Development of an Auxiliary Engagement System for Earth Moving Equipment Hydromechanical Transmission

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Abstract:

The scope of this paper is to study the failure of hydro-mechanical transmission in earth moving equipments and introducing an auxiliary system to overcome the transmission disengagement on working sites. The main aim of the auxiliary system is to get the transmission in gear in order to get the machine to move and avoid the crane cost lift the machine for maintenance. A 936F caterpillar wheel loader is used as a case study where mostly for the hydromechanical transmission under study; the speed selector spool is failure in the transmission hydraulic controls causes the loader cannot move. So an auxiliary control module to engaged No.6 clutch has been presented and have first speed forward only to rescue the equipments. In the present work, a mathematical model has been developed to simulate the electrohydraulic control system for an auxiliary control module to engage the first speed. An auxiliary module consists of an electrohydraulic directional control valve that used to engage the first speed by actuated the clutch engagement piston. Experimental setup has been conducted to study the dynamic behavior of auxiliary controller on the engage and disengage of the first speed train. The experimental test rig included an auxiliary pump as well as directional control valve where the transmission input and output speeds are monitored as well as the actuating pressure. The results of both simulation and experimental test shows that the purposed auxiliary system is able to engage the transmission into first gear in case of main system pressure leakage. Mathematical model has been proposed for the electrohydraulic control system that used to be the auxiliary engagement system.

Key words: Hydromechanical transmission, electrohydraulic control valve, directional control valve, auxiliary engagement system.

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1. INTRODUCTION

The transmission is a mechanical component designed to transmit power from the engine to the drive axle, by varying the gear ratio, the transmission alters the levels of power and speed to the wheels. The transmission under study is a hydromechanical transmission where the power from diesel engine is sent from the flywheel to torque converter. The torque converter output gear is connected to the transmission input gear. Six hydraulically activated clutches in the transmission, give four forward speeds and four reverse speeds. Speed and direction selections are made manually. The transmission output shaft is connected to a gear in the output transfer gear case by splines. Power is sent through the gear to the rear drive pinion and the front drive pinion. The pinions, bevel gears and gears of each differential turn their respective axles. The axles are connected to final drives which turn the wheels.
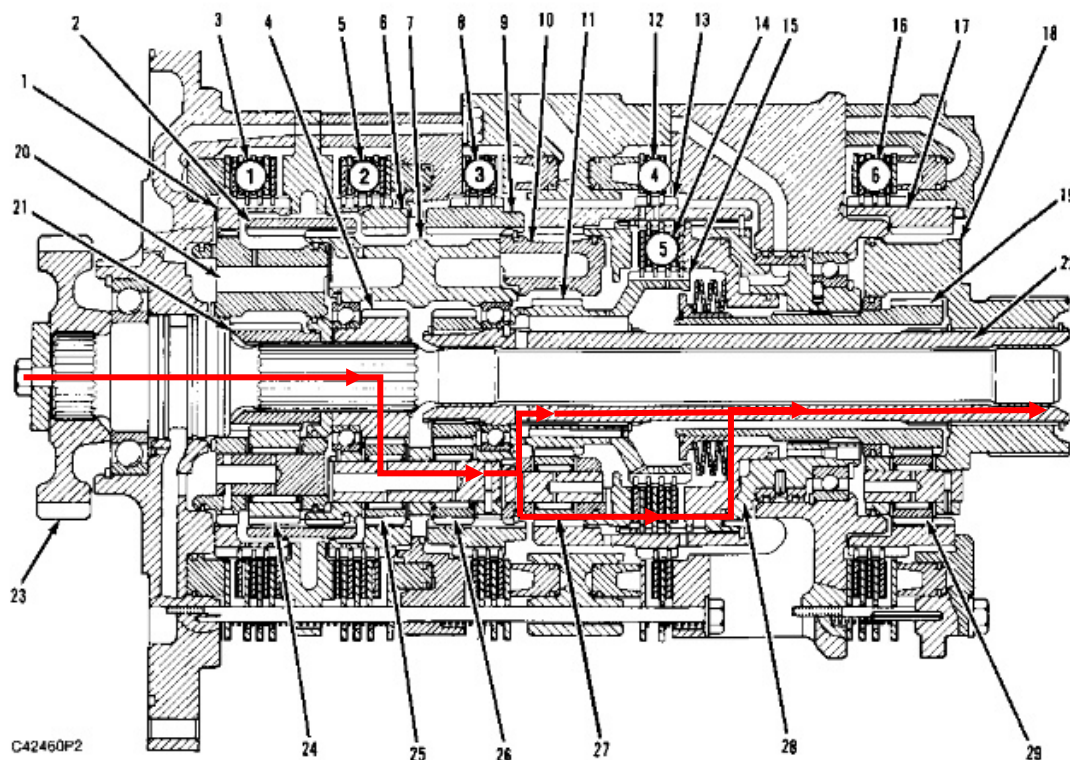

