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ELIMINATING SIZING ERROR IN DIRECT-DRIVEN HYDRAULICS

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Abstract. This paper investegates a realization of decentralized hydraulics for industrial applications and off-road machinery by means of direct-driven hydraulics (DDH). DDH consists of two pump/motor units driven by a servomotor. These units should be dimensioned to the match expected flows of the corresponding chamber in a differential cylinder. However, the sizing error between the available pump units causes an excess pressure rise in the chambers and degrades the system efficiency. Therefore, this paper investigates elimination of the sizing error with proposed solutions utilizing hydraulic accumulator and gear selection respectively. Created Matlab/Simulink models were utilized to investigate the system performance from an energy efficiency point of view. The results showed that both of the proposed solutions are viable approaches to improve the energy efficiency of the DDH setup. The proposed hydraulic solution was implemented in the DDH setup of a crane and measurements were performed to validate the built DDH model.

Keywords: Direct-driven hydraulics (DDH), Sizing error, Energy efficiency, Hydraulic accumulator, Gear

INTRODUCTION

Since standard hydraulic architectures suffer from low efficiency and engine emission standards are increasingly stringent, more energy-efficient solutions are needed for stationary applications [1] and off-road machinery [2].
As one major loss in the hydraulic system, the key challenges to eliminate or minimize throttling loss have been addressed with a number of established methods, such as Load Sensing (LS), Independent Metering (IM), Digital Flow Control (DFC), Displacement Control (DC), Hydraulic Transformer (HT), and Constant Pressure Systems (CPS). **TABLE 1** compares these methods from the angle of throttling loss, energy recovery and investment cost [3]. According to this comparison, there are two methods with lower throttling losses, DC and HT, which can increase energy savings.

TABLE 1. Comparison of Energy Saving Methods [3].

Methods	Throttling Losses	Energy Recovery	Investment Costs
LS	Bad	Bad	Good
IM	ok	Good	Good
DC	Good	Good	Bad
DFC	ok	Good	Good
HT	Good	Good	Bad
CPS	ok	Good	Good

As can be seen in **TABLE 1,** HT, which has high theoretical efficiency, is perhaps an ideal method, but commercial units are not available [3, 4]. DC solution is more efficient than valve-controlled methods, such as LS and IM, and this has been proven in multiple research studies for off-road mobile and stationary applications [3]. The lack of acceptance today of valveless pump actuation is due to a combination of its reliance on sensors and electronic controllers, higher costs caused by the current low-volume production of the necessary components, as well as the users' unfamiliarity with the system [4].
Pump-controlled circuits (DC) have been well developed mostly for double rod cylinders [9, 10]. For differential cylinders, valves and variable displacement pumps are usually implemented for balancing the uneven flow of the two chambers.
The research on variable speed constant displacement hydraulic drives currently continues in two projects EL-Zon and IZIF, which focuses on the advancement of the decentralized or zonal hydraulics for off-road

machinery and industrial applications by means of direct-driven hydraulics (DDH). The advantages of the DDH include simplifying the system architecture (as shown in **FIGURE 1**(a)), eliminating the throttle losses, and only delivering the required amount of flow to the end actuator. Therefore, the only remaining power losses in DDH are those of the pump, the pipes, and the cylinder.

The efficiency of a DDH system may degrade as a result of unwanted pressure rise if the displacement ratio of the pump/motor units does not match the area ratio of the two chambers in the differential cylinder. However, due to the displacement ratio of the commercially available pump/motors, fully eliminating this sizing error by dimensioning cannot be guaranteed.

In [11] and [12], a solution was suggested that mitigating a sizing error larger than 4-5% is possible by implementing a hydraulic accumulator. However, this studies [11, 12] did not consider other methods to cancel the sizing effect. Therefore, the purpose of this paper is to investigate the proposed methods for eliminating the sizing error and to demonstrate their effects on the energy efficiency of a DDH setup. As shown in **FIGURE 1**, two methods are proposed for eliminating the sizing error, mechanical compensation and hydraulic compensation. The mechanical compensation is illustrated in **FIGURE 1**(b), implemented with a gear drive. **FIGURE 1**(c) illustrated the other method, hydraulic compensation, which is based on compensating the sizing error with a hydraulic accumulator.

