

Adaptive Robust Motion Control of an Excavator Hydraulic Hybrid Swing Drive

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ABSTRACT

Over the last decade, a number of hybrid architectures have been proposed with the main goal of minimizing energy consumption of off-highway vehicles. One of the architecture subsets which has progressively gained attention is hydraulic hybrids for earth-moving equipment. Among these architectures, hydraulic hybrids with secondary-controlled drives have proven to be a reliable, implementable, and highly efficient alternative with the potential for up to 50% engine downsizing when applied to excavator truck-loading cycles. Multi-input multi-output (MIMO) robust linear control strategies have been developed by the authors' group with notable improvements on the control of the state of charge of the high pressure accumulator. Nonetheless, the challenge remains to improve the actuator position and velocity tracking. Recent developments on adaptive robust control (ARC) algorithms for displacement-controlled actuation systems by the authors' group have opened the door for the application of more advanced control strategies for hydraulic hybrid systems. This paper presents the first time synthesis of an ARC strategy for the motion control of the secondary-controlled hydraulic hybrid system. Similar to work previously developed by the authors' group, the controller is formulated to compensate for uncertain parameters through online parameter adaptation. Additionally, its structure allows for the inclusion of un-modeled nonlinearities such as friction, nonlinear friction of low frequency content and external loads and disturbances. Transient performance, and tracking accuracy are also guaranteed in the presence of both parametric uncertainties and uncertain nonlinearities and asymptotic tracking is achieved in the presence of parametric uncertainties. To evaluate the synthesized ARC controller, a prototype excavator is utilized and the measurement results are compared with those of a PI and an H_∞ controller.

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INTRODUCTION

Off-highway vehicles' systems have been the subject of extensive engineering research for the past few decades. In particular, novel architectures and control algorithms have been successfully developed which maximize the overall system performance and efficiency. A sector that has demonstrated large improvements in both aspects is the earthmoving equipment and much attention has been paid to excavators due to their largely cyclical operation. Additionally, the high inertial forces experienced by the machine makes it ideal for energy recuperation. One approach to exploit this type of machines' operation are hybridized systems. Hybrid excavators with electric storage devices were first introduced in the market by Kobelco and Komatsu. In their systems, an electric motor is installed as the drive actuator and an electric capacitor is installed for storage of the kinetic energy during braking. These manufacturers have advertised up to 41% energy savings for specific working cycles [1], [2] and [3]. However, the biggest disadvantage of the electrical hybrid systems is their high cost. More recently, CAT commercialized a hydraulic hybrid excavator [4]. In this case, swing braking energy is recuperated in a high pressure accumulator and stored for reuse. Besides the commercially available hydraulic hybrid concepts, the

authors' group at Purdue has studied, implemented and tested the concept of a secondary controlled hybrid swing drive as part of the NSF center for compact and efficient fluid power (CCEFP) [5], [6].

The concept of secondary control actuation is not new to fluid power systems. The idea was originally patented in [7] as a more efficient alternative to valve controlled actuation. The hydraulic circuit shown in [Figure 1](#) illustrates the concept. The working principle of secondary controlled actuation may be explained as follows. In reference to [Figure 1](#), the primary unit (1) is employed to control the high pressure at the working port of the secondary units (2) and (3). This is achieved via the primary unit (1) pressure-compensated control (3). To control either the actuator position and velocity or torque, the secondary units (2) and (3) displacement control (5) and (6) respectively are employed. It is important to note that depending on the system operation, the secondary unit (2) may operate as a pump or a motor. To prevent this unit from over speeding it must be controlled closing the loop on its shaft's rotational speed.

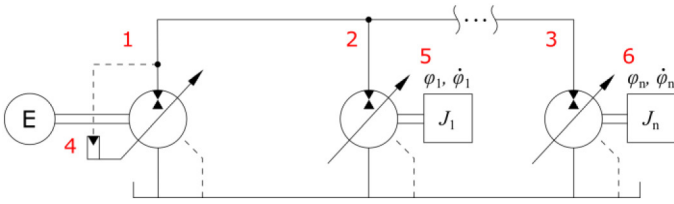


Figure 1. Basic secondary-controlled drive hydraulic circuit

Since the conception of this technology, many advances in the control of the actuator position, velocity and torque have been developed. Linear control theory was utilized by Murrenhoff and Hass [8] and [9] to control the secondary unit position and velocity. These approaches suffered of low performance relative to valve controlled actuators. More advanced control concepts were developed in [10]. Here, an adaptive controller was developed based on the system linear state space. Since no observers were used for this control law, the controller is equivalent to a cascaded proportional control. Similar to the linear approaches in [8] and [9], this controller achieved a limited bandwidth. Some of these challenges posed by the highly nonlinear nature of the system have been overcome through the development of nonlinear techniques as is the case of [11]. Robust linear controllers have also been investigated [12]. All of the aforementioned approaches assume a constant pressure at the working port of the secondary unit. To overcome this impediment, the authors' group developed and implemented a robust MIMO linear controller [13] which takes into account the changes in the accumulator state of charge as a structured uncertainty.

