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Development and Validation of a Mini Excavator Model: A Numerical Tool for Hydraulic and Control System Design

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Abstract. This work stems from the need in the fluid power industry to achieve energy and cost savings while testing different design and control alternatives in work vehicles. One way to achieve this goal is to include more modeling and simulation activities in the industrial process of system and component design. However, developing a model that is detailed, reliable, and characterized by reasonable computational times is a complex task. In this context, this paper describes the process followed to obtain a numerical tool capable of replicating the behavior of a reference mini excavator, which can be used for different purposes in the design and testing phase by the company involved in the project. The same modeling process can be also used as guidelines by other industrial engineers and researchers involved in the development of similar tasks. In this work, a lumped parameter model of the hydraulic system is thus developed exploiting the commercial software Simcenter Amesim®. The modeling focuses on the fluid power generation group, which includes a variable displacement swash plate pump, a detailed representation of the main valves, and the arm actuators. The two main valves in the system are a post-compensated directional valve and a displacement regulator. The boom movements (boom, arm, and bucket) are modeled using both 2D and 3D multi-body approach to ensure better correlation with field data. Dedicated field tests, including a complete digging cycle, were in fact conducted in collaboration with the company to obtain data for the experimental-numerical comparison and the consequent validation of the model and the load sensing control logic tested. The validated model allows us to obtain a numerical tool capable of replicating the system behavior with different operating parameters, different valves or exploring alternative control strategies using a dedicated interface that enables seamless integration with MATLAB Simulink®. An example of the model application is also presented, obtained by virtually testing two different control strategies for the pump displacement.

Nomenclature

Roman letters

$\vec{a}$	Acceleration vector [m s^{-2}]
A_{act}	Piston area for displacement regulation [m^2]
B	Bulk modulus [Pa]
$\vec{F}$	Force vector [N]
F_{act}	Piston force for displacement regulation [N]
F_k	Spring force [N]
F_{pist_y}	Piston thrusting force [N]
J	Moment of inertia [kg m^2]
k	Spring rate [N m^{-1}]
l	Spring length [m]
l_0	Spring free length [m]
m	Mass [kg]
$\overrightarrow{M(G)}$	Moment vector around point G [N m]
M_{pist_z}	Piston tilting moment acting on the swash plate [N m]
p	Pressure [Pa]
p_{reg}	Pressure regulated for displacement control [Pa]
Q_{in}	Flow rate entering the piston chamber [m^3 s^{-1}]
Q_{out}	Flow rate exiting the piston chamber [m^3 s^{-1}]
R_m	Piston pitch radius [m]
V	Piston chamber volume [m^3]
x_{pist}	Axial position of the *i-th* piston [m]
$z_{F_{pist}}$	Position of piston thrusting force [m]
$z_{F_{pist_{tot}}}$	Position of total piston thrusting force [m]

Greek letters

$\vec{\alpha}$	Angular acceleration [rad s^{-2}]
β	Swash plate angle [rad]
θ	Piston angular position [rad]
ω	Angular velocity of the pump shaft [rad s^{-1}]

Dimensionless groups

n	Piston number

Superscripts and subscripts

i	i-th piston of the pump
LS	Load sensing
P	Pump delivery
T	Discharge tank

1 Introduction

Fluid power systems for mobile applications are evolving into complex electro-hydraulic systems to meet performance demands while ensuring high efficiency to reduce fuel consumption, especially for thermal engine vehicles. However, more efficient fluid power systems are helpful also in sustaining the electrification or hybridization of the vehicle. Consequently, it is increasingly important to assist fluid power companies in their effort to design more efficient components and systems with analysis, studies and dedicated tools as well as to propose and explore novel circuit architectures and control strategies that enhance both performance and efficiency. In recent years, numerous research works have been published on these topics, some of them focusing on the attempt to reduce losses of a typical load sensing architecture used in many mobile applications, (e.g., [1, 2, 3, 4, 5]), others exploring different circuit architectures (e.g., [6, 7, 8, 9, 10, 11, 12]).

The article focuses on the boom actuations of a mini excavator as it was already available at Walvoil S.p.A. for on-field test, but the same process can be applied to other vehicles implementing similar systems. The standard design process used by the company involves the experimental characterization of the vehicle behavior, where new components and/or new control strategies are implemented. However, basing the entire design process using experimental tests can be time and cost consuming. In this context, this article describes the modeling process adopted to create a robust virtual tool that can be used for different purposes supporting the design and testing phase. For example, it is possible to explore the impact on the system behavior of different valve configurations, control strategies or operating parameters. The development of a model that is both detailed and, at the same time, robust, reliable, and characterized by reasonable computational times is however a complex task. The study presented in the article thus focuses not only on creating this numerical tool, but also on establishing a set of guidelines for other industrial engineers and researchers involved in the development of similar models. Finally, the article also shows as example the results of an application of the numerical model presented, obtained by implementing and comparing two different control strategies for the pump displacement.