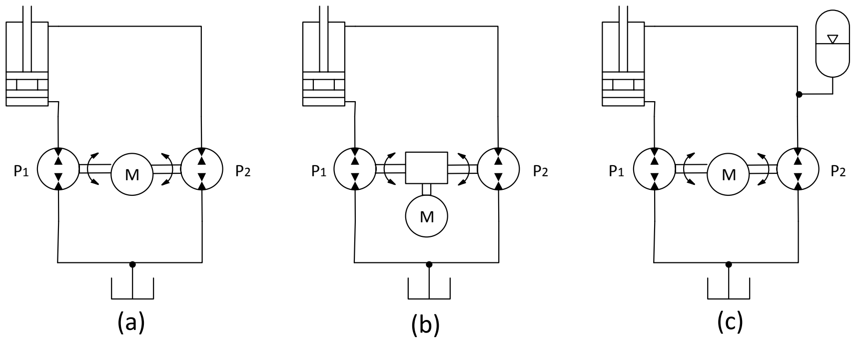


FIGURE 1. Flow Compensation Methods: (a) Ideal Setup, (b) Mechanical Compensation by Gear Drive, (c) Hydraulic Compensation by Hydraulic Accumulator.

The following section introduces a DDH test setup with a sizing error. After that, based on this DDH setup, a Matlab/Simulink model is constructed and validated by the experiment. Further, simulation studies of the proposed mechanical and hydraulic compensation methods are carried out using parameter-sweep to evaluate the effect of varying parameters on the energy efficiency of the DDH. Final sections contain a discussion of the simulation and experimental results and concluding remarks.

A CRANE SETUP WITH SIZING ERROR

This section describes the components of the DDH setup and defines the sizing error. The DDH setup is built around a single-extension mobile boom crane. **FIGURE 2** presents the prototype and the schematic diagram of the test setup. **TABLE 2** illustrates the components and sensors, including their specifications and the sizing error.

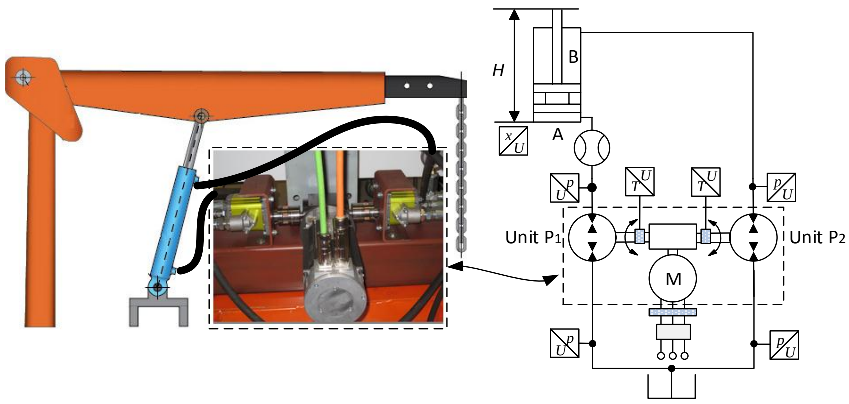


FIGURE 2. The Prototype and Schematic Diagram of the DDH Test Setup.

TABLE 2. Utilized Components.

Component			Value	Sizing Error
Servomotor Unimotor 115U2C, by Emerson Control Techniques [13]		Rated Torque	8.1 N·m	-
		Rated Speed	314.2 rad/s	-
XV-2M Internal Gear Pump/motors by Vivoil [14]	Unit P_1	Displacement	3.6 cm^3/rad	$R_{\mathrm{real}} = \frac{2.3}{3.6} \approx 0.63$
	Unit P_2		2.3 cm^3/rad	
ylinder MIRO C-10-60/30 ×400		Piston/Rod Diameter	60/30 mm	$R_{\mathrm{ideal}} = \frac{0.06^2 - 0.03^2}{0.06^2} = 0.75$
		Stoke	400 mm	
Payload		Weight	120 kg	-
GEMS 3100 Pressure Transducers [15]		Maximum Pressure	40 MPa	-
Gear Type Flow Meters by KRACHT [16]		Rated Flowrate	0.4-80 L/min	-
		Maximum Pressure	40 MPa	
SIKO SGI Wire Incremental Encoder [17]		Resolution	0.1 mm	-
NCTE 2000 Rotary Torque Sensors [18]		Rated Torque	17.5 N·m	-