Fig. 1. Caterpillar 936F loader wheel hydromechanical transmission [7]
(1) Ring gear for No. 1 clutch. (2) Coupling gear. (3) No. 1 clutch. (4) No. 2 sun gear. (5) No. 2 clutch. (6) Ring gear for No. 2 clutch. (7) No. 2 and No. 3 carrier. (8) No. 3 clutch. (9) Ring gear for No. 3 clutch. (10) No. 4 carrier. (11) No. 4 sun gear. (12) No. 4 clutch. (13) Ring gear for No. 4 clutch. (14) No. 5 clutch. (15) Rotating hub [part of sun gear (11)]. (16) No. 6 clutch. (17) Ring gear for No. 6 clutch. (18) No. 6 carrier. (19) No. 6 sun gear. (20) No. 1 carrier. (21) No. 1 sun gear. (22) Output shaft. (23) Input gear and shaft. (24) No. 1 planetary gears. (25) No. 2 planetary gears. (26) No. 3 planetary gears. (27) No. 4 planetary gears. (28) Housing assembly. (29) Ring gear for No. 6 clutch.

For the transmission is in first speed forward, as shown in Fig. (1), No. 6 and No. 2 clutches are engaged by the transmission hydraulic control where No. 6 clutch for first speed and No. 2 clutch for forward direction. Mostly the speed selector spool is failure in the transmission hydraulic controls causes the loader cannot move.

Hydromechanical transmission (HMT) is a good area of research. It is divided into three main elements (torque convertor, gear train and hydraulic control system). Many researchers study the effect of torque convertor on the performance of HMT. As in 2001 Metwally M. [8] discusses the effect of unidirectional clutch bearing design on the performance of HMT. Also the effect of the stator blade design on the performance of stator and its influence on the HMT performance

In this work the electrohydraulic control system has been discussed from the usage as a auxiliary controller point of view. Many researchers are studies the electrohydraulic control system from different areas. In 2001, Elmayyah [9] Problem of two speeds meshed simultaneously, in mobile, has been investigated. Modeling of hydraulic transmission has been done to study the behavior of internal physical parameters considering their effect on the system behavior. In 2001, Aik [10] studied new proportional assist ventilation (PAV) method using a proportional solenoid valve (PSV) to control air supply to patients suffering from respiratory disabilities. The outlet flow and pressure from the proportional solenoid valve at various air supply pressures were tested and proven to be suitable for pressure and flow control in a PAV system. In 2001, Ke Li [10] presented the design of an electro-hydraulic system which consists of three flow-control proportional valves. The speed regulation of the cap and the synchronization control of the two cylinders were also presented. In 2005, Dasgupta [12] analyzed the dynamic characteristics of a proportional solenoid controlled piloted relief valve through bond-graph method. In 1995, Cheung [2], [3], [4], [5] and [6] converted a low cost, highly nonlinear switching solenoid into a proportional actuator through intelligent control. In 1991, W. T. Adam [1] developed a strategy to control poppet position for proportional solenoid valve. Valve response was measured experimentally. The results were used to develop a simplified model of the position response as a function of current.

Mostly for the hydromechanical transmission under study, the speed selector spool is failure in the transmission hydraulic controls causes the loader cannot move. So an auxiliary control module to engaged No.6 clutch has been presented and have first speed forward only to rescue the equipments. In the present work, a mathematical model has been developed to simulate the electrohydraulic control system for an auxiliary control module to engage the first speed. An auxiliary module consists of an electrohydraulic directional control valve that used to engage the first speed by actuated the clutch engagement piston. Experimental setup has been conducted to study the dynamic behavior of auxiliary controller on the engage and disengage of the first speed train.

2. EXPERIMENTAL WORK

2.1 Introduction

This paper presents the test rig construction, the testing procedures and samples of the recorded data. The hydromechanical transmission performance could be evaluated by measuring the pressure of different parts such as input port, speed and direction clutch. Experimental tests have been applied to a hydrodynamic one set transmission clutch of a loader (caterpillar 936F).