The virtual tool should have the following intrinsic characteristics to offer an alternative approach for developing this design phase using simulations:

- It should preferably be a multi-physical model to consider the interactions between the different subsystems. The mini excavator model, for example, should integrate the arm and its movements, the hydraulic system that supplies power and controls the vehicle movements, and finally the electronic control unit that monitors and regulates the hydraulic system according to specific algorithms.
- It should be validated against experimental data to ensure reliability.
- It should be computationally efficient to achieve execution times smaller than those required for experimental activities.

Other features must also be considered, depending on the preferences and expertise of the designer using the virtual tool:

- The modeling approach (e.g., representing the nonlinear dynamic model of the entire system [10, 13, 14, 15], linearizing the system or a critical portion of it [16, 17, 15], or using a mixed approach, involving black-box or grey-box modeling [18, 19]).
- The preferred modeling environment.
- The ease of model parameterization, which is a critical phase that is never inherently complex but may discourage the inexperienced users.

In this study, we chose to adopt a lumped parameter approach to develop the model in collaboration with the company involved. For the hydraulic circuit, we used functional models for components considered non-critical in determining the dynamic behavior of the system. We instead created physical detailed models for those components that are critical, especially those that interact with the control logic algorithm. The mechanical arm was represented using both a 2D and a 3D multi-body model. The control algorithm for the electro-hydraulic components was implemented according to the control strategy defined by the company. Simcenter Amesim [20] is the simulation environment chosen. This is a commercial software that allows detailed modeling of multi-physical systems, like the one considered in this study. Using this software, in fact, it was possible to develop lumped parameter models of the hydraulic circuit and the mechanical arm to be finally integrated with the control logic. The model also must be suitable for co-simulation with MATLAB Simulink [21] which is the environment in which the company directly develops the complex logics for the control of their electro-hydraulic components.

At the end of this activity, the company has access to a simulation platform that enables the selection of different simulation approaches, depending on the test objective and the time available. The lumped parameter approach for these types of systems is considered the state of the art and has proven to be a robust and relatively fast method for simulating the dynamic behavior of hydraulic and hydro-mechanical systems. Of course, model validation is necessary. However, achieving a perfect match between experimental and simulated behavior is challenging due to inherent simplifications in the model and uncertainties in defining some key parameters. These include, for example, friction contributions in kinematic-mechanical models and flow parameters in metering passages of valves, ports, and orifices ([22, 23]).

Nevertheless, it is more practical to focus on reliability and computational efficiency over absolute accuracy when the aim of the model is to support the design and the initial validation of the control logic of the systems. The following section describes the baseline architecture of the hydraulic circuit used to control the arm of the mini excavator. Then the model of the hydraulic circuit and the mechanical arm are presented followed by the description of the interface with the control logic initially implemented (pressure-based control) Finally, we present the comparison with experimental data from the field, that validates the model in the context of the purpose described. At the end, new opportunities for the control of the system are presented (flow matching) and compared with the baseline pressure-based logic.

2 Mini Excavator Hydraulic System

As anticipated, the hydraulic circuit of the system considered in this study consists of three main groups of components:

- Fluid power generation group.
- Hydraulic valve for pump displacement regulation.
- Distributor block and hydraulic actuators for boom movements.

The main element of the generator group is a Walvoil *PWLS53* [24], which is an axial piston swash plate pump with variable displacement. The reference system integrates into the pump a load sensing logic providing the desired flow rate for several users, connected in a parallel architecture, as a function of the opening of the different sections in the distributor block. The pump flow rate does not depend on user pressures, but it is regulated to maintain a target pump margin $p_P - p_{LS}$ [25].

The load sensing architecture, and so the pump displacement regulation, can be realized with a differential pressure regulator or with an electro-hydraulic regulator. The differential pressure regulator can be purely mechanical, where the pump margin is fixed and defined by a spring [25], or electronic, where a variable pump margin can be defined by the operator. An example of these latter systems is *Adaptive Load Sensing* (*ALS*) by Walvoil [26]. However, the system considered integrates a prototypical electro-hydraulic regulator, called *AFC* (Fig. 1), where the control moves the main spool according to the desired pressure to be conveyed to the pump displacement actuator replicating a load sensing logic. This valve and its functioning are described in more detail in the section dedicated to its modeling.

The actuators are controlled using the Walvoil *DPX* [27], which consists of post-compensated proportional directional valve, operated via a joystick in the vehicle cabin. *DPX* allows to control the velocity of the actuator through the opening (flow area) of the directional valve and independently of the load on the actuator. Moreover, the actuators may need to move simultaneously, under different loads. For this reason, a local compensator is provided after the metering section in the directional valve. Post-compensation ensures the management of the operating condition of the flow saturation. The three actuations studied in the project are the lift movement (boom), the extension or retraction movement (arm) and the tilt of the bucket (bucket). Figure 2 presents the *DPX* hydraulic scheme highlighting the three sections of interest for the movements of the mini excavator arm.

