Boosting Efficiency of an Excavator by Zonal Hydraulics

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Hybridization is frequently applied in order to increase the energy efficiency of off-road and construction machinery. One such novel energy saving method was proposed for working hydraulics, based on an established zonal concept for airplanes. The introduced method supplied the power on demand to the actuators, utilizing direct-driven hydraulics in machinery. However, component selection plays a significant role in order to achieve high efficiency. Therefore, the primary goal is to evaluate the energy efficiency of selected components for the front hoe of the micro excavator under the digging and levelling cycles. The efficiency of the zonal hydraulic pre-selected components was evaluated utilising a developed Matlab/Simulink model and energy efficiency maps.

Keywords: Excavator, Zonal Hydraulics, Energy Efficiency, Efficiency Map, Sizing, Direct Driven Hydraulics

Target audience: Mobile Hydraulics, Modelling, Design Process

1 Introduction

Due to high fossil fuel consumption and new CO₂ emission regulations, different concepts and methods to improve energy efficiency in heavy construction machinery have received considerable attention. Conventional construction machinery relies heavily to hydraulic system due to the power demands of their duty cycle. The hydraulic system components should be not just powerful and compact, but they need to be robust and able to withstand contamination and shock loads /1/.

In conventional construction machinery, the working hydraulics is usually supplied through a load-sensing (LS) control topology, which is widely implemented. In this topology, energy losses are significant, up to 35% in the control valves and 29% in the pump for a typical digging cycle of 27 seconds /2/. The simplified locations of the energy losses in a conventional construction machinery are demonstrated in Figure 1 for an excavator example. This conventional excavator is equipped with a single power unit and switching valve stack to control the actuators simultaneously.

Recent research efforts have focused on hybridization as a viable solution to improve the construction machinery efficiency in particular. For instance, /3/ described the energy efficiency improvements utilising a hybrid drive in a 20-tonne excavator. In /4/, a displacement-controlled architecture was applied to a 5-tonne excavator with the aim of capturing braking energy of the cabin swing motion. However, the construction machinery and off-road industry still faces challenges on energy saving, due to the high output power required on its duty cycle and fuel consumption. Also for its limited space and utilised asymmetrical actuators. In our previous study /5/, a novel energy saving method for working hydraulics was proposed, based on a zonal concept well established by the aircraft industry. This new topology supplies power directly to actuators with direct driven hydraulics (DDH), mainly composed of two fixed displacement pump/motor units and a servomotor. Independent DDH units are placed next to the actuators and convert electric energy into hydraulic on demand (see Figure 2). Figure 2 illustrates a new system configuration on the excavator and location of energy losses. In this configuration, for working hydraulics, DDH losses consist of electric motor, pump, and cylinder losses. The energy storage, frequency converter, electric motor and the pump (located under operator seat) are for driving powertrain and are excluded from the scope of this research. In /5/, the authors demonstrated with a simulation study an increase of overall efficiency from 18.3% to 71.3% for a 1-tonne micro excavator and highlighted the requirements for selecting of components for the DDH units.

2 Test Case

A JCB 1-tonne micro excavator was chosen as a platform for the analysis. In previous project, the 14 kW diesel engine of the excavator was replaced with a 10 kW electric motor and a lithium-titanate battery. In this study, the original LS control circuit for the boom, stick, and bucket were replaced with independent direct driven hydraulics (DDH) units as proposed in Figure 2. The Schematics of the single DDH unit is shown in Figure 3. Each DDH unit consists of two fixed displacement pump/motors directly controlled by a speed-controlled electric servomotor without conventional directional control valves. Two on/off valves were applied to hold the load positions, and one low pressure accumulator is utilised as a reservoir. For clarity in this paper, the pump/motor refers to the reversible motors which is utilised in pumping and motoring mode. The components for the DDH units should be selected based on the dimensions of the excavator cylinders. The dimensions of the utilised cylinders in the micro excavator are illustrated in Table 1.