Two groups of experiments have been done. The first was the Preliminary experiments and the second have concentrated on measuring the clutching and declutching process. A test rig was constructed to avoid the disadvantages of the loader experimental work.

Which may be summarized as?

1- It was not available all the time to have same loader or to have another loader with the same technical conditions to continue the experiments.
2- It's so difficult to control the driver foot to have constant drive speed or maintain the internal combustion engine running at maximum speed all the time needed to the experiment.
3- It is so difficult to modify or change any components in the working system.
4- It is difficult to move the loader while recording data so the direction spool must have only neutral position.

Because of these reasons the test rig has been built up on the test rig two groups of experiments were carried out. The first group was the preliminary experiments to explore the system behavior and verify the test rig operation.

The second group was measured after the modification in the test rig.

2.2 **Test Rig Construction**

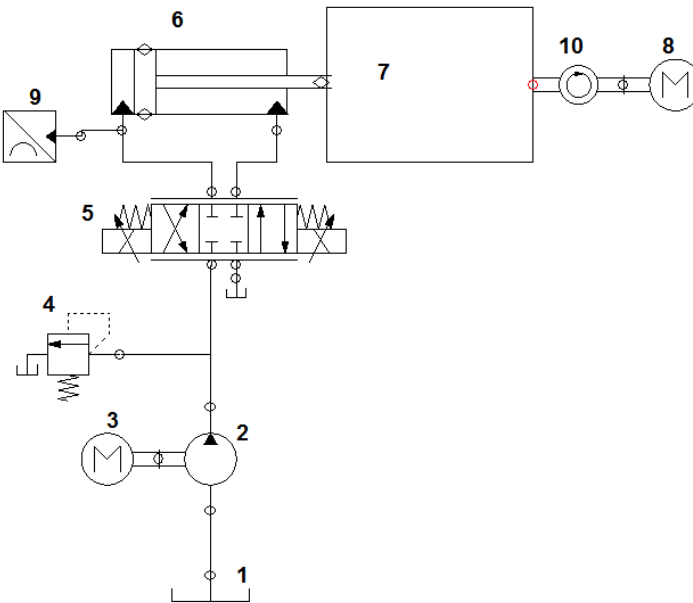The proposed test rig consists of the components as shown in Fig. (2)

2.3 Operating System

Fig. 2. Operating system in test rig

1- Tank
2- Hydraulic pump
3- Electrical motor (1- phase)
4- Relief valve
5- 4/3 direction Control valve

6- Linear actuator
7- One set hydraulic transmission clutch.
8- Electrical motor (3 phase)
9- Pressure transducer
10-RPM sensor

The hydraulic pump (2) as shown in Fig. (3) is a gear pump. It's instead of hydraulic transmission pump of a loader. It's coupled to electric motor by belt the front element of the pump supplies oil to the transmission control valve the rear element takes oil from the oil reservoir. The electric driving motor (3) that shown in Fig. (4) is a one phase A/C motor . it was used to drive the hydraulic pump (2) instead of the fly wheel of the torque converter on the test loader where it's turn by internal combustion engine. Relief valve (4) that shown in Fig. (5) Adjusted at 21 bar to return the exceed oil to the tank. It's connecting between the hydraulic pump (2) and direction control valve (6). It's instead of a relief valve in transmission control box of a loader
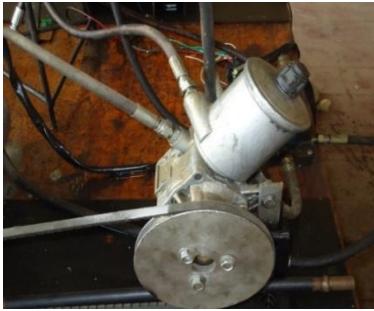
Fig. 3. Hydraulic pump

Fig. 4. One Phase A/C electric motor

. One set hydraulic transmission clutch (7) that shown in Fig. (6), Fig. (7), and Fig. (8) Consists of disks, plates, planetary gear and piston. It's a part of planetary transmission of loader 936F caterpillar.
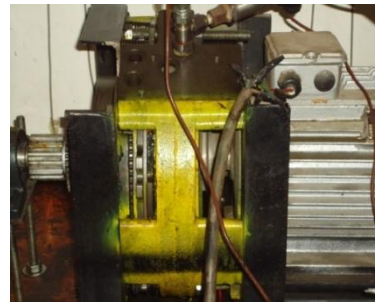