Adaptive robust control (ARC) is a technique developed by Yao [14] which combines both adaptive control as well as robust control while overcoming the challenges of both. As a control algorithm, it has shown extremely accurate actuator motion control both in simulation and implementation for electrical systems, hydraulic valve-controlled (VC) systems [14], [15], [16] and [17] and for hydraulic displacement-controlled (DC) systems [18]. Some of the advantages of ARC, which are an improvement over backstepping adaptive control, include guaranteed transient performance and final tracking accuracy to a certain degree. An essential feature of the ARC approach is the consideration of unknown nonlinear functions, which are not accounted for in backstepping adaptive control. This is achieved by using the deterministic robust control (DRC) technique as a baseline control law when formulating the controller. Additionally, parameter adaptation is utilized to reduce model uncertainties due to parametric uncertainties and improved tracking performance [14]. Finally, different from backstepping adaptive control, which deals with uncertain parameters through exact cancellation, ARC relies on strong feedback control to guarantee stability.

For a secondary-controlled hydraulic hybrid drive, a constant pressure net is unreasonable. With this in mind, the authors' group has developed advanced control concepts for the power management of such system. On the actuator level, one of the requirements is that the accumulator state of charge is maintained at a commanded level dictated by the supervisory controller. This parameter may vary in

magnitude with low frequency content depending on the commanded drive operation and the inertial load level. A second requirement, which may be more stringent, is a good tracking of the actuator speed or position. For the excavator application, the challenge lays on the highly dynamic operation as well as the largely varying loads. For this reason, the author's group developed a robust MIMO linear controller [13] which takes into account the operation of both the primary and the secondary units to improve tracking. In the work presented in [13] it was evident that the pressure tracking was very much improved; however, the cabin velocity and position tracking was not. The work developed in this paper focuses on the synthesis of an ARC nonlinear controller which aims to improve actuator velocity and position tracking. The proposed idea is to retain the previously proposed robust MIMO controller synthesized in [13] for the control of the primary unit and utilize the adaptive robust nonlinear controller for the control of the inertial load motion.

EXCAVATOR SYSTEM

A DC excavator prototype is the subject of study in this paper. The working hydraulics consist of four-18 cc/rev variable displacement axial piston machines, six single rod linear hydraulic cylinders to actuate the boom, arm, bucket, offset, and blade, two fixed displacement radial piston motors for the tracks and a variable displacement axial piston pump/motor for the secondary-controlled hybrid swing drive. To illustrate the general idea of DC actuation, [Figure 2](#) shows a hydraulic circuit of a single linear actuator. In reference to [Figure 2](#), the actuator (8) is controlled by varying the hydraulic unit's (2) displacement, in the case where the hydraulic unit is of variable displacement type, or varying the prime mover's (1) rotational speed, in the case where the hydraulic unit displacement is fixed. For the absorption and compensation of the actuator (8) differential volume the pilot-operated (PO) check valves (3 and 4) are connected to a constant pressure flow source (7). This pressure source (7) typically comprises a fixed displacement external gear pump and a relief valve to set the pressure level (usually between 15 and 30 bar). Installing a low pressure accumulator in this subsystem allows for the minimization of the external gear pump volumetric displacement thereby reducing total losses. Finally, to prevent over-pressurization, pressure relief valves (5 and 6) are installed.