![Figure 1: Energy losses in conventional excavator](image1.png)

![Figure 2: Energy losses in electro-hydraulic excavator with zonal hydraulics](image2.png)
<table>
<thead>
<tr>
<th>Actuator</th>
<th>Piston Diameter [mm]</th>
<th>Rod Diameter [mm]</th>
<th>Stroke Length [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bucket</td>
<td>50</td>
<td>30</td>
<td>290</td>
</tr>
<tr>
<td>Stick</td>
<td>50</td>
<td>30</td>
<td>410</td>
</tr>
<tr>
<td>Boom</td>
<td>60</td>
<td>30</td>
<td>325</td>
</tr>
</tbody>
</table>

Table 1: Dimensions of cylinders.

Figure 3: Schematic of a single DDH unit composed by: 1- Asymmetric hydraulic actuator, 2- On/Off valves, 3- External gear pump/motor for piston side of cylinder, 4- Servomotor, 5- External gear motor for piston rod side and 6- Hydraulic accumulator.

The work cycle of the excavator consists of the continuous cooperation of travelling, slewing, and working hydraulics. This investigation is concentrated only on the working hydraulics of the front attachment of the micro excavator and, therefore, the swing system of the front attachment is not considered, as well as the slewing system of the upper-structure, and the hydrostatic transmission for travelling.

Usually digging and levelling work requires the cooperation of three cylinders for lifting, lowering, and digging, represented by a trench digging cycle in loose gravel up to 27 seconds [2]. In [6], it is concluded that the experienced operators had an average cycle time of 24.5 seconds using a 20-tonne excavator. For this simulation study, a typical working cycle for the excavator from [2] was adopted as the input reference for the DDH system. In addition, utilised cycle was shortened to 20 seconds, considering the size differences between a JCB1-tonne micro and 20-tonne excavator, and assumption that experienced operator drives machine. The term-mechanical forces acting on the bucket were simplified and replaced with a time-dependent load [2]. The load force acting on the cylinder during the working cycle is defined according to CECE 2:1 standard [7]. To calculate maximum load fit to the bucket the density of clay and gravel of 2100 kg/m³ has been utilised for this simulation. Thus, the maximum load of 72kg (706 N) acts vertically downward along gravity direction.

In addition, a levelling cycle was added to widen investigation of this work. Operating sequences of both levelling and digging cycles are illustrated in Figure 4, along with the relative position of the acting cylinders.

For evaluation purposes, the simulation model was utilised track the operation points of the DDH system. Following section introduces the developed model and efficiency maps for the analysis.

Figure 4: Levelling cycle (Left) and Digging cycle (Right) adopted for micro excavator simulation.

3 Model
This study created a Matlab/Simscape model and efficiency maps for evaluation of the hydraulic and electric components of the DDH circuit for the excavator case. The DDH unit consists of the controller, multibody, hydraulic components, and electric motor. The simplified hydraulic system of the DDH unit Simscape model was adopted from [8]. In the model, the cylinder position is controlled with a closed-loop PI control. The 3D solid mechanical model was constructed in PTC Creo and exported into Matlab utilizing Simscape Multibody Link Plug-In [9]. Most of the detailed explanations refer to the work in [8, 10].

The efficiency maps are built according to particular features of each hydraulic pump, such as the coefficients presented in the section 3.1. Once obtained the maps, the operation points during the selected cycles investigated are plotted over these maps to verify how the system behaves during the operation and which range of efficiency that pump/motor is working in section 4.1. Identical work was carried out for electrical machine in section 3.2.

The following section introduces in detail the explanation for building efficiency maps for the hydraulic pump/motor and electric machine, respectively. Section 3.2 presents parameters of selected components.

3.1 Efficiency map for hydraulic pump/motor
In order to estimate the flow rate and torque losses in a wider range of operation points, this paper utilised the Schlosser model to calculate the pressure and speed dependency of pump/motor efficiencies, which can model it with an acceptable amount of accuracy [11].