The DDH was installed with internal gear motors by Vivoil [14], which can be operated in pumping and motoring mode. To avoid confusion, the motors will be referred to as the pump units from here on.
The calculation of sizing error is presented in **TABLE 2.** As illustrated in **FIGURE 2**, the flow to each cylinder chamber is dependent on the flow produced by the pump, which is determined by the servomotor speed and the displacement of the corresponding unit. As the double-acting single-rod cylinder was utilized, an ideal ratio R_{ideal}=0.75 was required for choosing the pump units. In order to study the effect of the sizing error, two pumps with the displacement of 3.6 and 2.3 cm^3/rad were chosen for P_1 and P_2 respectively. Therefore, the realized ratio R_{real} becomes approximately 0.63, which leads to sizing error as unit P_2 is 16% under-dimensioned. Thus, the fluid in the cylinder B-chamber is not pumped out at a demanded rate corresponding to that of the fluid pumped into the A-chamber while the piston is extending. This causes a pressure rise in the B-chamber, which should be mitigated. The following section investigates the effect of the sizing error on energy efficiency by simulation and measurement.

MODEL VALIDATION

In this section, a mathematical model for the DDH components will be presented and briefly introduced. As shown in **FIGURE 3**, this model consists of an electric motor, a gear drive, two external gear pumps, hose sections, a hydraulic cylinder and a crane mechanism. Further, this model was constructed in Matlab/Simulink, as illustrated in **FIGURE 4**, and the detailed explanations of the model refer to work in [11, 19].

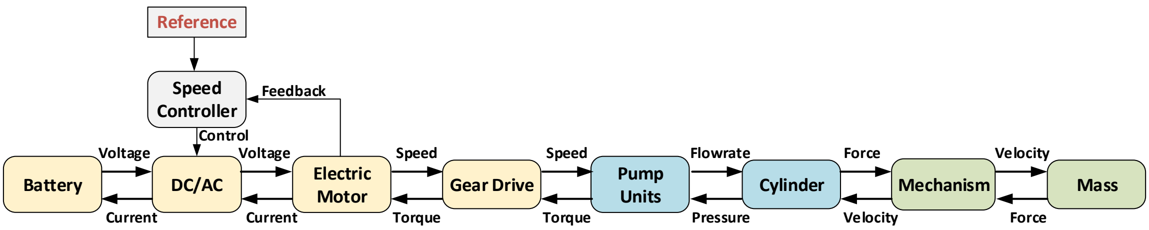


FIGURE 3. Flowchart of the Modelling and Control.

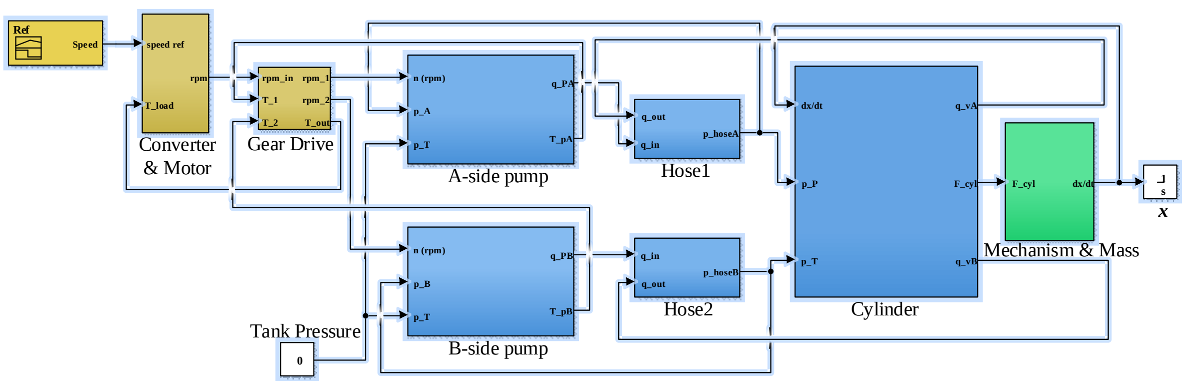


FIGURE 4. Simulink Model of the DDH Setup.

For this application, the Nykänen model was utilized to describe the compressibility of hydraulic fluid [20, 21]. The cylinder model describing the pressure is divided into two parts, one for each chamber. The friction of the cylinder was computed by utilizing LuGre model and parameters of the LuGre model were obtained from the measurements [11, 22]. Internal and external leakages of the pump were modelled with the dependency on pressure across the pump and the coefficients of the leakage model were estimated by the least square method with measured data [11]. In the model, the hydro-mechanical efficiency of the pump was set to be 85% [14].
In addition, a permanent magnet synchronous motor (PMSM) model with 95% energy efficiency was built to drive the gear box and the mechanical efficiency of gear transmission was set to be 96% based on its datasheet[23]. The mathematical model was implemented into a Matlab/Simulink environment and simulated with the ODE23s/Rosenbrock solver. In order to validate the model, earlier obtained experimental data was utilized. To make the result more comparable, the measured speed of the cycle was chosen as the reference speed for the model of this setup in this simulation. **FIGURE 5** shows the experimental data and the simulation results of one working cycle with a payload of 120 kg at an ambient temperature of 20 ℃, including the speed of the electric motor, the piston position of the cylinder and the pressure levels of the both chambers.