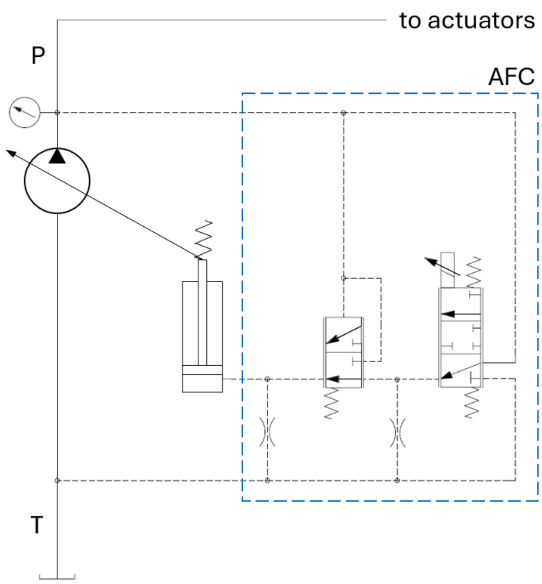

Figure 1: AFC regulator valve scheme for displacement control

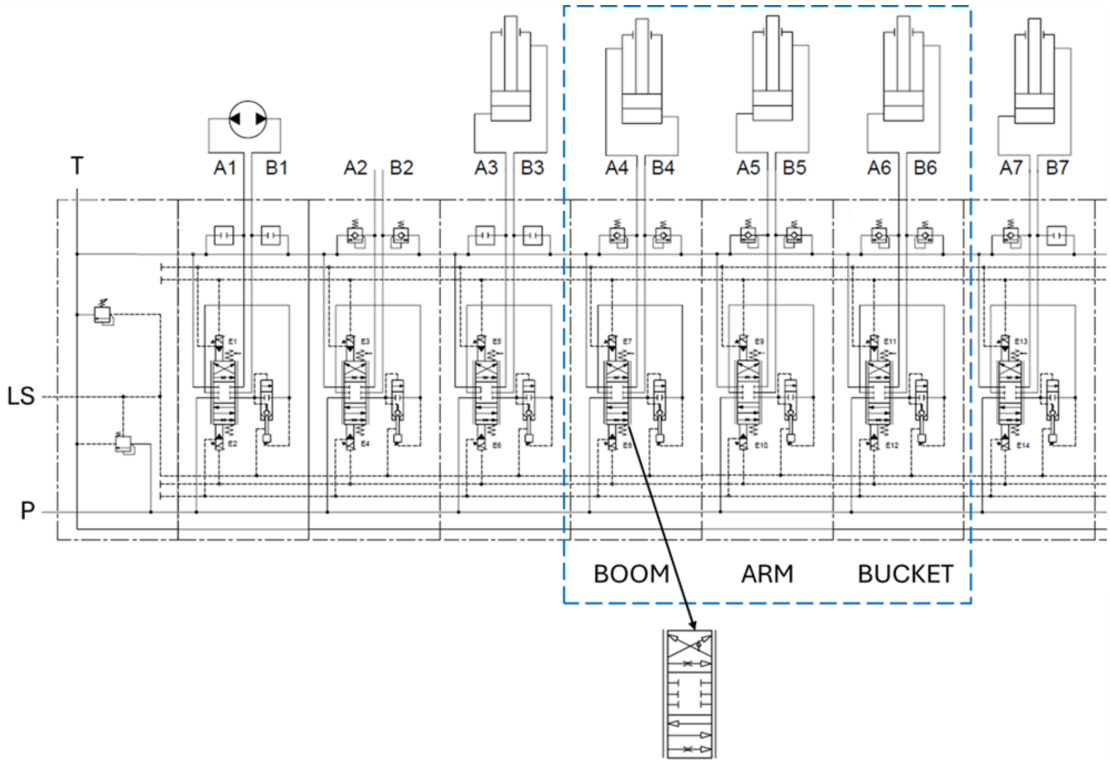

Figure 2: DPX directional valve scheme for the movements of mini-excavator arm, with boom, arm and bucket sections highlighted

3 System modeling details

This section provides the modeling details of the main components of interest of the system, including the variable displacement pump, the displacement regulator, the directional proportional valve, and the excavator arm actuators.

3.1 Variable displacement swash plate pump - PWLS

We considered the *PWLS53* pump installed on the reference mini excavator. This is a variable swash plate axial piston pump with nine pistons and a maximum nominal displacement of 53 cc/rev at 18.3° of swash plate inclination. Note that it was not mandatory to model all the aspects of this machine in detail, as considering a heavy-detailed model for system simulations, in fact, may lead to very long simulation time. We thus modeled only the aspects of interest.

One of these aspects was the replication of the pump flow rate and so the piston chamber variable volumes, starting from the shaft speed and the swash plate angle. The piston chamber variation, which depends on the i-th piston stroke x_{pist_i} (Eq. (1) [28]), must be interfaced with the correct opening and closing of the suction and delivery kidney ports. These latter were modeled as equivalent orifices with lookup tables for their correct opening as function of the i-th piston angular position.

$$x_{pist_i} = -R_m \cdot \tan\beta \cdot (1 - \cos(\theta_i - \pi)) \tag{1}$$

Equation (2) provides the pressure buildup equation used for the evaluation of the pressure within the i-th piston chamber. We replicated this model for each of the nine pistons of the pump.

$$\frac{dp}{d\theta_i} \cdot \omega = \frac{B}{V} \cdot \left(Q_{in} - Q_{out} - \frac{dV}{d\theta_i} \cdot \omega\right) \tag{2}$$