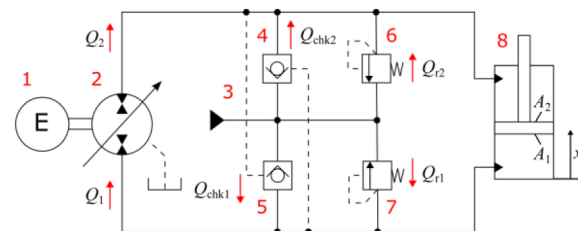


Figure 2. Displacement-controlled single-rod cylinder hydraulic circuit

The secondary-controlled swing drive in the excavator prototype follows the underlying working principles of secondary-control as illustrated in [Figure 1](#). Nonetheless, the hydraulic architecture has been modified to include a 6-liters hydraulic accumulator with a pre-charge of 200 bar for energy storage and the pressure

compensation system has been replaced with an electro-hydraulic system which allows direct control of the primary unit's displacement. Then, both hydraulic units are allowed to operate as pumps or motors based on the machine operation, which ultimately allows for efficient energy distribution.

EXCAVATOR MODEL

The excavator under consideration is a Bobcat 435, 5-ton compact machine. For illustration purposes, the excavator model layout is shown in Figure 3.

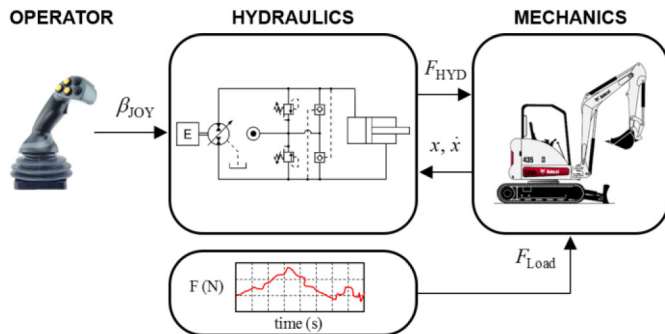


Figure 3. Excavator system model layout

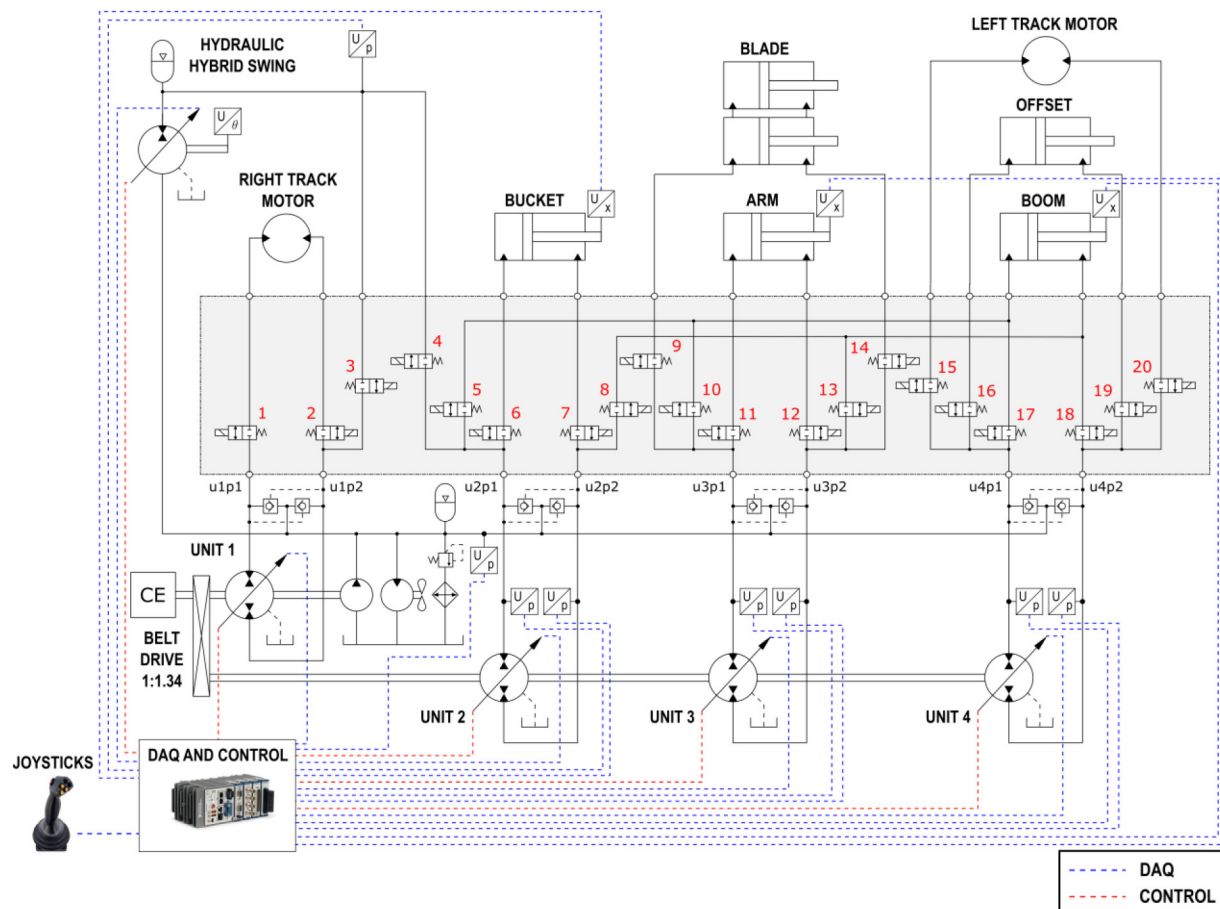


Figure 4. Displacement-controlled excavator prototype working hydraulics circuit with data acquisition and control instrumentation

To accurately describe the machine dynamic behavior a high-fidelity model is created in MATLAB SimMechanics. Its mechanical system is important due to the coupling imposed by each of the connected components upon the entire structure. This model is coupled with the hydraulic model where Simulink is used to calculate the forces and torques generated by the hydraulic system and SimMechanics feeds back the resulting dynamics (i.e. actuator positions and velocities). Additionally, a nonlinear model of the engine is created in the same interface to replicate its dynamics. All together, the models allow for the determination of the loads imposed on the engine through the hydraulic units and their effect on its rotational speed.

Hydraulic System Model

