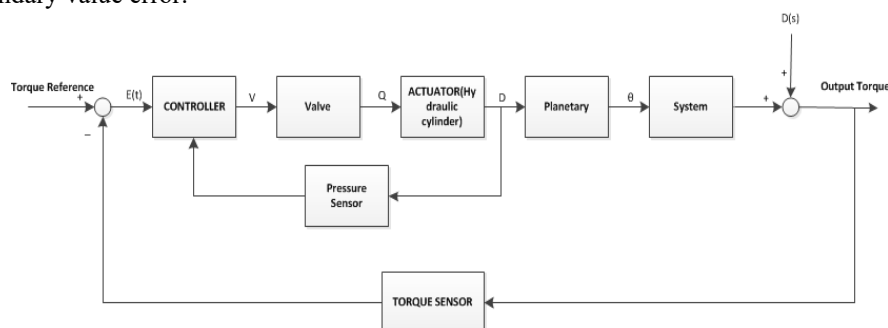


Fig. 10. Torque control loop with cylinder pressure feedback

Here, we want to design controller which controls hydraulic actuator position in order to stabilize the torque. At the beginning an on –off controller is used. Then a PID controller is used which improves the system output and precision respectively.

Therefore, in order to reduce errors valve cylinder is used. In figure 11 the control loop with hydraulic pressure feedback and torque is shown.

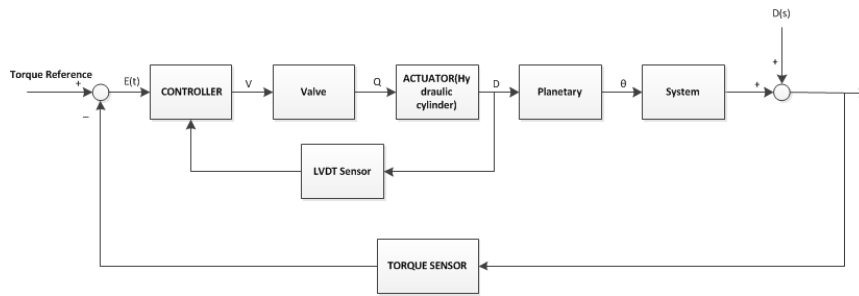


Fig. 11. Torque control loop with cylinder pressure feedback

As we mentioned here we also compare the result of the controller effort in the presence of the noise in the entrance of the control input signal to the plant. In figure 12 this block diagram is shown.

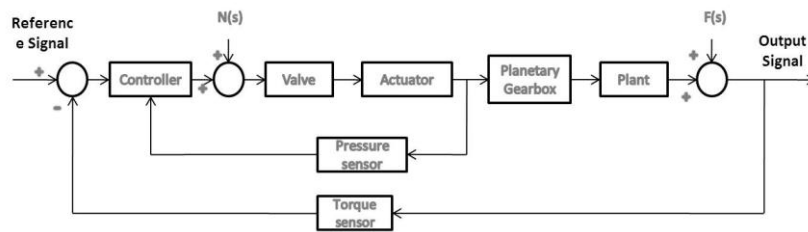
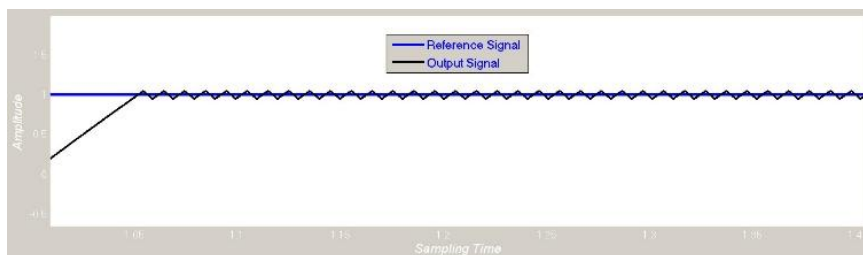


Fig. 12. System block Diagram with Noise in Signal Input

5. SIMULATION RESULTS

To test the feasibility of the developed controller, co-simulation based on AMESim and Matlab/Simulink was carried out. As discussed, in order to verify model parameters, the load simulator model was built in AMESim and developed controller was achieved in Matlab. In figure 13 the model response into the different inputs for on-off controller is shown. The result of PID Type I controller also is shown in figure 14. As illustrated in figures 14 and 15 test rig system improves in the second method. Moreover, all the PID factors are reached by try and error methods for evaluation each procedure. What's more, Figure 13, 14 and 15 show the optimization progress for each iteration and found an optimal solution within the specified tolerances. Steady state error and peak overshoot have been reduced to zero, settling time and rise time as have also been reduced in PID controller technique. In table 2 the results is shown.

The simulation results for PID Type II controllers in the presence of noise signal in the control input entrance in to the plant, shows its ability to control the system output even in such difficult situation. Therefore, the designed test rig could control the situation even in the presence of any different kind of distraction.