Fig. 5. Relief valve

Fig. 6. One set hydraulic transmission clutch

We choose that set specially because the planetary transmission of that loader is more expensive and it's only has all ports to engage and disengage all speeds first, second, third and forth by the transmission control box which it's fixed over it. So we can engage and disengaged the first speed by a new simple direction control valve. The electric driving motor (8) that shown in Fig. (9) Is a 3 phase A/C motor. It was used to drive the carrier of planetary gear set. The electric driving motor (8) that shown in Fig. (9) Is a 3 phase A/C motor. It was used to drive the carrier of planetary gear set. The control valve (4/3) that shown in Fig. (10) Control the clutching and declutching the first speed by an electric switch. It's instead of the transmission control box of a loader. Purposed use it when the transmission control box is failure.

Fig. 7. One set hydraulic transmission clutch

Fig.8. One set hydraulic transmission clutch

Fig. 9. Three phase A/C motor

Fig. 10. 4/3 direction control valve.

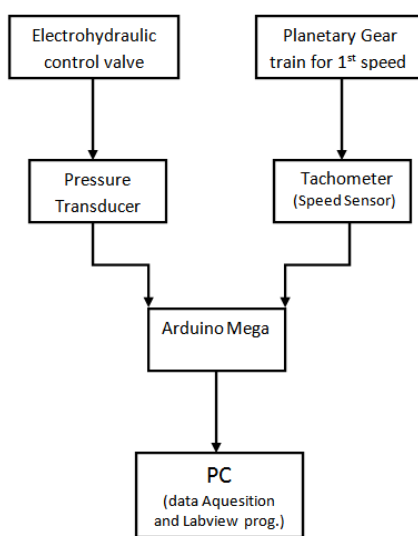## 2.4. Measuring Devices

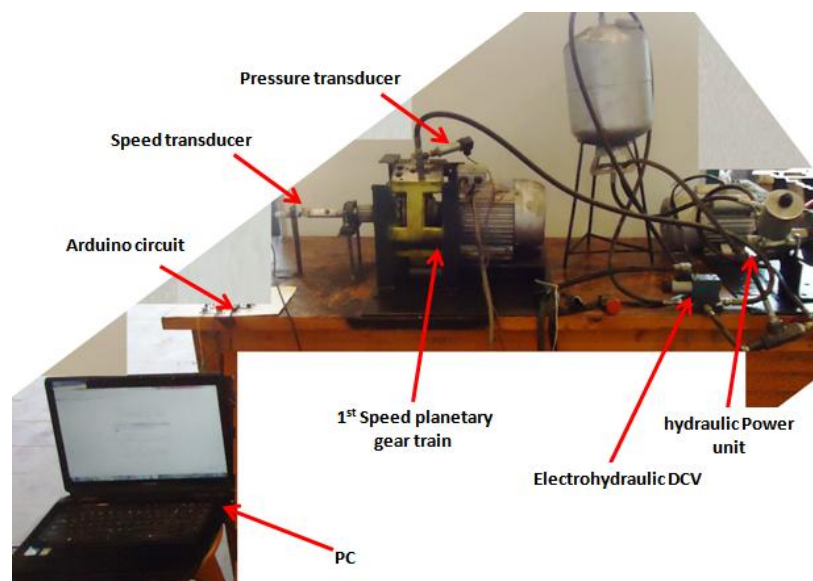Fig. 11. Measuring devices

Fig. 12. Experimental test rig

Experimental test rig that used to measure the behavior of an auxiliary control unit is shown in Fig. (12).

2.5. Experimental Measurements

The first step to have accurate measurement is to calibrate the measuring devices. The software used to collect and save data is a program made by Labview NI. Before measuring check has to be done by running the electric motors for 10 sec

The following steps should be followed to assure accurate collecting and recording of measured data.

1-before measuring and recording data the electric motor has to be started and run for minimum 15 minute as the system reaches steady state

2- Control valve in neutral

3-the plug of the pressure to be measured and recorded has to be connected to the pressure transducer.