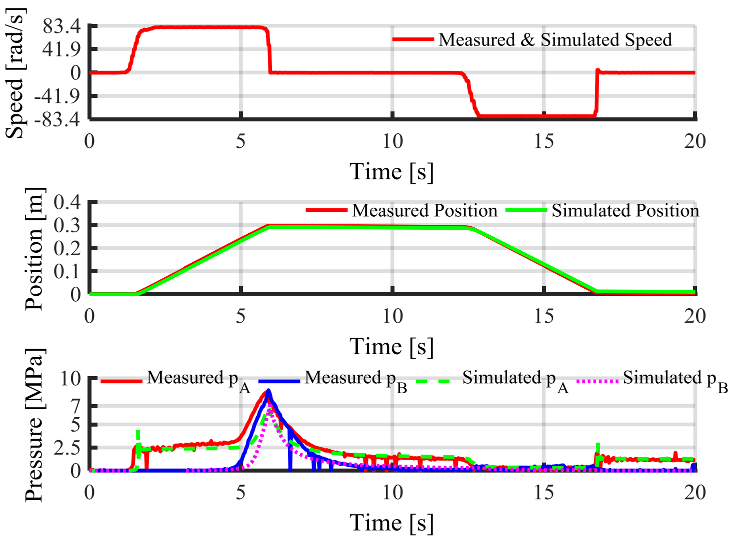


FIGURE 5. Comparison of Simulation and Measurement with Sizing error.

FIGURE 5 shows good corresponding tendencies of position and pressure levels between the simulation and the measurement. Furthermore, **TABLE 3** illustrates the deviations of measured and simulated values for the position (1.8%), the pressure in chamber A (20.2%) and the pressure in chamber B (16.1%). As a result, the correlation between the simulation and experimental data is acceptable. Therefore, this simulation model was utilized for finding the optimal compensation method from the perspective of energy efficiency. For this purpose, the following section will introduce the simulations and analyses of the proposed methods based on the validated model.

TABLE 3. Error between Simulation and Measurement with Sizing Error.

Results	**Position**		**Pressure in Chamber A**		**Pressure in Chamber B**	
	Max [mm]	**Error [%]**	**Max [MPa]**	**Error [%]**	**Max [MPa]**	**Error [%]**
Simulated	0.294	+1.8	6.8	-20.2	7.3	-16.1
Measured	0.289		8.5		8.7	

SIMULATION STUDY

This section investigates compensating the sizing error by a gear drive (mechanical compensation) and by a hydraulic accumulator (hydraulic compensation) as the unit P_2 is slightly under-dimensioned in the setup The simulations and measurements with compensation were carried out with a payload of 120 kg at an ambient temperature of 20 ℃. The energy efficiency for the DDH is computed using Eq. (1) for a 20 s cycle including lifting and lowering,

$$\eta = \frac{E_{\text{cylinder}}}{E_{\text{e-motor}}} = \frac{\int_0^{20} P_{\text{cylinder}} dt}{\int_0^{20} P_{\text{e-motor}} dt} = \frac{\int_0^{20} (p_A A_A - p_B A_B)\dot{x} dt}{\int_0^{20} (v_a i_a + v_b i_b + v_c i_c) dt} \quad (P_{\text{cylinder}} > 0, P_{\text{e-motor}} > 0), \tag{1}$$

where the efficiency denotes the proportion of the input hydraulic energy to the cylinder to the electric energy consumed by the electric motor without considering regeneration during lowering; p_A and p_B are the pressures of piston side and rod side of the cylinder, Pa; A_A and A_B are the effective cross areas of cylinder piston side and rod side, m^2; $\dot{x}$ is the piston velocity of the cylinder, m/s; v_a, v_b, and v_c are the phase voltages of the electric motor, V; i_a, i_b, and i_c are the phase currents of the electric motor, A.
In the following sub-sections, the measured speed of the cycle with or without the hydraulic accumulator was utilized as a reference speed for the gear and accumulator compensation simulation respectively.