The flow rate model for the pumping mode is determined by Equation (1), which considers laminar leakages and turbulent leakages.

\[
Q_{\text{pump}} = \frac{\dot{e} V_i}{\omega} - \frac{\dot{W}_{\text{vis}}}{\omega} - C_d V_{i}^{1/2} \sqrt{\frac{\Delta p}{\rho}}
\]

(1)

where \( Q \) is the flow rate delivered by the pump, \( \dot{e} \) is the setting of displacement \( (e \approx 1, \text{ fixed displacement}) \); \( V_i \) is the pump/motor displacement; \( \omega \) is the rotational speed; \( C_d \) is the slip coefficient; \( \Delta p \) is the pressure difference over the pump; \( \nu = 32 \text{ m}^2/\text{s} \) is the kinematic viscosity of hydraulic fluid; \( \rho = 860 \text{ kg/m}^3 \) is the density of fluid; and \( C_d \) is the turbulent slip coefficient.

The volumetric efficiency is given by Equation (2), dividing the calculated flow rate per theoretical flow rate.

\[
\eta_{V\text{-pump}} = \frac{Q_{\text{pump}}}{\dot{e} V_i}.
\]

(2)
To obtain the hydro-mechanical efficiency, the Schlösser torque model considers Coulomb friction, viscous friction, and hydrodynamic friction. The torque is given by Equation (3):

$$T_{\text{pump}} = \frac{c_v \rho V}{2\pi} + \frac{c_f \rho V}{2\pi} + \frac{c_h \rho V}{2\pi} + \frac{c_g}{4\pi} \omega \rho V,$$

(3)

where $c_v$ is the Coulomb friction coefficient; $c_f$ is the viscous friction coefficient; and $c_h$ is the hydrodynamic friction coefficient.

Equation (4) presents the hydro-mechanical efficiency, dividing the theoretical torque per calculated torque.

$$\eta_{\text{pump}} = \frac{\eta_{\text{motor}}}{\eta_{\text{motor}}}. $$

(4)

For the motoring mode, the flow rate mode and torque mode are the same equations but inverting the operator’s signals as shown in the Equations (5) and (6), respectively.

$$Q_{\text{motor}} = \frac{c_v \rho V}{2\pi} + \frac{c_f \rho V}{2\pi} + \frac{c_h \rho V}{2\pi} + \frac{c_g}{4\pi} \omega \rho V,$$

(5)

$$T_{\text{motor}} = \frac{c_v \rho V}{2\pi} + \frac{c_f \rho V}{2\pi} - \frac{c_h \rho V}{2\pi} - \frac{c_g}{4\pi} \omega \rho V,$$

(6)

In addition, the volumetric and the hydro-mechanical efficiency are given by Equation (7) and (8), respectively.

$$\eta_V = \frac{c_v \rho V}{\eta_{\text{motor}}},$$

(7)

$$\eta_{\text{m}} = \frac{2\pi T_{\text{motor}}}{4\pi V \rho V}.$$  

(8)

The total efficiency $\eta_T$ is then obtained by multiplying both efficiencies previously calculated for pump and motor modes, as shown in Equation (9).

$$\eta_T = \eta_T \eta_{\text{m}}.$$  

(9)

Efficiency map for the selected pump/motor calculated using the manufacture given parameters illustrates in section Results.

### 3.2 Efficiency map for electric machine

In this study, the efficiency map of the permanent magnet synchronous machine (PMSM) were carried out based on the machine design calculations and loss analysis according to the procedure described in [12].

These losses are composed of the following elements: stator and rotor resistive losses, iron losses, additional losses and mechanical losses. The copper loss is expressed as:

$$P_{\text{Cu}} = 3 R_s i_s^2,$$

(10)

where $R_s$ is the stator resistance, and $i_s$ the stator phase current.