4-measuring beings by running the program using the personal computer these steps have been repeated to measure and record all pressure in the system

2.7 Test Rig Experimental Results

The test rig allows some modification in the system. The tests applied on the test rig were 2 groups of tests. The first in primary tests to estimate the system behavior and verify the test rig behavior. The second group has been concentrated on recording the pressure of the clutching and declutching process and speed of output shaft.

Figure (13) present the first speed clutch pressure. Pressure builds up from zero bar to 15.5 bar in 2.7 seconds. The dynamic behavior of first gear train speed at engagement is shown in Fig. (14).
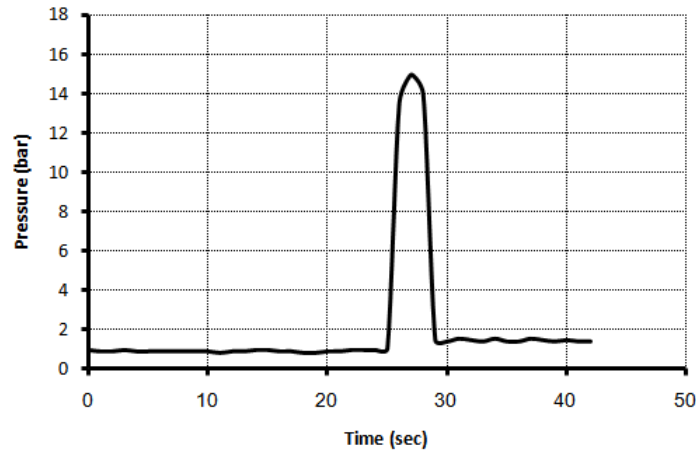

Fig.   13.   Dynamic response of the engaged pressure of the first speed by electrohydraulic control valve.
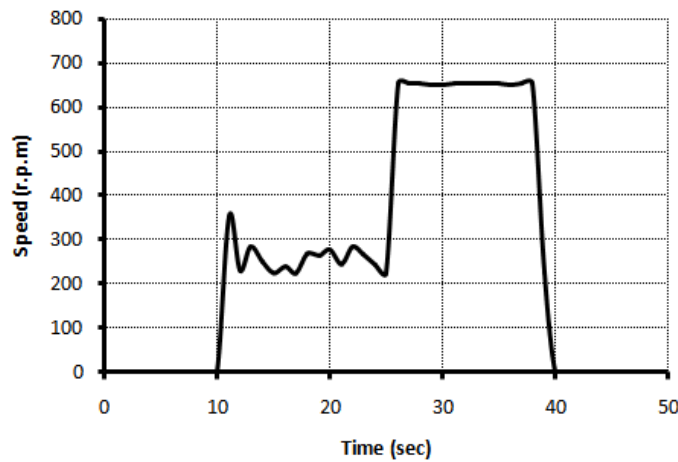

Fig.   14.   Dynamic response of the speed of the first gear train engaged by electrohydraulic control valve.
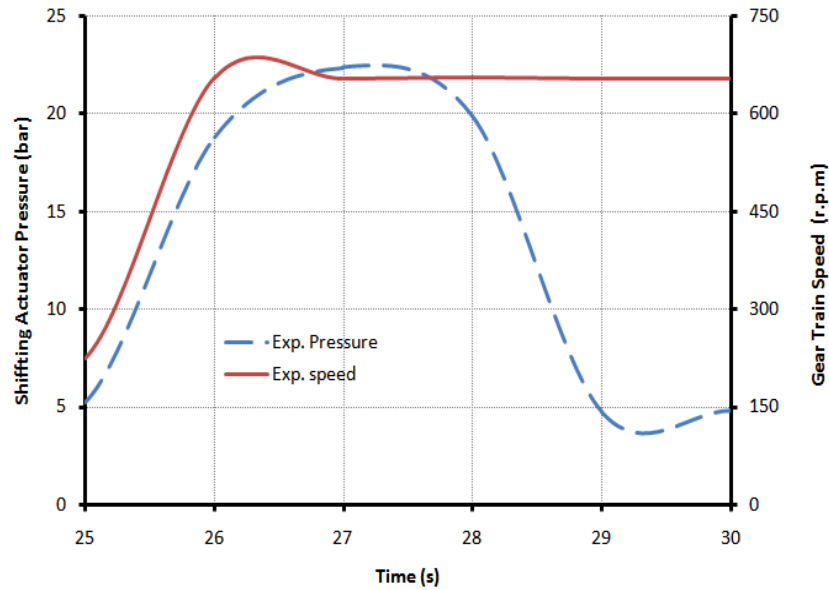
Fig. 15. Experimentally Dynamic response of the first speed by auxiliary electrohydraulic control valve.