It was also mandatory to model the swash plate oscillation to simulate the pump dynamics while increasing/decreasing the displacement. This swash plate rotation is controlled by the regulation piston and consequently by the displacement regulator valve. The main forces/moments acting on the swash plate dynamic equilibrium are:

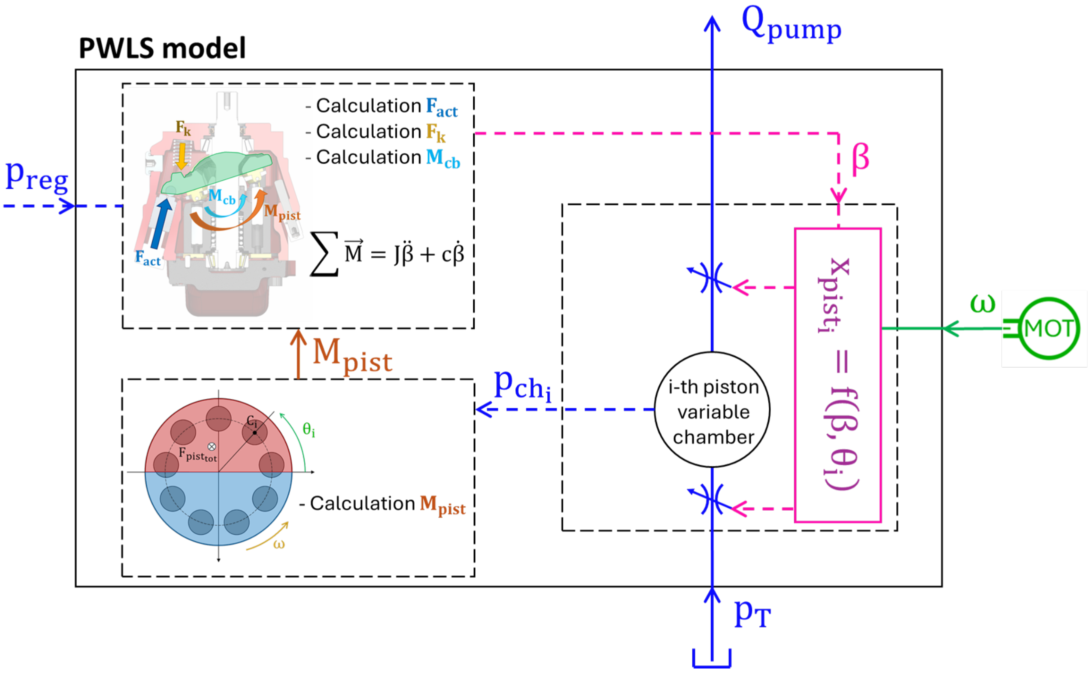

Figure 3: Block diagram of the PWLS pump model

- Actuator force, given in Eq. (3) where A_{act} is the actuator piston thrusting area. The regulator must correct its opening (hydro-mechanically or electrically depending on the type of regulator) according to the desired pressure p_{reg} to obtain/maintain the correct swash plate angle of the pump.

$$F_{act} = p_{reg} \cdot A_{act} \tag{3}$$

- Spring force, expressed in Eq. (4), where l_0 is the free length of the spring and l is the length of the spring while the swash plate is tilting. This force pushes to maximum displacement when the actuator force is zero in all the swash plate angular positions.

$$F_k = k \cdot (l_0 - l) \tag{4}$$

- Cylinder block reaction moment. This moment could be calculated from the multibody dynamic analysis of the unit considering the various bodies interfacing with each other. However, we decided to simplify this calculation by exploiting a parametric and empirical lookup table to evaluate this term.
- Piston reaction moment, provided in Eq. (5). This is the reaction moment exerted by the n pistons resulting from the different pressure acting on the suction and delivery ports. This moment can be evaluated considering an equivalent total piston thrusting force and its position or, once again, using empirical lookup table. Note the position of the total piston force is not constant during the shaft rotation [28].

$$M_{pist_z} = \sum_{i=1}^{n} \left(F_{pist_{y_i}} \cdot z_{F_{pist_i}}\right) = \sum_{i=1}^{n} F_{pist_{y_i}} \cdot z_{F_{pist_{tot}}} \tag{5}$$

Figure 3 shows the block diagram of the PWLS pump model.

3.2 Displacement regulator valve - AFC

Simcenter Amesim also enables the precise physical modeling of hydraulic system components using the *Hydraulic Component Design* (*HCD*) library [29]. This approach allows to represent the valves with a high level of detail in each part. The *HCD* library includes, in fact, the basic building blocks of any hydro-mechanical system, such as piston thrusting surfaces, leakages or poppets with various geometries.