Iron losses were estimated based on the calculated by following equation:

$$P_{\text{Fe}} = k_r P_r \left(\frac{6}{127}\right) i_{s},$$

(11)

where correction coefficient $k_r$ for yoke is 1.5 and 1.7 for teeth, $m_o$ is mass of area. Mechanical losses include friction in the motor bearings and the fan for air cooling. To evaluate the friction losses is utilised the following equation.

$$P_f = k_f P_r (l + 0.6i_r) v_r^2,$$

(12)

where $k_f = 10 W e^{-3} m^2$ is an experimental factor, $l$ the rotor length, $v_r$ the volume of the rotor in m³, $i_r$ is the stator pole pitch in m, and $D_r$ is the rotor diameter in m.

Based on [12], the varying additional losses and the permanent magnet losses were estimated to be 7.5% and 5% from output power, respectively.

$$P_{\text{ad}} = 0.075 \cdot P_{\text{out}},$$

(13)

$$P_{\text{pm}} = 0.05 \cdot P_{\text{out}}.$$  

(14)

Efficiency map for the selected electric machine calculated using the manufacture given parameters illustrates in section Results.

### 3.3 Component selection

Based on initial calculations, preliminary components were selected for the micro excavator hybridization by means of zonal hydraulics (ODH units). These hydraulic pump/motors should be dimensioned to the match expected flows of the corresponding chamber in an utilised differential cylinder. In order to avoid higher pressure rise in the chambers due to the sizing error between the available pump units. For this study, external gear reversible motors by Bosch Rexroth were selected, and Table 2 presents specifications of these components.

<table>
<thead>
<tr>
<th>Pump/motor</th>
<th>Displacement, [cm³/rev]</th>
<th>Speed, [rpm]</th>
<th>Pressure, [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bucket (Piston Side)</td>
<td>6.3</td>
<td>750-3500</td>
<td>220</td>
</tr>
<tr>
<td>Bucket (Rod Side)</td>
<td>4.0</td>
<td>750-4000</td>
<td>220</td>
</tr>
<tr>
<td>Stick (Piston Side)</td>
<td>6.3</td>
<td>750-3500</td>
<td>220</td>
</tr>
<tr>
<td>Stick (Rod Side)</td>
<td>4.0</td>
<td>750-4000</td>
<td>220</td>
</tr>
<tr>
<td>Boom (Piston Side)</td>
<td>11.0</td>
<td>500-3500</td>
<td>210</td>
</tr>
<tr>
<td>Boom (Rod Side)</td>
<td>8.0</td>
<td>500-4000</td>
<td>210</td>
</tr>
</tbody>
</table>

**Table 2: Pump/motor parameters [13].**

Permanent magnet synchronous servomotors MST130C-0200F servomotor by Bosch Rexroth was selected, and its rated parameters are listed in Table 3. Following section demonstrates the simulation results.

<table>
<thead>
<tr>
<th>Electric machine parameters</th>
<th>Values</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum stator voltage $U_{\text{in}}$</td>
<td>380</td>
<td>[V]</td>
</tr>
<tr>
<td>Rated stator current $I_s$</td>
<td>15.2</td>
<td>[A]</td>
</tr>
<tr>
<td>Rated torque $T_e$</td>
<td>25</td>
<td>[Nm]</td>
</tr>
<tr>
<td>Rated active power $P_e$</td>
<td>5.24</td>
<td>[kW]</td>
</tr>
<tr>
<td>Rated speed $n_r$</td>
<td>3500</td>
<td>[rpm]</td>
</tr>
<tr>
<td>Number of pole pairs $p_r$</td>
<td>10</td>
<td>[-]</td>
</tr>
<tr>
<td>Rated stator flux linkage $\psi_r$</td>
<td>0.27</td>
<td>[Vs]</td>
</tr>
<tr>
<td>Synchronous inductance, d axis value $L_d$</td>
<td>6.6</td>
<td>[mH]</td>
</tr>
<tr>
<td>Synchronous inductance, q axis value $L_q$</td>
<td>6.6</td>
<td>[mH]</td>
</tr>
<tr>
<td>Synchronous resistance $R_s$</td>
<td>1.6</td>
<td>[Ω]</td>
</tr>
<tr>
<td>Synchronous resistance at room temperature $R_o$</td>
<td>0.0018</td>
<td>[kg/m³]</td>
</tr>
</tbody>
</table>