3. Mathematical Model

The mathematical model has been developed for electrohydraulic directional control valve and the hydraulic actuator as shown in Fig. (16). Both represent the hydraulic engage and disengaged system for the auxiliary control system. That system is responsible to engage the first gear train of the first speed in order to rescue the equipment from the field or to withstand to climb the trailer to send to the repair area.
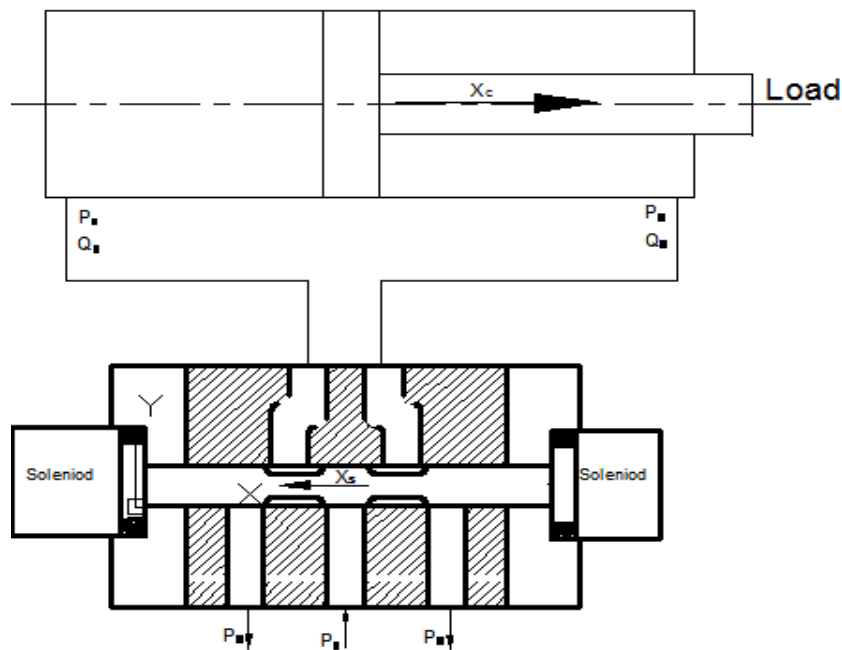

Fig. 16. Schematic diagram of the auxiliary control system (Electrohydraulic directional control valve) used to engage the first gear train.

3.1 Forces Acting on The Solenoid

In the steady state, the core displacement has been determined by considering the solenoid attraction force on core (FM), spring forces (F). Thus, forces acting on the solenoid core could be written as follows:

$$F_M = CANi \, / \, 1 \, , \qquad F = K_c \, x_c \qquad \qquad (1)$$

Where:

F_M….. Magnetic force, N

F…… Spring force, N

A…... Cross-sectional area of the coil, m^2

l……. Armature length, m

C…... Capacitance, F

K_c….. Spring stiffness, N/m

x_c…... Solenoid core displacement, m

Neglecting the effect of magnetic hysteresis, the equation of motion of the solenoid core can be described by the following equations

$$F = m_s \frac{d^2 x_s}{dt^2} + f_s \frac{dx_s}{dt} + F_{seat} \qquad \qquad (2)$$

Where:

$$F_{seat} = \begin{cases} 0 & \text{For } |x_s - x_c| > 0 \\ K_{seat}|x_s - x_c| - f_{seat}(\dot{x}_c + \dot{x}_s) & \text{For } |x_s - x_c| \leq 0 \end{cases} \qquad (3)$$

The mathematical model describing the dynamic response of the electrohydraulic directional control valve (EHDCV) has been presented by equations (1) through (3). For the dynamic response of the hydraulic actuator, the flow rates through the valve restriction areas are presented below under the assumption that the effect of transmission lines connecting the valve to the symmetrical cylinder is neglected.