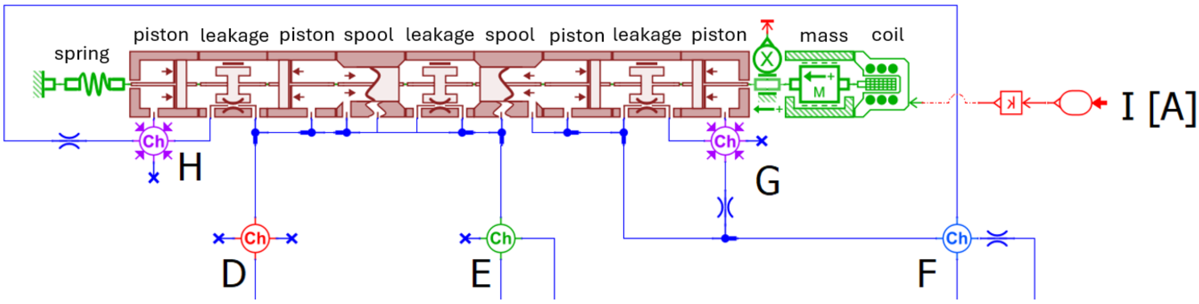

Figure 4: Simcenter Amesim model of AFC valve

It was thus mandatory to divide the regulator spools and hydraulic ports into main representative blocks. The combination between *HCD* elements and elements of *Hydraulic* and *1D Mechanical* libraries led to the detailed modeling of this regulator valve, as can be seen in Fig. 4.

AFC valve does not operate as a conventional pressure regulator. In fact, the valve receives the input current signal, which, in the reference load sensing logic, actuates the spool to adjust the pump margin and maintain it at the desired value. The regulated pressure is thus obtained controlling the spool position and so opening/closing the connection with the supply and discharge lines. The spool undergoes frequent and minor displacements since the correction occurs in a closed-loop system, responding immediately to small variations in the system. For this reason, the *AFC* spool requires specifically designed grooves so that even small displacements correspond to desired opening and closing. Note that it was possible to incorporate these geometric details using the *HCD* library blocks labeled as “spool” in Fig 4.

After defining these main feature elements mentioned, the CAD model of the valve helped us to set the correct values of more or less all the geometric parameters of the Amesim valve model. However, the tuning of some other information (e.g. frictions, flow forces, but also geometric tolerance for diametral clearance) is not easy as for the unambiguous geometric ones. It was thus mandatory to exploit the results of previous experimental tests to align these unknown parameters. We replicated the experimental tests in the Amesim environment and modified the unknown parameter until reaching a good results alignment between the experimental and numerical results to validate the *AFC* valve.

3.3 Proportional directional block - DPX

The same modeling approach using the Amesim *HCD* library was used to develop the model of the proportional directional valve *DPX*. The valve scheme shown in Fig. 2, in fact, consists of different sections with different spool electronically controlled, each of which manages the movements of an actuator of the mini excavator arm. The correct modeling of every feature of the sections is thus mandatory to realize a consistent model of the directional valve. The development, parameterization, and validation of the *DPX* model was achieved in a previous project [30].

3.4 Mini excavator arm actuators

The Bobcat 435 mini excavator is the reference vehicle for the presented study, whose mechanism main parts are depicted in Fig. 5. We developed the model of the excavator arm using the mechanical multibody approach of the Amesim software. In particular, we created two models as follows:

- The 2D model of the excavator arm was created with the Amesim *2D Mechanical*, where we considered only the movements of the arm on its plane (no cabin rotation nor arm swing).
- The 3D model of the excavator arm was created with the Amesim *3D Mechanical* library, where we considered the movements of the arm but also its swing and the cabin rotation.

The *2D Mechanical* library can be used for any type of 2D mechanical system using bodies with 3 degrees of freedom, pivot joints, translation joints and/or a combination of both. The mechanical library is based on the mathematical constraint equations from mechanics. The body submodels use differential equations to calculate the generalized coordinates. The joint submodels use the Baumgarte stabilization schemes applied to the geometric, kinematics and acceleration constraint equation [29]. Similarly, the *3D Mechanical* library models the 6 degrees-of-freedom kinetic and dynamic behavior of multibody systems in 3D space. The library consists of rigid or flexible bodies connected by functional junctions [29].

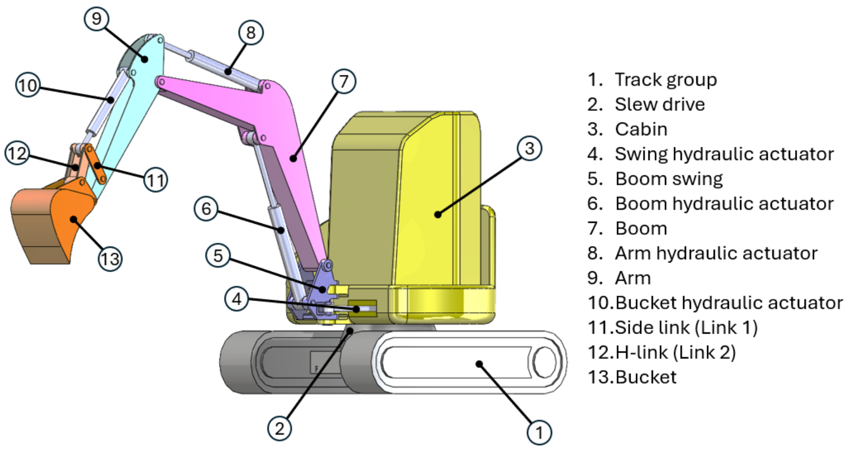

Figure 5: Main parts of the mini excavator mechanism

The modeling approach is the same in both cases. The first step is the identification of every rigid body of the 2D/3D structure and its connection ports with the others through the appropriate joint. Second, it is mandatory to set all the correct inertial (masses and moments of inertia) and geometrical parameters of the model. The multibody approach requires information concerning the geometrical position of each CoG (Center of Gravity) and connection port for every rigid body. We obtained all the geometrical and inertial information from the 3D CAD model of the reference mini excavator. The Simcenter Amesim tool named *2D Mechanical Assistant* and *3D Mechanical Assistant* helped to visualize the multibody system both during the model development and animated the mechanical system after the simulation.