Mechanical Compensation

For the mechanical solution, as shown in **FIGURE 1**(b), a gear drive was selected and installed for connecting the electric model to the two pumps. A gear ratio 1.2 determining the speed of the unit P_2 to that of the unit P_1 was adopted to compensate the sizing error, since the ideal pump ratio R_{ideal} is 0.75 and the realized pump ratio R_{real} equals to 0.63. A parameter-sweep for the gear ratio (in the range of 1-1.4 with an increment of 0.02) was performed to obtain the optimal efficiency and reduce the maximum pressure level in chamber B.

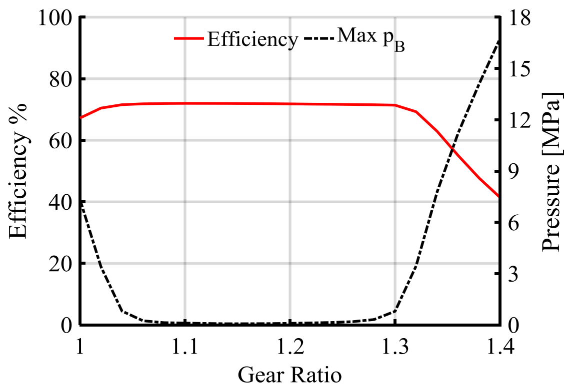


FIGURE 6. Simulation Results: the Effect of Differing Gear Ratio on Efficiency and Pressure in Chamber B.

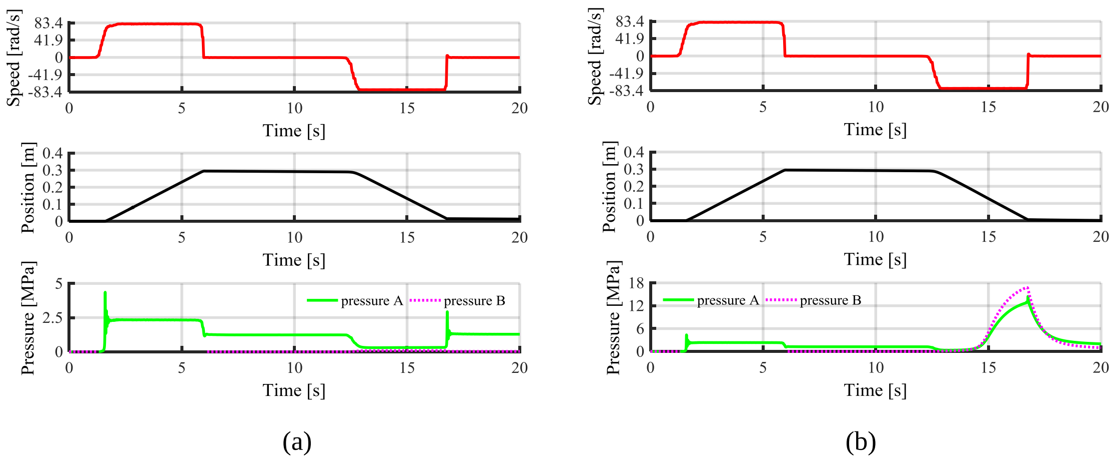


FIGURE 7. Simulation Result of Mechanical Compensation: (a) the best case with a Gear Ratio of 1.14, (b) the worst case with a Gear Ratio of 1.4.

FIGURE 6 demonstrates the simulation results: the effect of differing gear ratio on the efficiency and max pressure in the chamber B. **FIGURE 7** illustrates the simulation results of the mechanical compensation for the best case with a gear ratio of 1.14 and the worst case with a gear ratio of 1.4. As presented in **FIGURE 6**, the result shows that the efficiency is directly related to the maximum pressure of the chamber B, which is generated by the under-dimensioned unit P_2. As the gear ratio remains within the range of 1.08-1.22, the maximum pressure stays approximately at 0.1 MPa and the system efficiency is around 71.9%. According to **FIGURE 6**, the optimal ratio of gear compensation is 1.14 with the lowest pressure rise and the worst ratio is shown to be 1.40 with the highest pressure rise. **FIGURE 7** and **TABLE 4** demonstrate that their maximum pressures in chamber B are 0.07 MPa and 16.8 MPa, and their efficiencies are 72.0% and 41.3% respectively. Hence, the sizing error was eliminated by the mechanical compensation.

TABLE 4. The Best and Worst Case with Mechanical Compensation.

