Efficiency of Direct Driven Hydraulic Drive for Non-Road Mobile Working Machines

T.A. Minav, C. Bonato, P. Sainio, M. Pietola

***Abstract* – As answer for tendencies for compact and powerful solution for electro-hydraulic systems, this paper investigates directly driven hydraulic setup for non-road mobile machinery (NRMM) application. The control of the system is implemented directly with a servo motor drive without conventional hydraulic control valves, which allows accurate control and regeneration. Speed of the double-acting cylinder is determined by in-coming oil flow from the pump, out-coming flow to the hydraulic motor and angular speed of the electric motor. The paper provides a detailed analysis of the system and an evaluation of setup usage for NRMM application. Finally, possible improvements of the suggested system are discussed.**

***Index Terms*--Drive, direct driven hydraulics, energy efficiency, energy-saving ratio, hydraulics, mobile machinery, non-road mobile machinery (NRMM), servomotor,variable speed drive control.**

I. NOMENCLATURE

E_{hyd}	Output power of hydraulic part of the system	J
E_{new_cycle}	Energy consumption of test setup	J
E_{new_down}	Recovered energy during lowering	J
E_{new_up}	Energy consumption of test setup during lifting	J
E_{pot}	Potential energy of the payload	J
E_{shaft}	Output energy of the shaft	J
E_{trad_cycle}	Energy consumption for conventional system	J
E_{trad_cycle}	Energy consumption in conventional system	J
g	Gravitational constant	m/s^2
H	Position of cylinder's piston	m
i_a, i_b, i_c	Phase current	A
m	Mass of payload	kg
p	System Pressure	Pa
P_{hydr}	Output energy of hydraulic part	W
P_{shaf}	Output energy of shaft	W
T	Motor torque	Nm
u_a, u_b, u_c	Phase voltage	V
V_c	Velocity of cylinder piston	m/s
Γ_s	Energy-saving ratio	-
η_{up_tot}	Total efficiency of the system during lifting	%
η_{down_tot}	Total efficiency of the system during lowering	%
η_{hyd_mec}	Pump/motor efficiency	%
η_{motor}	Efficiency of the electric motor	%
η_{gen}	Efficiency of the generator	%
Ω	Angular speed	rad/sec

II. INTRODUCTION

Industry is looking for compact, efficient and powerful solutions for Non-road mobile machinery (NRMM) control and power train applications. As a result electro-hydraulic systems with motor- or pump-controlled systems are observed on the market and research areas [1-6]. These electro-hydraulic systems are becoming interested in industry by advantages of their small size-to-power ratios and the ability to produce large force and torque.

At present, the valve-controlled system is applied widely, but the most significant disadvantages of valve-controlled system are throttled pressure losses, lower efficiency and heat generation. As alternative solution, displacement-controlled or the pump-controlled electro-hydraulic servo systems directly driven by servo motor have appeared [1-6]. These solutions are overcoming the disadvantages of traditional valve-controlled system and performing specific characteristics on non-linearity and time-varying cycles. The pump-controlled system, compared with traditional system, has property of compact structure and high efficiency with advantage of speed regulation loop directly driven by servo motor [7, 8]. Some of researches have been reported in the field of construction machinery to avoid disadvantages of conventional machine [9 - 11].

In this paper, direct driven hydraulic setup (DDH) as a pump-controlled system is researched from energy point of view. The remainder of this paper is organized as follows:

Section 3 gives scheme and the principles of the system are described in detail. Section 4 shows formulas which were used for theoretical evaluation of the system. Measurement results and analysis are described in Section 5. Section 6 contains concluding remarks.

III. OVERVIEW OF TEST SETUP

Figure 1 illustrates the experimental test setup simplified

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schematics with the locations of pressure and height sensors. The setup uses a speed-controlled electric servo motor drive rotating a hydraulic pump to directly control position of cylinder. The XV-2M internal gear hydraulic pump/motor with displacement 22.8 cm^3/rev by Vivoil [12] creates a flow depending on the rotating speed of the servo motor and, at the same time, the second hydraulic pump/motor with displacement 14.4 cm^3/rev controls the amount of oil pumped out from second side of the double-acting cylinder. The oil pressure is determined by the payload. Unimotor 115U2C manufactured by Emerson Control Techniques [13] and Unidrive SP1406 drive [14] offer accurate control in test setup. A program for the electric drive controls both the electrical and hydraulic sides of the system and thus allows good controlled lifting-lowering movement at different speeds and payloads.