**Table 3: PMSM parameters [14].**
4 Results

The main objective of this investigation is to validate the selected components size with target to optimise the energy efficiency of a micro excavator with zonal hydraulics under a typical digging and levelling cycle. Developed efficiency maps illustrate how much the efficiency of the component will change during the utilised cycles.

Following section presents energy efficiency maps as a function of parameters, such as required power and/or speed, for hydraulic pump/motor and electric motor in Sections 4.1 and 4.2, respectively. In figures pumping mode for pump/motor is correspond to positive speed, and motoring mode is correspond to negative speed. For electric machine: the same sign for speed and torque correspond to motoring mode, and opposite sign correspond to generating mode. Section 4.3 includes analysis of simulation results.

4.1 Pump/motor

Figures 5 and 6 illustrate the efficiency maps for stick actuation utilising the selected pump/motors for the levelling and digging cycle for the piston and the piston rod sides, respectively. It can be seen in Figure 5 that the pump/motor is switching the operation mode in both cycles. During pumping mode, the pump/motor operates in high efficiency zone in range of 80 and 84%. However, when it changes to motoring mode, the pressure decreases in comparison to the pumping mode due to slight dimension imbalance between expected flows of the corresponding chamber in the utilised differential cylinder. Thus, the piston side pump/motor is slightly oversized.

In Figure 6, piston rod side pump/motor during pumping and motoring mode for stick are working in high efficiency zone most of the time.

Figures 7 and 8 illustrate efficiency maps for selected pump/motors for boom actuation for digging and levelling cycles for piston and piston rod side, respectively.

![Figure 5: Efficiency map for piston side pump/motor (Stick DDH) for digging and levelling cycle.](image1)

![Figure 6: Efficiency map for piston rod side pump/motor (Stick DDH) for digging and levelling cycle.](image2)

![Figure 7: Efficiency map for piston side pump/motor (Boom DDH) for digging and levelling cycle.](image3)
It can be seen in Figure 7 identical behaviour as in Figure 5 for digging cycle. Figure 8 demonstrates that piston rod side pump/motor for boom selection operates in high pressure zone located outside high efficiency region. Efficiency of the piston rod side pump/motor is varies between 73% and 80%. In contrast, piston side pump/motor is mostly collated in high efficiency region and low pressure zone. The efficiency of the piston side pump/motor is varies between 83% and 80% during pumping mode.

4.2 Electric machine

Figures 9 and 10 demonstrate the efficiency maps for the selected electric machine with operation points for boom, stick and bucket actuators for levelling cycle and digging cycle, respectively. It can be seen that electric machine is working in rated torque region, and is slightly over dimensioned. However, it leads to a safety gap to perform high power demand cycles with lower overheating risk. Boom efficiency varies in range of 94 and 80%. Stick operation points are mostly located in high efficiency in range of 95 and 97%. The electric machine for the bucket cylinder has operation points which are located outside developed efficiency map.

Figure 10 demonstrates that the bucket and stick actuators are powered with high efficiency in range of 95% and 97%. The maximum efficiency of the boom electric machine is 95% for digging cycle.

4.3 Total system

In this study, the electric machine, the inverter, and the hydraulic machine will have operating points following particular working cycles (shown in Figs. 4). The data from developed simulation model are utilised for determination of the cycle efficiency of independent DDH for excavator with zonal hydraulics. The efficiencies of the pump/motor and the electric motor are depicted in Figures 5-8 and 9-10, respectively. Cycle total efficiency was calculated with Equation (15).
\[
\eta_{\text{cylin}} = \frac{F_{\text{out}}}{F_{\text{in}}} = \frac{P_{\text{out}}}{P_{\text{in}}} \tag{15}
\]

where \( v \) is the cylinder velocity; \( F \) is the output force of the cylinder; \( f \) is the output motor torque; and \( \omega \) is the motor speed. The efficiencies for selected components and the cycle total efficiency of the excavator are listed in Table 4 for digging cycle with a payload of 72 kg and for levelling cycle.