The implemented working hydraulics of the excavator prototype are shown in Figure 4. It can be observed that the excavator comprises 8 actuators; however, for this study, only the digging operation has been taken into account. Therefore, the distributing manifold valves 2, 3, 6, 7, 11, 12, 17 and 18 remain open while the rest are closed. An important assumption for the hydraulic system model is that the pressure losses across on/off valves shown in Figure 4 between the hydraulic units and the linear actuators are negligible since they remain open. Additionally, since in this study the controller is tested using a truck loading cycle, only the swing, boom, arm and bucket actuators are simulated. The load actuator dynamics of each linear actuator can be expressed using the pressure buildup equation according to the DC system shown in Figure 2.

$$\begin{aligned}\dot{p}_1 &= \frac{K}{V_1} (Q_1 - Q_s + Q_{chk1} - A_1 \dot{x} - Q_{Li} - Q_{r1}) \\ \dot{p}_2 &= \frac{K}{V_2} (-Q_2 + Q_s - Q_{chk2} + A_2 \dot{x} + Q_{Li} - Q_{r2})\end{aligned}\quad (1)$$

where p_1 and p_2 are the actuator bore and rod side pressures, V_1 and V_2 are the fluid volumes in either actuator's chambers, which are given by $V_1 = V_{dead} + A_{bore}x + V_L$ and $V_2 = V_{dead} - A_{rod}(x_{max} - x) + V_L$ where V_{dead} is the assumed cylinder dead volume, A_{bore} and A_{rod} are the bore and rod side areas respectively, x is the actuator displacement and its time derivative denotes the actuator velocity, x_{max} is the maximum actuator displacement, and V_L is the line volume. Also, K is the effective bulk modulus, Q_{Li} is the internal leakage flow across the actuator chambers and Q_{r1} and Q_{r2} are the relief valves' flows, which may be obtained as

$$Q_{r1} = \begin{cases} C_r (p_1 - p_{setting}) & \text{for } p_1 > p_{setting} \\ 0 & \text{for } p_1 < p_{setting} \end{cases}\quad (2)$$

$$Q_{r2} = \begin{cases} C_r (p_2 - p_{setting}) & \text{for } p_2 > p_{setting} \\ 0 & \text{for } p_2 < p_{setting} \end{cases}\quad (3)$$

where C_r is a linear flow coefficient and $p_{setting}$ is the relief valve pressure setting. The pump flows, Q_1 and Q_2 , as well as their volumetric losses, Q_s , are obtained based on the unit's speed, displacement and differential pressure. Finally, Q_{chk1} and Q_{chk2} are the PO check valves' flows in line 1 and 2. For line 1 Q_{chk1} may be expressed according to Figure 5 and given by Eq. (4)

$$Q_{chk1} = \left(\alpha_D \pi d_{PO} \text{sign}(p_{lp} - p_1) \sqrt{2|p_{lp} - p_1|/\rho} \right) y_1\quad (4)$$