The fundamental equations underlying the modeling of the mechanical part of the mini excavator can be traced back to Newton's second law of motion [31]. The linear and angular positions, velocities and accelerations of rigid bodies can be found based on the forces and torques applied to them, as given in Eq. (6) and (7).

$$\sum_{i=1}^{j} \vec{F_i} = m\vec{a} \tag{6}$$

$$\sum_{i=1}^{j} \overrightarrow{M(G)_i} = J\vec{\alpha} \tag{7}$$

Both these models can be coupled with the other components described in the previous subsections. Of course, the 2D model is less complex than the 3D one, as it can be reduced only to the movement of the excavator arm, without considering the cabin, the track group, the boom swing and the boom swing actuator.

Since only the boom movements of the mini excavator were considered for the experimental tests and the subsequent model validation, we adopted the 2D mechanical model.

4 Matlab-Amesim interface for control

Simcenter Amesim allows to create an interface to perform simulations with a combination of Amesim and Matlab Simulink models. Since there are two software packages involved, the interfaces provide three main options: importing the Amesim model into Simulink, importing the Simulink model into Amesim or performing a co-simulation [29].

This Amesim feature thus allowed us to couple the multi-environmental model of the mini excavator with a control logic developed directly into Matlab Simulink without translating it into the Amesim sketch using the *Signal, Control* library [29]. In this way, it is also possible to make the control logic more similar to the actual algorithm acting on the real system. Although the control logic discussed in this work is based on simple PID controllers that could also be implemented using the Amesim *Signal, Control* library, setting up the co-simulation interface with Simulink can be particularly useful from an

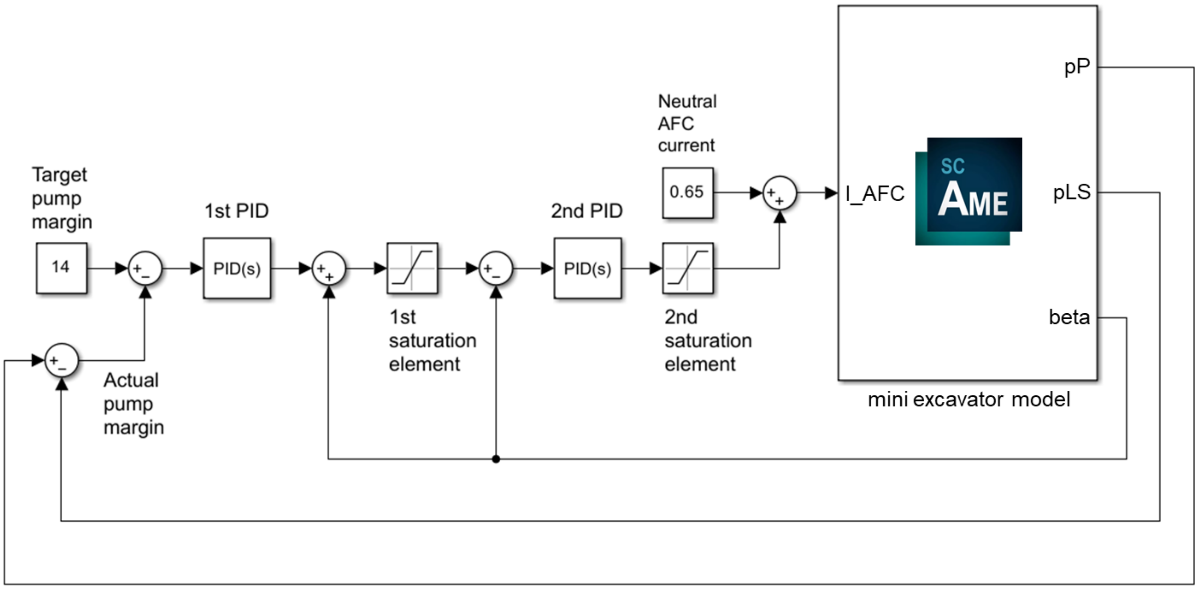

Figure 6: Simulink model using the Amesim block with pressure-based control logic

industrial perspective. In fact, the company involved implements and tests control algorithms directly in Simulink, making this setup aligned with their standard workflow and allowing for integration of more advanced control strategies in future developments.

Figure 6 presents the Simulink sketch with the Amesim block (representing the Amesim model of the considered hydro-mechanical system with the correct Amesim/Simulink interface block) connected to the pressure-based control logic replicating a load sensing system.