<table>
<thead>
<tr>
<th>Actuator</th>
<th>Levelling cycle</th>
<th>Digging cycle</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Efficiency of electric machine (%)</td>
<td>Efficiency of pump/motor(*) (%)</td>
</tr>
<tr>
<td></td>
<td>max</td>
<td>min</td>
</tr>
<tr>
<td>Bucket</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Stick</td>
<td>97</td>
<td>80</td>
</tr>
<tr>
<td>Boom</td>
<td>95</td>
<td>80</td>
</tr>
</tbody>
</table>

(*) Efficiency calculated only for retreating motion due to regeneration mode during lowering.

According to Table 4, efficiency of the electric machine in average higher than 80 % and will boost significantly to overall excavator efficiency in comparison to conventional system which efficiency is 10-20 %/10.

5 Discussion

This study is a continuation of the investigation of a micro excavator with zonal hydraulics as a part of IZIF project. This paper evaluate selected components to the zonal hydraulics based on the energy efficiency map as a function of parameters, such as pressure, torque, and speed.

The efficiency of the selected components (pump/motor and electric motor) on the total energy efficiency for zonal hydraulics was investigated based on a developed Matlab/Simulink model and efficiency maps. The developed electro-hydraulic model was coupled with multibody dynamics and utilised a typical working cycle for an excavator. The developed Matlab/Simulink model of the DDH was validated against measurements in previous studies /8/ and /9/. In addition, the generation mode was not captured by the developed model, despite an early experimental study, which demonstrated the ability to recuperate energy.

The developed efficiency maps were utilised to validate the selected component size with the target to optimise the energy efficiency of a micro excavator with zonal hydraulics under a typical digging and levelling cycle. The efficiency maps are based on the Schloesser hydraulic loss model and basics of electric machine design. The maps illustrated how much the efficiency of the component varied during the utilised cycles. The efficiency map for the electric machine illustrated that the selected component was oversized, and combination of 3 units for the excavator will become quite a powerful combination. However, efficiency maps do not take into account near zero speed operation, which is the critical operation for DDH.

The simulation results demonstrated the total efficiency of the excavator with zonal hydraulics is with range 38 and 44%, where hydraulic pump/motor efficiency varies between 77 and 84%, and the electric machine efficiency varies between 97 and 80%. The developed pump/motor loss and electric machine models and selected duty cycles are intended to be typical; however, it certainly does not represent all possible configurations. Therefore, the aim of this work was to demonstrate that the impact of the selected component efficiency on the total efficiency of DDH units to the systems is significant. This effect is particularly noticeable at partial actuator velocities (motor speeds) and loads (operating pressure and required torque), where machines operate most of the time.

Despite simplifications regarding losses and cycles in the model, significant differences in total energy efficiency between the components were captured. For this research, the performed simplifications are considered acceptable.

However, the effect of various working cycles with extreme operation points such as speed and pressure should be investigated in future. Despite the demonstrated high efficiency of DDH, it is important to notice that it is coming with the price of extra electric components, such as an expensive battery, electric motors, and motor controllers. Furthermore, these results should be validated against measurements, and utilised for the development of a proof of concept for the application of zonal hydraulics for the micro excavator and other off-road machinery.