Results	Gear Ratio	Max p_B [MPa]	$E_{cylinder}$[kJ]	$E_{e\text{-}motor}$ [kJ]	Efficiency [%]
Without Compensation	1.00	7.34	1.90	2.83	67.1
The Best Case	1.14	0.07	1.93	2.69	72.0
The Worst Case	1.40	16.8	1.93	4.67	41.3

Hydraulic Compensation

For hydraulic compensation, as shown in **FIGURE 1**(c), a hydraulic accumulator was chosen and installed between the rod side of the cylinder and the unit P_2. Criteria for selecting hydraulic accumulator as a compensator were as follows:

- η_{max}, maximizing the energy efficiency;
- $p_{pre}A_B>F_{r,\,0.05\,m/s}$, to overcome the friction force with the desired piston velocity 0.05 m/s for lowering which corresponds to 1.0 MPa in the rod side chamber [11];
- $p_{max}/p_{pre}<3.5$, to fulfill the criteria described in the accumulator specification, the maximum pressure should not be 4 times higher than the precharge pressure (considered the potential inaccuracy of this simulation model, this value was changed to 3.5);
- $V_0>(A_B\text{-}A_AD_{p2}/D_{p1})\,x_{max}=0.13$L, the nominal volume of the accumulator should be greater than the overall fluid volume error for the rod side chamber, the difference between the volume in the rod side and fluid discharges by the unit P_2 when the piston chamber is fully charged by the unit P_1;
- $p_B>0$, for preventing cavitation;
- $V_0<1.0$L, for making the DDH package as compact as possible.

In order to find the optimal accumulator dimension for the DDH, a set of simulations were performed for the nominal volume of the accumulator differing from 0.15-1.0 L with an increment of 0.05 L and for the precharge pressure varying from 1.0 MPa to 4.0 MPa with an increment of 0.5 MPa. **FIGURE 8** shows the effect of nominal volume and precharge pressure on the energy efficiency of the DDH. In this case, the efficiency varies slightly as the nominal volume varies but the precharge pressure possesses a relatively higher impact on the efficiency. **FIGURE 9** and **TABLE 5** illustrate the best case and the worst case when utilizing the corresponding hydraulic accumulators with p_{pre}=1.0MPa, V_0=1L and with p_{pre}=4.0MPa, V_0=0.15L, where the maximum pressures in chamber B are 1.6 MPa and 4.2 MPa, and their efficiencies are 73.0% and 70.5% respectively. Therefore, the sizing error was eliminated by the hydraulic compensation with the accumulator p_{pre}=1.0MPa, V_0=1L.

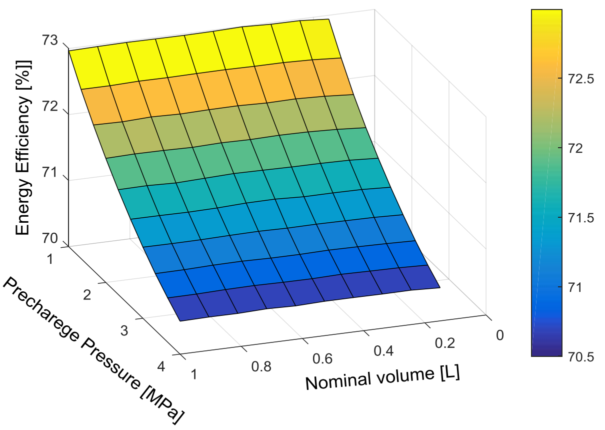
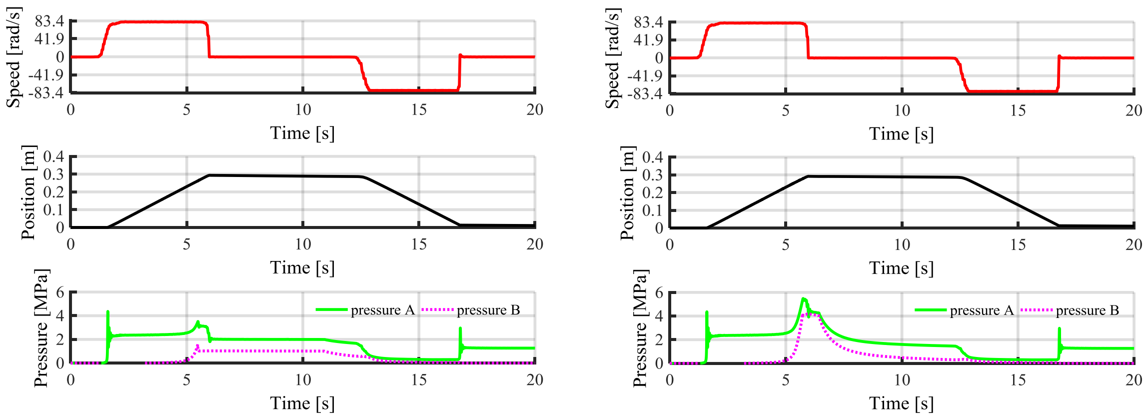


FIGURE 8. Simulation Results: the Effect of Varying Accumulator Parameters on Energy Efficiency of the DDH.