The Simulink control logic consists of the following elements:

- A first PID controller taking as input the difference between the target pump margin p_{margin} of the system, defined by the user, and the actual current pump margin $p_P - p_{LS}$. The output of the PID is the adjustment on the swash plate angle needed $\Delta\beta$.
- A second PID controller taking as input the difference between the adjusted angle of the swash plate $\beta + \Delta\beta$, and the current swash plate angle β. The output of the PID is the adjustment on the AFC valve current, ΔI_{AFC}. The sum of this correction with the neutral AFC current is the actual current to be supplied to the AFC valve.
- A first saturation element before the second PID. This element considers the mechanical end strokes for the swash plate angle, avoiding the adjusted swash plate angle to reach unfeasible values.
- A second saturation element after the second PID. This element limits the output correction on the AFC current to desired limited values.

It was also mandatory to select appropriate Simulink simulation settings to obtain consistent results. After a tuning phase, we selected the settings as provided in Table 1.

Table 1: Simulink simulation settings

Variable-step solver	Minimum step size	Relative tolerance
ode15s	1e-8	1e-5

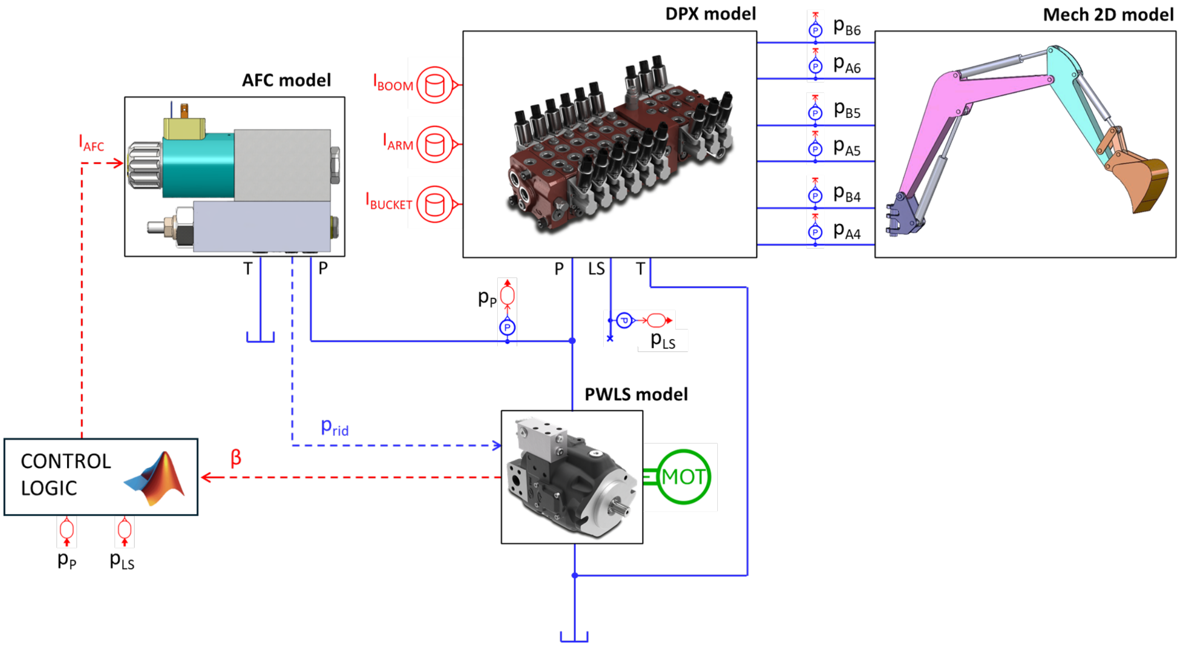

Figure 7: Block diagram of the complete system model

5 Comparison with experimental results and model validation

After modeling and parameterizing the individual components of the hydraulic circuit, it was possible to assemble the complete model of the system considered, as shown in Fig. 7. The final model includes the following blocks:

- Model of *PWLS* pump with piston chambers.
- 2D mechanical model of the mini excavator arm.
- Model of displacement regulator *AFC*.
- Model of directional valve *DPX*.
- *AFC* pressure-based control logic.

Note that the lumped parameter approach allows to model the components as interconnected blocks, each defined by parameters that determine their properties and interactions. The model can be thus used to test the impact of different parameters, components or even control logic on the system by simply varying the input values of the individual blocks.

The goal for the creation of the complete model was the improvement of the model accuracy without significantly increasing its complexity, finding a good compromise between accuracy and simulation time. For this reason, on-field tests were conducted on the reference mini excavator to support the model alignment process and test its actual functionality. The vehicle performed no-load excavation cycles, and movement of individual actuators, i.e. boom, arm and bucket. The mini excavator was equipped with the following sensors also shown in Fig. 7 to obtain the necessary alignment data:

- p_{A4}, p_{B4}, p_{A5}, p_{B5}, p_{A6}, p_{B6} are the pressure sensors positioned at the directional valve ports. Note that $A4$, $B4$, $A5$, $B5$, $A6$, $B6$ refer to the ports of the *DPX* connected to the piston or rod chamber of the actuators of the mini excavator arm, as shown in Fig. 2.
- p_{LS}, p_P, p_T are respectively the pressure sensors positioned on the load sensing, supply and discharge line.
- β is the angular position sensor of the pump swash plate regulator.