where α_D is the valve discharge coefficient, d_{PO} is the PO check valve spool diameter, p_{lp} is the low pressure system pressure, which in this case has been assumed to be a constant value, and y_1 is the PO check valve spool displacement which can be calculated by balancing the forces on the valve spool according to Figure 5. For the case of Q_{chk1} , the spool motion can be described by

$$y_{1 \text{ normal}} = \left(A_{PO} (p_{lp} - p_1) - F_{crack} \right) / k\quad (5)$$

$$y_{1 \text{ opposite}} = \left(A_{PO} (r_{PO} p_2 + (1 - r_{PO}) p_{lp} - p_1) - F_{crack} \right) / k\quad (6)$$

where A_{PO} is the PO check spool area, F_{crack} is the valve cracking force, k is the valve spring constant and r_{PO} is the valve pilot ratio. In this study, the PO check spool dynamics are neglected due to their fast nature. Spool stroke limits are set to prevent oscillations and avoid cavitation. The above derived expressions can be modified to characterize the PO check in line 2.

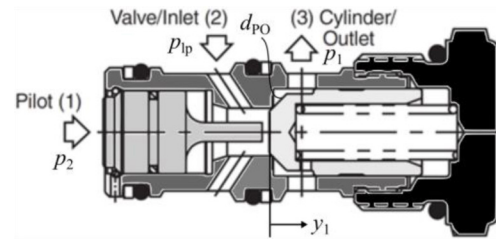


Figure 5. Pilot-operated check valve schematic

To model the hydraulic hybrid accumulator state of charge, its dynamics can be expressed using the pressure build-up equation as

$$\dot{p}_{hp} = \frac{1}{C_H} (n_1 V_1 \beta_1 - Q_{s1} - \dot{\phi}_{TOT} V_2 \beta_2 - Q_{s2})\quad (7)$$

where n_1 is the primary unit speed, V_1 , V_2 , β_1 and β_2 are the primary and secondary units' maximum volumetric displacements and normalized displacements respectively, ϕ is the excavator cabin angular position and its time derivative denotes its angular velocity and i_{TOT} is the total gear ratio between the secondary unit pinion and cabin ring gear. It is important to mention that the hydraulic units' dynamics are governed by the dynamics of their control valves, which are of second order [19]. Finally, Q_{s1} and Q_{s2} are the primary and secondary unit's volumetric losses, which may be obtained through the four-quadrant unit operation map, and C_H is the hydraulic capacitance which may be defined as

$$C_H = \frac{V_o}{N} \left(\frac{p_o^N}{p_{hp}^{N+1/N}} \right)\quad (8)$$

where V_o is the accumulator gas volume, N is the assumed ideal gas polytrophic exponent, $N = 1.25$, and p_o is the accumulator pre-charge.

Dynamics Model

The excavator structural components' geometrical properties are shown in Figure 6 and details are provided in Table 1. Additionally, the coordinates for each components' center of gravity as well as their mass properties are given in Table 2. Finally, each component inertia tensor is given by Eq. (9), Eq. (10) and Eq. (11). The excavator cabin which mass is 1,955 kg, is simulated with its rotational axis referenced to the ground and a single degree of freedom.

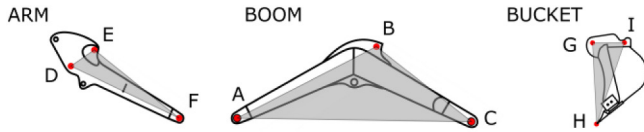


Figure 6. Excavator components physical dimensions

Table 1. Structural components geometrical dimensions

Point	x (m)	y (m)	z (m)
A	0.000	0.000	0.000
B	2.641	0.000	0.000
C	1.568	0.819	0.000
D	-0.298	0.229	0.000
E	0.148	0.311	0.000
F	1.371	0.000	0.000
G	0.000	0.000	0.000
H	-0.024	-0.881	0.000
I	0.220	0.000	0.000