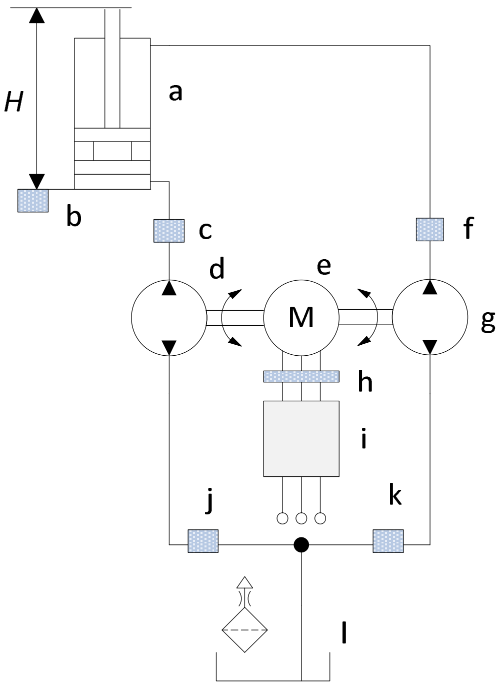

Fig. 1. Simplified schematics of test setup: a) double-acting cylinder, b) wire-actuated encoder, c) pressure sensor, d) reversible gear pump/motor_1, e) PMSM motor/generator, f) pressure sensor, g) reversible gear pump/motor_2, h) current and voltage probes, i) frequency converter, j) pressure sensor in tank line, k) pressure sensor in tank line and l) oil tank.

The Unidrive SP1406 uses an incremental encoder (4096 pulses per revolution, resolution 14 bit) as the motor rotor position feedback. Drive software was used to measure the rotating speed and torque of the PMSM. Hioki 3390 Power analyser with a sampling time of 50 µs was used for measuring the voltages, currents and active powers. The system pressures were measured by means of Gems 3100R0400S pressure transducers [15], installed at the pump's inlet and outlet.

Hydraulic cylinder C-10-60/30 x 400 manufactures by MIRO was used. The actual velocity and height of the cylinder's piston rod were measured by a wire-actuated encoder SIKO SGI (IV58M-0039) [16].

Figure 2 illustrates photograph of test setup. T-shaped gearbox was used to realize schematics in Figure 1.

Note: These components where chosen to the test set up because they were available or fast to purchase at the time of construction. They do not have any specific properties for this kind of application.

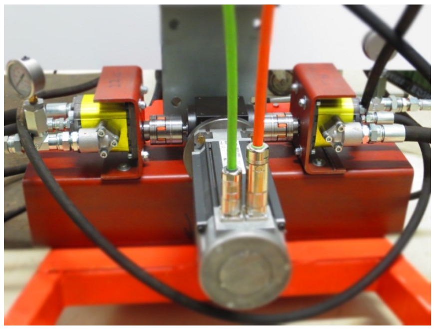

Fig.2. PMSM motor/generator connected through T-gear two internal gear motor/pumps [11].

IV. Used Equations

The total efficiencies of the system during lifting and lowering were calculated as:

$$\eta_{\mathrm{up\_tot}} = \frac{E_{\mathrm{pot}}}{E_{\mathrm{motor}}}, \quad (1)$$

$$\eta_{\mathrm{down\_tot}} = \frac{E_{\mathrm{motor}}}{E_{\mathrm{pot}}}, \quad (2)$$

where E_{pot} is the potential energy of the payload in J and E_{motor} is the input energy from the electric motor in J and calculated as the integral of the motor power. Input motor power P_{motor} in W was calculated as:

$$P_{\mathrm{motor}} = u_a i_a + u_b i_b + u_c i_c, \quad (3)$$

where u_a is phase voltage in V and i_a - phase current in A.

Potential energy of the payload E_{pot} in J was calculated as:

$$E_{\mathrm{pot}} = mgH, \quad (4)$$

where m is mass of payload in kg, g=9.81 is gravitational constant in m/s^2, H is position of cylinder's piston in m.
Electric motor and generator efficiency were calculated as:

$$\eta_{\mathrm{motor}} = \frac{E_{\mathrm{shaft}}}{E_{\mathrm{motor\_up}}}, \quad (5)$$

$$\eta_{\mathrm{gen}} = \frac{E_{\mathrm{motor\_down}}}{E_{\mathrm{shaft}}}, \quad (6)$$

where E_{shaft} is energy of the shaft in J and calculated as the integral of the power at the shaft. Output power of shaft P_{shaft} in W were calculated as:

$$P_{\mathrm{shaft}} = T\Omega, \quad (7)$$

where T is motor torque in Nm and Ω is angular speed in rad/sec.
The hydro-mechanical efficiency $\eta_{\mathrm{hyd\_mec}}$ is calculated as:

$$\eta_{\mathrm{hyd\_mec}} = \frac{E_{\mathrm{hyd}}}{E_{\mathrm{shaft}}}. \quad (8)$$