(a) (b)

FIGURE 9. Simulation Results of Hydraulic Accumulator Compensation: (a) the best case with V_0=1L and p_{pre}=1.0MPa, (b) the worst case with V_0=0.15L and p_{pre}=4.0MPa.

TABLE 5. The Best and Worst Case with Hydraulic Compensation.

Results	Hydraulic Accumulator		Max p_B [MPa]	$E_{cylinder}$[kJ]	$E_{e\text{-}motor}$ [kJ]	Efficiency [%]
	V_0 [L]	p_{pre} [MPa]				
The Best Case	1.0	1.0	1.6	1.92	2.64	73.0
The Worst Case	0.15	4.0	4.2	1.91	2.71	70.5

For validation study, a hydraulic accumulator with the nominal volume of 0.7 L and the precharge pressure of 1.0 MPa was implemented into the DDH setup as a hydraulic compensator for the sizing error, due to the negligible effect of nominal volume on the system efficiency and the available hydraulic accumulators on the lab-shelf. A measurement and a simulation were performed using this hydraulic accumulator as a compensator and the results are shown in **FIGURE 10.** The comparison of the positions and pressures shows the same tendency for the simulation and measurement but differs slightly. Thus, the accuracy of the model with the hydraulic compensation is acceptable for this study.

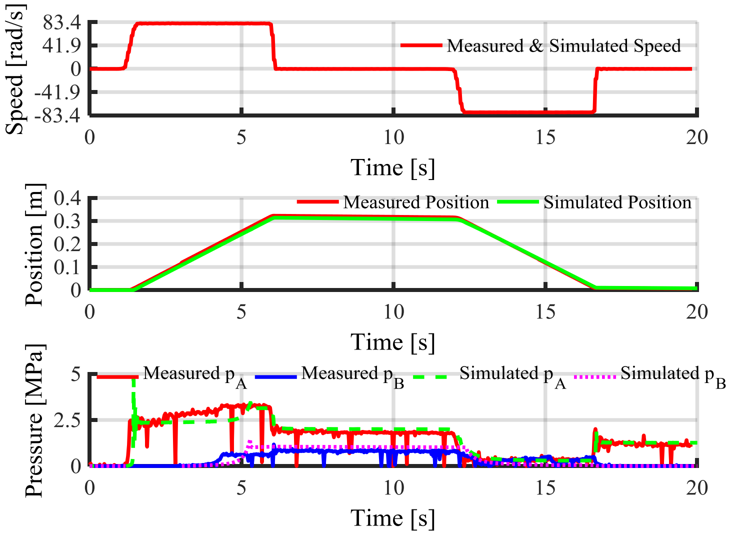


FIGURE 10. Comparison of Simulation and Measurement with Hydraulic Compensation.

DISCUSSION

In this research, two methods were proposed for correcting the sizing error. The model was constructed and investigated with a payload of 120 kg and a temperature of 20 ℃, which consisted of an electric motor, a gear drive, two pumps, hoses, a cylinder and a crane mechanism. The models were constructed for each proposed method in Matlab/Simulink based on the following simplifications: the overall efficiencies of the simplified electric motor and gear drive were set to be 95% and 96% respectively. The leakage of the pump was considered to be only varying with respect to the pressure difference across it and the hydro-mechanical efficiency of it had a constant value of 85%. The developed model predicts the energy consumption and the efficiency of the DDH at a constant operating temperature. Utilized simplifications in the model are considered only for initial analysis, as proposed model utilized simplifications such as only a single payload and temperature are utilized. In addition to it, internal and external leakages of the cylinder are not considered.