6 Conclusion

In domain of the IZIF project, the implementation of zonal hydraulics with direct driven hydraulics (DDH) was investigated for hybridization purposes. In this study, evaluation of selected components for DDH was exploited for a micro excavator case. The resulting evaluation of the selected hydraulic and electric components circuit was performed based on a created multibody electro-hydraulic model and developed energy efficiency maps from the energy efficiency point of view. The investigation demonstrated the variation on the energy efficiency for pump/motors between 73% and 84% and electric machine between 80% and 97%. The cycle total efficiency of the independent DDH unit was boosted to 38% and 44% for the selected digging and levelling cycles.

7 Acknowledgements

The research was enabled by the financial support of the IZIF (Implementation of zonal integrated future fluid power systems) project, and internal funding at the Department of Mechanical Engineering at Aalto University, Finland. Also the Natural Science Foundation of Fujian Province, China (No.2016J01303) and by the Scientific Research Fund of Fujian University of Technology (No. GY-Z13096). In addition, grateful for all the support from LASHIP group (UFSC, Brazil). Special thanks to Heikki Kauranne for help and support for writing this paper.

### Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \Delta p )</td>
<td>Pressure Difference</td>
<td>[bar]</td>
</tr>
<tr>
<td>( \varepsilon )</td>
<td>Setting of Displacement</td>
<td>-</td>
</tr>
<tr>
<td>( \eta_{HM} )</td>
<td>Hydro-mechanical efficiency</td>
<td>-</td>
</tr>
<tr>
<td>( \eta_T )</td>
<td>Total Efficiency</td>
<td>-</td>
</tr>
<tr>
<td>( \eta_V )</td>
<td>Volumetric Efficiency</td>
<td>-</td>
</tr>
<tr>
<td>( \rho )</td>
<td>Density</td>
<td>[kg/m³]</td>
</tr>
<tr>
<td>( \nu )</td>
<td>Kinematic Viscosity</td>
<td>[m²/s]</td>
</tr>
<tr>
<td>( \nu_l )</td>
<td>Volume of Rotor</td>
<td>[m³]</td>
</tr>
<tr>
<td>( \tau_l )</td>
<td>Stator Pole Pitch</td>
<td>[m]</td>
</tr>
<tr>
<td>( \psi_l )</td>
<td>Flux Linkage</td>
<td>[Vs]</td>
</tr>
<tr>
<td>( \omega_l )</td>
<td>Rotational Speed</td>
<td>[rad/s]</td>
</tr>
<tr>
<td>( \xi_f )</td>
<td>Coulomb Friction Coefficient</td>
<td>-</td>
</tr>
<tr>
<td>( \xi_h )</td>
<td>Hydrodynamic Friction Coefficient</td>
<td>-</td>
</tr>
<tr>
<td>( \xi_s )</td>
<td>Slip Coefficient</td>
<td>-</td>
</tr>
<tr>
<td>( \xi_{st} )</td>
<td>Turbulent Slip Coefficient</td>
<td>-</td>
</tr>
</tbody>
</table>
$C_f$ Viscous Friction Coefficient

$D_r$ Rotor Diamenter [m]

$F$ Force [N]

$I_e$ Stator Phase Current [A]

$I_{en}$ Inertia [kgm$^2$]

$k_d$ Correction Coefficient -

$k_p$ Experimental Factor -

$l$ Rotor Length [m]

$L_d$ Synchronous Inductance, d axis [H]

$L_q$ Synchronous Inductance, q axis [H]

$m_a$ Mass of area [kg]

$n_a$ Rotational Speed of the Electric Motor [rpm]

$p_a$ Pole pairs -

$P$ Power [W]

$Q$ Flow Rate [m$^3$/s]

$R_s$ Stator Resistance [Ω]

$T$ Torque [Nm]

$T_{n}$ Nominal torque of the Electric Motor [Nm]

$U$ Voltage [V]

$v$ Velocity [m/s]

$V_i$ Displacement [m$^3$/rev]

References


[9/] Matlab, Enable Simscape Multibody Link Creo-Pro/ E Plug-In, [online] Available at: https://se.mathworks.com/help/phylsm/smlink/ref/linking-and-unlinking-simmechanics-link-software-with-proengineer.html