Output energy of hydraulic part E_{hydr} is calculated as the integral of output hydraulic power:

$$P_{\mathrm{hyd}} = p v_c A, \quad (9)$$

where p is pressure in Pa, v_c is velocity of cylinder piston in m/s and A is cross area of cylinder piston in m^2. To show how much energy can be saved Energy-saving ratio Γ_s for a lifting-lowering cycle was calculated:

$$\Gamma_s = \frac{E_{trad\_cycle} - E_{new\_cycle}}{E_{trad\_cycle}}, \quad (10)$$

where E_{trad_cycle} energy consumption in conventional system. For conventional system E_{trad_cycle} energy consumption is equal to E_{motor} motor energy consumption during lifting as energy consumption during lowering is assumed to be zero. Energy consumption of test setup E_{new_cycle} is calculated as:

$$E_{new\_cycle} = E_{new\_up} - E_{new\_down} = E_{motor\_up} - E_{motor\_down}, \quad (11)$$

where E_{new_down} is recovered energy during lowering and E_{new_up} is energy consumption of test setup during lifting.

Following section presents measurement results and their analysis.

V. Analysis of Measurements

Figure 3 shows an example of measured data (speed, torque, pressure and position) for motor speed 300 rpm and payload 175 kg. Measurements were carried out in room temperature. Effect of temperature was not investigated in this research.

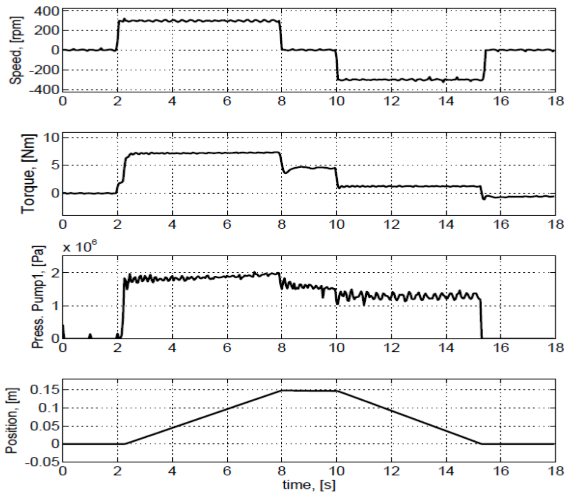

Fig. 3. Measured data: motor speed, torque, pressure in first pump/motor line, cylinder's rod position.

In Figure 3, lifting starts from 2 to 8 seconds, lowering performed from 10 to 15 seconds. During lifting motor torque is 7 Nm, during lowering torque is around 1 Nm. Pressure in pump/motor_1 during lifting is about 20 bar, during lowering is around 10 bar. For pump/motor_2 pressure is close to atmosphere pressure.

Figure 4 shows the measurement results for the total efficiency. The experimental setup was tested with payload of 175 kg at motor speed ranges from 300 to 500 rpm. Positive motor speed corresponds to lifting, negative speeds corresponds to lowering motion. As Shown in Figure 4, total lifting efficiency varies with motor speed from 48 to 20 %. During lowering, total efficiency is in the range of 8 to 32 % depending on the motor speed and payload.

Figure 5 shows the measured electric machine efficiencies. According to the test results, increasing the load from 25 to 175 kg degrades the motor efficiency by 5 per cent units at higher speeds and by 10 per cent units at lower speeds. The maximum motor efficiency was 90 % and was reached at the speed of 300 rpm with the 100 kg payload.

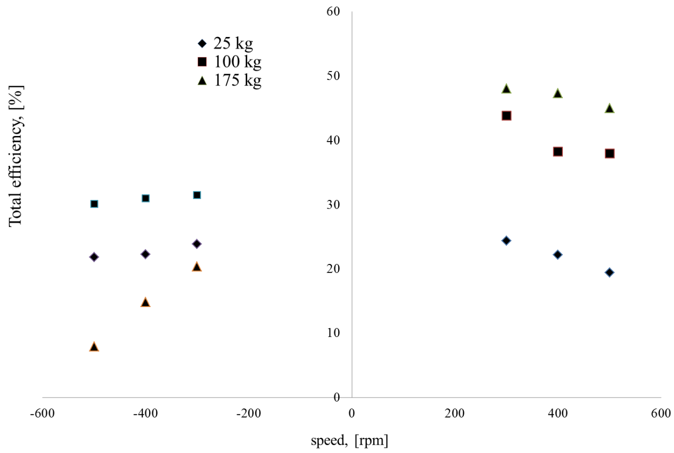

Fig. 4. Total efficiency.