3.2 Flow Equations:

$$Q_a = C_d A_a \sqrt{\frac{2}{\rho}(P_x - P_0)} \qquad \qquad (4)$$

$$Q_b = C_d A_b \sqrt{\frac{2}{\rho}(P_1 - P_x)} \qquad \qquad (5)$$

$$Q_c = C_d A_c \sqrt{\frac{2}{\rho}(P_1 - P_B)} \qquad \qquad (6)$$

$$Q_d = C_d A_d \sqrt{\frac{2}{\rho}(P_B - P_0)} \qquad \qquad (7)$$

$$Q_{th} = C_d A_{th} \sqrt{\frac{2}{\rho}(P_A - P_x)}$$

(8)

Where :

$$\begin{cases} A_b = A_d = A_{rc} + A(x) = \pi(D_s - C)\sqrt{C^2 + X^2} \\ A_a = A_c = A_{rc} = \pi(D_s - C).C \end{cases}$$

$$\begin{cases} A_a = A_c = A_{rc} + A(x) = \pi(D_s - C)\sqrt{C^2 + X^2} \\ A_b = A_d = A_{rc} = \pi(D_s - C).C \end{cases}$$

3.3 Continuity equations:

Applying the continuity equation to the cylinder chambers, considering the internal leakage and neglecting the external leakage, the following equations were obtained:

$$Q_b - Q_a + Q_A - \frac{V_x}{B}\frac{dP_x}{dt} = 0$$

(9)

$$Q_c - Q_d - A_B \frac{dy}{dt} + \frac{(P_A - P_B)}{R_i} - \frac{V_B + A_B y}{B}\frac{dP_B}{dt} = 0$$

(10)

$$A_A \frac{dy}{dt} - Q_A - \frac{V_A + A_A y}{B}\frac{dP_A}{dt} = 0$$

(11)

3.4 Actuator Equation of Motion:

The pressure force drives the piston. The motion of the piston is under the action of pressure, viscous friction, inertia and external forces. It could be described by the equation:

$$P_B A_B - P_A A_A + F_L = m_P \frac{d^2 y}{dt^2} + f_P \frac{dy}{dt} + F_{Aseat}$$

(12)

Where:

$$F_{Aseat} = \begin{cases} K_{Aseat}(y - y_{max}) + f_{Aseat}\frac{dy}{dt} & \text{for} \quad y \geq y_{max} \\ 0 & \text{for} \quad y_{min} \leq y \geq y_{max} \\ K_{Aseat}|y - y_{min}| + f_{Aseat}\frac{dy}{dt} & \text{for} \quad y \leq y_{min} \end{cases}$$

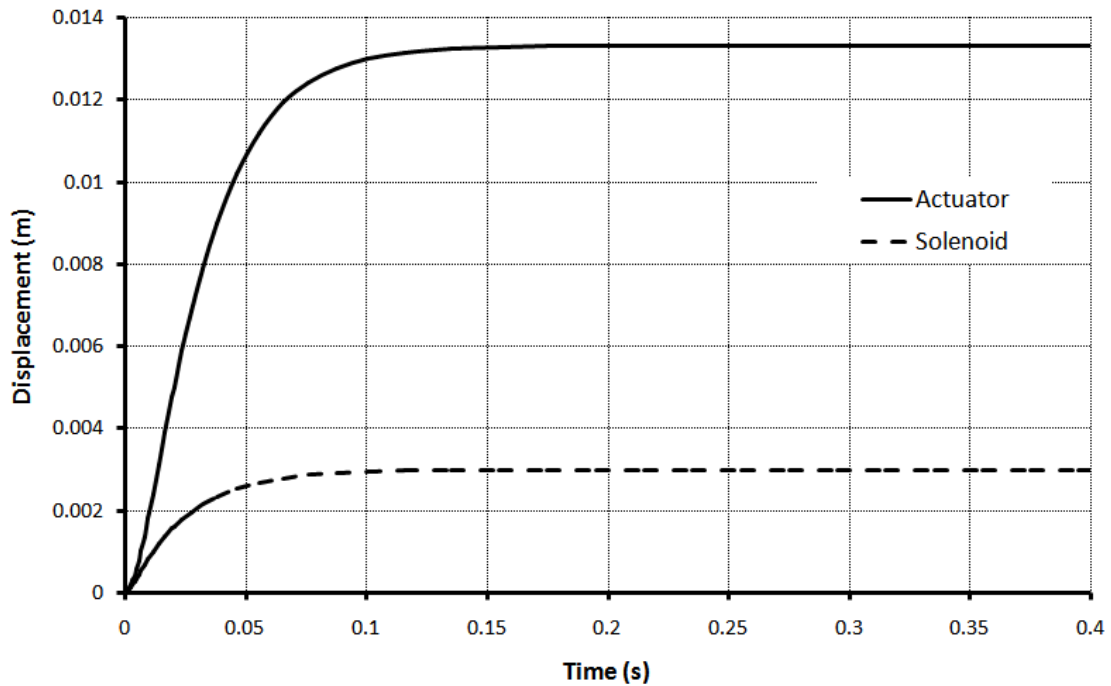## 3.5 Mathematical Model Simulation and Results.