By utilizing the created Matlab/Simulink model, this study evaluates the consequences of the various accumulator parameters and gear ratios. For this purpose, a set of simulations were performed with different parameters of the hydraulic accumulator or gear ratios as the input variables. As illustrated in **TABLE 4** , the system efficiency is 67.1% without the mechanical compensation (the gear ratio 1.0). With the gear ratio in the range of 1.08-1.22, the maximum pressure in the chamber B of the cylinder becomes approximately to 0.1 MPa and the system efficiency is around 71.9%. As a result, the gear ratio 1.14 with the lowest pressure rise is considered to be the optimal ratio and best case for the mechanical compensation, and the corresponding system efficiency is 72.0%. Using the hydraulic accumulator with a precharge pressure from 1.0 to 4.0 MPa and a nominal volume from 0.1 to 1 L, the maximum pressure rise in chamber B is in the range of 1.6-4.2 MPa and the system efficiency varies from 70.5 to 73.0%. Based on the listed criteria, the best parameters of the hydraulic accumulator are that the precharge pressure equals 1.0 MPa and the nominal volume is 1 L. Comparison of the simulation results in **TABLE 4** and **TABLE 5** demonstrates the almost identical behavior of the hydraulic and mechanical compensation from the energy efficiency point of view. Also, the energy consumption for utilized cycles was demonstrated and only varies slightly.

In the simulation, although the gear ratio or hydraulic accumulator was adopted to compensate the sizing error, cavitation may occur during some period, caused by the bulk modulus of hydraulic fluid (the bulk modulus changing dramatically at low pressures with a fixed 4.5% of air) and the leakages of the pump units varying with respect to the system pressure and load pressure respectively. For the both proposed compensation methods, the simulations demonstrated that pressure is around vapor pressure during lifting (1.5-6.0s) and lowering stage (12.0-16.5s) of the cycle, which indicates possibilities of a cavitation problem. In addition, the mechanical solution is characterized as a constant pressure in chamber B with near 0 Pa during the holding stage (6.0-12.0s) of the cycle.

In the setup, there are more factors which will affect leakages of the pumps and cylinder and bulk modulus of the hydraulic fluid (varying with the system pressure, the air content and the temperature), such as the internal leakages of the cylinder (varying with the temperature and payload), the internal and external leakages of the pump units (varying with the load pressure, rotational speed, and temperature), and aging factors as well. However, this negative phenomenon can be avoided by installing a pair of anti-cavitation valves for both chambers.

In the hydraulic compensation, obviously, the size and precharge pressure of the hydraulic accumulator defined overall system efficiency. Bigger size and smaller precharge pressure give the best results. However, the mechanical compensation requires the adjustment of only one parameter: gear ratio. The required gear ratio for mechanical compensation can be achieved by a belt drive or a direct gear drive, which could simplify the setup construction.

Therefore, the decision about the application of hydraulic or mechanical compensation depends on the space and weight requirements of an application. Further, the combination of these proposed methods (the gear, hydraulic accumulator, and anti-cavitation valve compensations) can increase the system flexibility and tolerance to external and internal factors. This was not considered in this paper and could be researched in next step. Additionally, improvements regarding the detail level of the model are required.

CONCLUSION

A rising trend of decentralized hydraulics outside of airplane industry is presented in this study and applied to off-road machinery and stationary industrial applications. The direct-driven hydraulic (DDH) test setup without control valves was described and investigated from an efficiency point of view. In this paper, a hydraulic solution and mechanical solution were proposed to correct the sizing error. Further, the proposed methods were investigated by simulation studies and measurements.

The developed models, without compensation and with hydraulic compensation, were validated against measurements. The simulation results show that the energy efficiency improvement of the proposed mechanical and hydraulic compensation solutions are 7.3% and 8.8% and the max pressures in chamber B drop from 7.35 MPa to 0.07 MPa and 1.6 MPa respectively. The energy efficiency behavior of the DDH based on the models gives rise to consideration of the sizing error of the overall system efficiency. Therefore, it can be concluded that sizing error plays an important role in the continuous operation of DDH in order to maintain safety and reliability. The results showed that both of the proposed solutions are viable approaches to improve the energy efficiency of the DDH setup.

NOMENCLATURE

Designation	Denotation	Unit
R	Displacement Ratio	—
η	System Efficiency	—
E	Energy	[J]
P	Power	[W]
$p_{A,B}$	Pressure in Cylinder Chamber A and B	[Pa]
$A_{A,B}$	Effective Cross Section Area of Cylinder Chamber A and B	[m^2]
x	Displacement	[m]
$v_{a,b,c}$	Phase Voltage	[V]
$i_{a,b,c}$	Phase Current	[A]
p_{pre}	Precharge Pressure of Hydraulic Accumulator	[Pa]
V_0	Nominal Volume of Hydraulic Accumulator	[m^3]

ABBREVIATION

LS Load Sensing
IM Independent Metering
DFC Digital flow control
DDH direct-driven hydraulics
DC Displacement Control
HT Hydraulic Transformer
CPS Constant Pressure Systems
PMSM permanent magnet synchronous motor

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