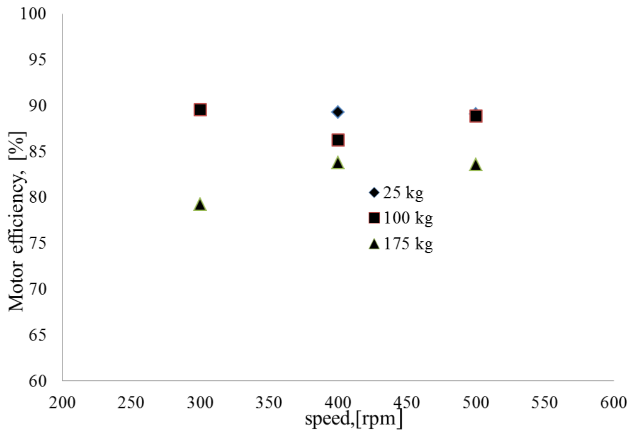

Fig. 5. Electric machine efficiency.

Equations (10) and (11) were used to calculated energy-saving ratio Γ_s for single lifting-lowering cycle example shown in Figure 3. Energy-saving ratio Γ_s is about 10 %. It shows how much energy can be saved. This value varies with change of speed and payload. Figures 6 and 7 show Sankey diagrams for measured efficiencies of the system during lifting and lowering with payload of 175 kg with motor speed 300 rpm, respectively.

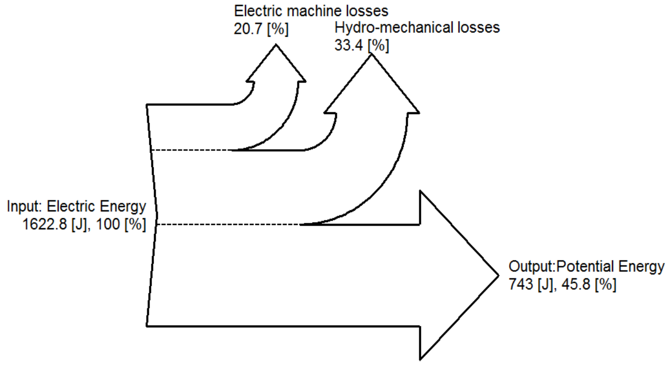

Fig. 6. Sankey diagram of the system during lifting with motor speed 300 rpm and payload of 175 kg (efficiency of the frequency converter is not included).

In Figures 6 and 7 the pump losses contain: pump/motor and t-gear, cylinder losses contain pipes and cylinder. During lifting, hydro-mechanical losses are 33.4 %. The frequency converter losses are not included in this diagram, as they were not measured. It can be observed that the hydro-mechanical losses dominate in Figures 6 and 7.

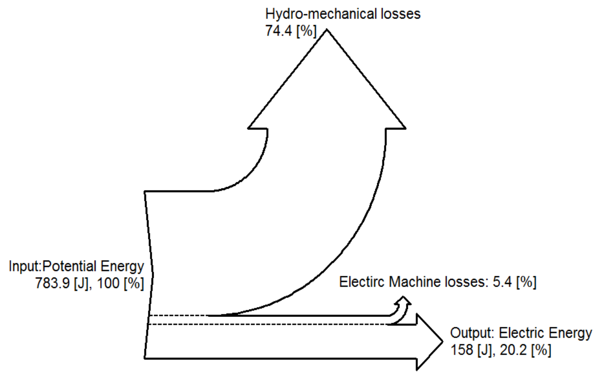

Fig.7. Sankey diagram of the system during lowering for motor speed 300 rpm and payload of the 175 kg.

During lowering, payload moves down by potential energy, its speed is controlled by opposing torque/force induced by electric motor. Figure 7 demonstrates division of losses in the system during lowering. The predominant losses are hydro-mechanical losses of 74.4 %.

VI. Conclusions

In this study efficiencies of electro-hydraulic components and the overall efficiency of the directly driven hydraulic test setup without control valves were determined. The tests showed that the direct driven hydraulic has advantages of fully self-contained electro-hydraulic actuator which combines high power density of hydraulics and accuracy of electric motor. This kind of approach although having expensive components in this research phase like high accuracy encoder and servo controller will bring along safety functions and monitoring capabilities of motor the controller to the hydraulic system. Developed motor controllers offer a lot of properties that can be used to instrument and monitor hydraulic system without additional sensors and it offers opportunities to duplicate conventional sensors based on either safety or reliability demands. Also maintainability can be estimated to achieve advantages.

The tests showed that the energy efficiency varies with the direction of cylinder's piston motion and the motor speed. Lifting efficiency varies from 48 to 20% with increasing of motor speed and varying payload, lowering efficiency – from 32 to 8 % with increasing of motor speed. Energy-saving ratio for single lifting-lowering cycle example was calculated. Calculations showed that saved energy is about 10 % and has effect on consumption reduction of the system. As Sankey diagrams showed, the hydro-mechanical losses dominate in the system. As expected, for DDH setup efficiency, the weak ring of the chain is found in the losses of the hydro-mechanical components of the system. Therefore, improvement of setup is required.

VII. Acknowledgment

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IX. Biographies

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