(a)

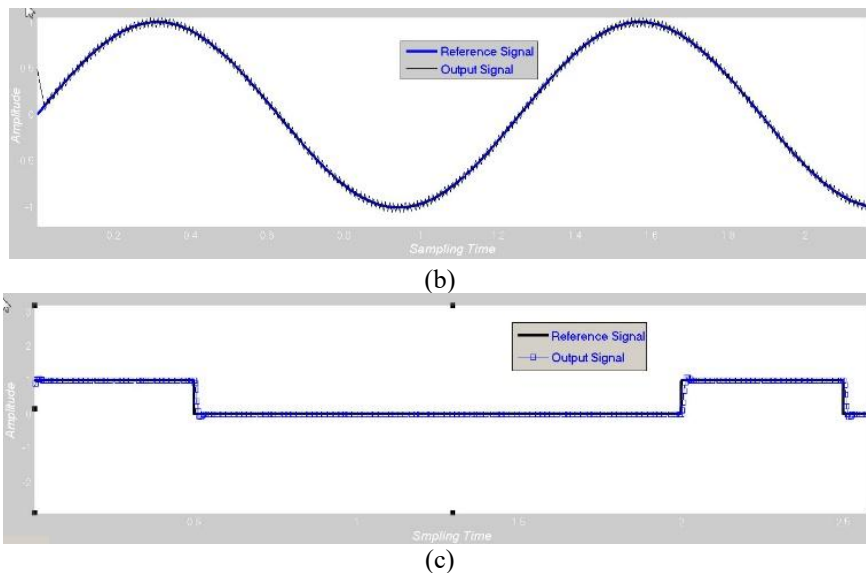


Fig. 14. Test rig with PID Type I controller-a) Step Reference Signal b) Sine Reference Signal c) Pulse Reference Signal

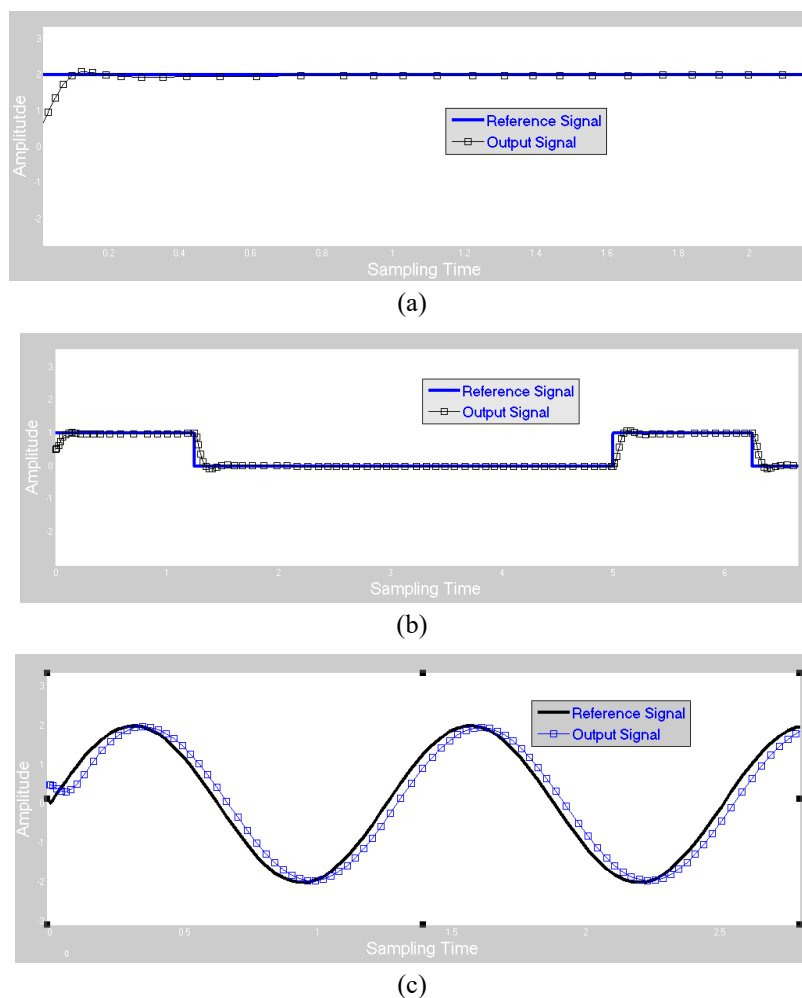


Fig. 15. Test rig with PID Type II controller and noise in signal input -a) Step Reference Signal b) Sine Reference Signal c) Pulse Reference Signal

Table 2. Rise Time and Settling Time of the Controller Types

	Type I	Type II
Rise Time	0.02	0.1
Settling Time	0.03	0.15

As is shown critical parameters could have been adjusted by the PID factors. Based on the error analysis, control effort and observation on the tracking performance, the PID control provides more convenient and better performance in position tracking control. What's more, this test rig is built by our group and is under the test in the laboratory. In later papers we will discuss about its achievements in the helicopter gearbox test. In figure 16 the built test rig is shown.

**Fig. 16.** Built test rig

6. CONCLUSION

In this paper a mechanically closed loop is used for testing a gearbox. The torque induction element was a planetary gearbox which induced torque in the closed loop by changing into of its ring angle. Two different ways for controlling this planetary gearbox is compared in this paper (On-Off Controller and PID controller). Results show that PID controller is more accurate, and accelerated in compare with On-Off controller. For this purpose two kinds of hydraulic actuators are used and their model are derived and verified by the software. Beside, the simulation results show that the noise entrance in different parts of the test rig could be controlled by PID controller. Moreover, the control parameter in these two methods was hydraulic jack's stroke. By this the controller could control the pressure behind the hydraulic jack's piston.

CONFLICTS OF INTEREST

The authors declare no conflict of interest.

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