Fig. 17. Dynamic response of the electrohydraulic control valve solenoid and actuator displacement of the first gear train engaged

Simulink matlab program has been used to simulate the electrohydraulic control system that used to engaged the first speed gear train. The simulation results have been presented in figs. (17) and (18) for the transient response of the speed and displacements of the solenoid and actuator under the step input current acting on electric solenoid.
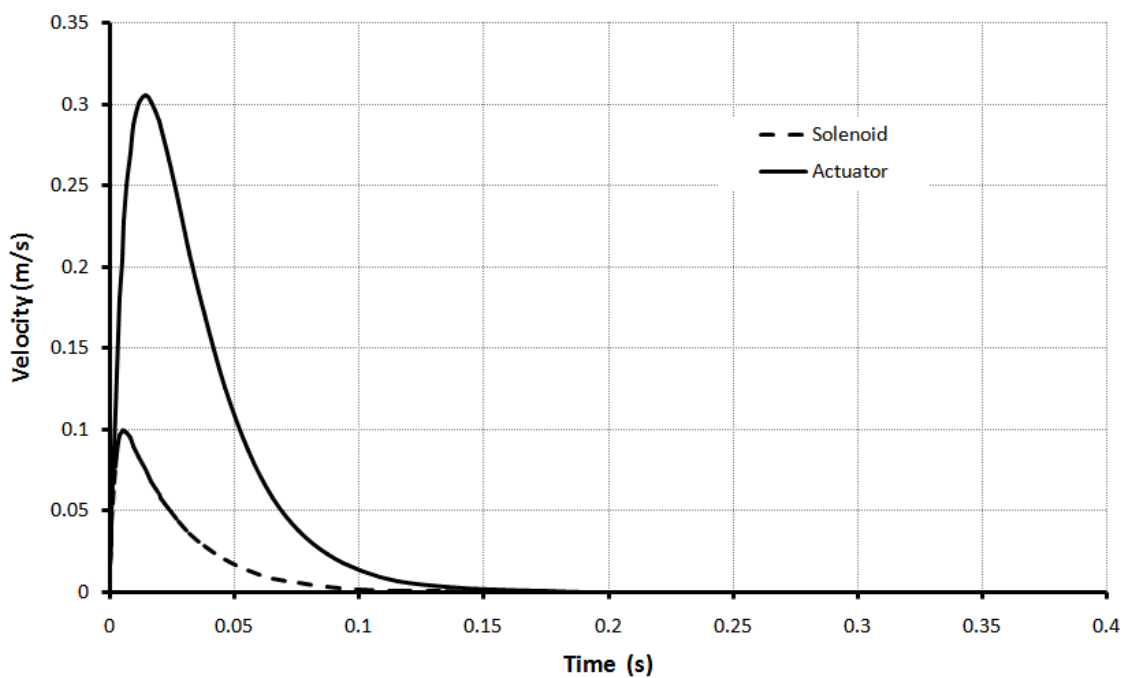

Fig. 18. Dynamic response of the electrohydraulic control valve solenoid and actuator velocity of the first gear train engaged

4. Conclusions

In this work, the failure of hydro-mechanical transmission in earth moving equipments has been discussed. An auxiliary system to overcome the transmission disengagement on working sites has been introduced experimentally and mathematically. The main aim of the auxiliary system is to get the transmission in gear in order to get the machine to move and avoid the crane cost lift the machine for maintenance. A 936F caterpillar wheel loader is used as a case study where an auxiliary control module is developed where simulation and experimental test are executed in order to validate the solution. The experimental test rig included an auxiliary pump as well as electrohydraulic directional control valve where the transmission input and output speeds are monitored as well as the actuating pressure. The results of both simulation and experimental test shows that the purposed auxiliary system is able to engage the machine into first gear in case of main system pressure leakage. Mathematical model has been proposed for the electrohydraulic control system that used to be the auxiliary engagement system. It has been concluded that it could be introduced an auxiliary control system to engage the first gear train to recover the equipment from the field to the repair area.

5. References

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