Table 2. Center of gravity coordinates and mass properties

Component	x _{CG} (m)	y _{CG} (m)	z _{CG} (m)	mass (kg)
Bucket	0.024	-0.292	0.000	202
Arm	0.516	0.096	0.000	120
Boom	1.395	0.361	0.000	190

$$\mathbf{I}_{CG \text{ boom}} = \begin{bmatrix} 14.133 & 1.162 & 0.119 \\ 1.162 & 115.423 & 0.0418 \\ 0.119 & 0.0418 & 127.309 \end{bmatrix} \text{ kg} \cdot \text{m}^2 \quad (9)$$

$$\mathbf{I}_{CG \text{ arm}} = \begin{bmatrix} 2.086 & 2.797 & -0.025 \\ 2.797 & 36.272 & 0.005 \\ -0.025 & 0.005 & 37.539 \end{bmatrix} \text{ kg} \cdot \text{m}^2 \quad (10)$$

$$\mathbf{I}_{CG \text{ bucket}} = \begin{bmatrix} 13.471 & 5.386 & -0.058 \\ 5.386 & 19.943 & 0.017 \\ -0.058 & 0.017 & 21.358 \end{bmatrix} \text{ kg} \cdot \text{m}^2 \quad (11)$$

Since the proposed ARC nonlinear controller is based on the plant's knowledge, a detailed mathematical model of the swing drive system is created. The secondary unit dynamics can be expressed in terms of the cabin torques according to Figure 7 and Eq. (12).

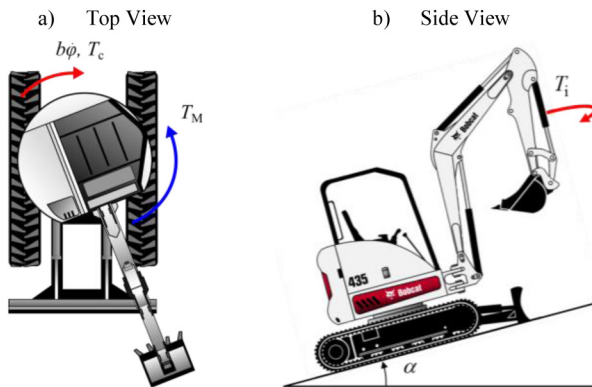


Figure 7. Free-body diagrams of the excavator top and side views

$$J(l_{\text{arm}}, l_{\text{boom}}, l_{\text{bucket}}) \ddot{\varphi} = V_M dp i_{\text{TOT}} \beta_M - b \dot{\varphi} - T_c \text{sign}(\dot{\varphi}) + T_i(\alpha) \quad (12)$$

where $J(l_{\text{arm}}, l_{\text{boom}}, l_{\text{bucket}})$ is the inertia of the excavator cabin, which is a function of the arm, boom and bucket actuator lengths l_{arm} , l_{boom} , and l_{bucket} , φ is the excavator cabin angular position with its first and second time derivatives being the cabin velocity and acceleration, V_M is the hydraulic motor maximum volumetric displacement, dp is the hydraulic motor differential pressure, i_{TOT} is the total gear ratio between the hydraulic motor pinion and the cabin ring gear, β_M is the hydraulic motor normalized displacement, b is the viscous friction coefficient, T_c is the Coulomb friction coefficient and T_i is the torque induced by the weight of the excavator cabin on an incline, which is a function of the incline angle α , as shown in Figure 7 b).

Engine Model

The engine under study is a naturally aspirated 36.5 kW Kubota stock diesel with a maximum rated speed of 2700 rpm. The engine dynamics, are given by

$$\dot{n}_E = \frac{1}{J_E} (M_{E \text{ eff}} - M_L) \quad (13)$$

where n_E is the engine speed, J_E is the engine inertia, $M_{E \text{ eff}}$ is the effective engine torque output which can be expressed as $M_{E \text{ eff}} = M_{E \text{ th}} - M_f$, where $M_{E \text{ th}}$ is the engine theoretical torque output and M_f is the engine torque loss due to friction. Finally, M_L is the load torque imposed on the engine, in this case by the hydraulic system. It is important to note that the maximum theoretical engine torque, $M_{E \text{ th}}$, can be obtained from measured data of the engine wide-open-throttle (WOT), which includes friction. So, to get the maximum theoretical engine torque, the WOT measurements must be modified to remove the measured friction as

$$M_{E \text{ th}} = u_E (M_{\text{WOT}} + M_f) \quad (14)$$

where u_E is the normalized engine control input at any operating engine speed and M_{WOT} is the measured WOT curve torque. The engine friction on the other hand is modeled based on published empirical data [20] of the effective mean pressure as

$$p_{\text{me}} = 75 + 0.048 n_E + 0.4 S_p^2 \quad (15)$$

Knowing that the effective mean pressure can be expressed as

$$p_{\text{me}} = 2\pi M n_E / V_{\text{eng}} \quad (16)$$

where V_{eng} is the engine displacement in liters and M is the torque resulting from the mean effective pressure, the torque loss due to friction can be expressed as