The sampling rate used for pressure sensors, published on the CANbus line, was 10 ms. Data on the *DPX* opening currents, acquired every 5 ms, were also extracted directly from CANbus.

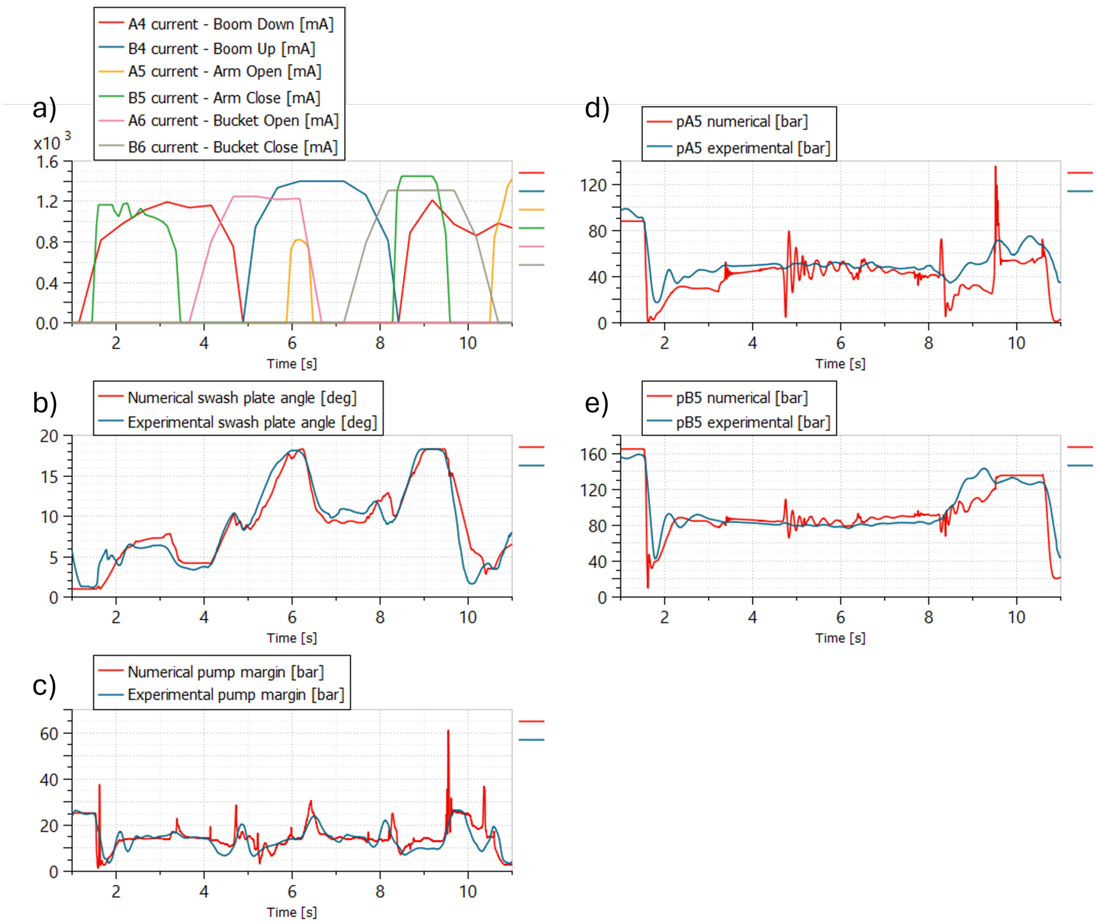

Figure 8: DPX input current (a) and comparison between Amesim results (red) and experimental data (blue): swash plate angle (b), pump margin (c) and pressure in arm actuator chambers, piston (d) and rod side (e)

Once we collected the alignment data, the next step was the validation of the model functionality by comparing simulation results with experimental measurements. We also used the data obtained from the tests covering the movements of individual actuators to parameterize their friction and inertia, ensuring better alignment. In contrast, the complete excavation cycle was used for the final comparison to validate the model accuracy. We used the current signals from the directional valve opening as inputs to the *DPX* model, enabling the reproduction of the digging cycle performed by the operator on the machine. Moreover, the target pump margin was set to match the one used on the mini excavator during the tests (i.e. 14 bar). The control logic, in fact, must ensure maintaining a constant pump margin during the cycle, as the system must operate replicating a load sensing, serving as a key parameter for verifying the reliability of the numerical model.

Figure 8 shows the result comparison performed on the complete excavation cycle.

Graph *c* of Fig. 8 shows how the numerical pump margin closely matches the actual one. This result confirms that the system operates correctly according to the load sensing logic, thanks to the control acting on the *AFC* controller. Similarly, graphs *d* and *e* show a good, but not perfect, alignment between with the experimental data concerning the pressure values reached in the actuator chambers, exhibiting, however, larger oscillations in some points. These discrepancies are due to the complexity of modeling multi-physical systems using a lumped parameter simulation approach. Specifically, the differences in these pressure trends are attributed to the challenge of adequately parameterizing the loads acting on the actuators and the system frictions. Additionally, the numerical model is more sensitive to system variations, leading to wider oscillations in the variables compared to the real system. Figure 8 only provides the comparison for the arm actuator, as the other actuators show a totally similar behavior.

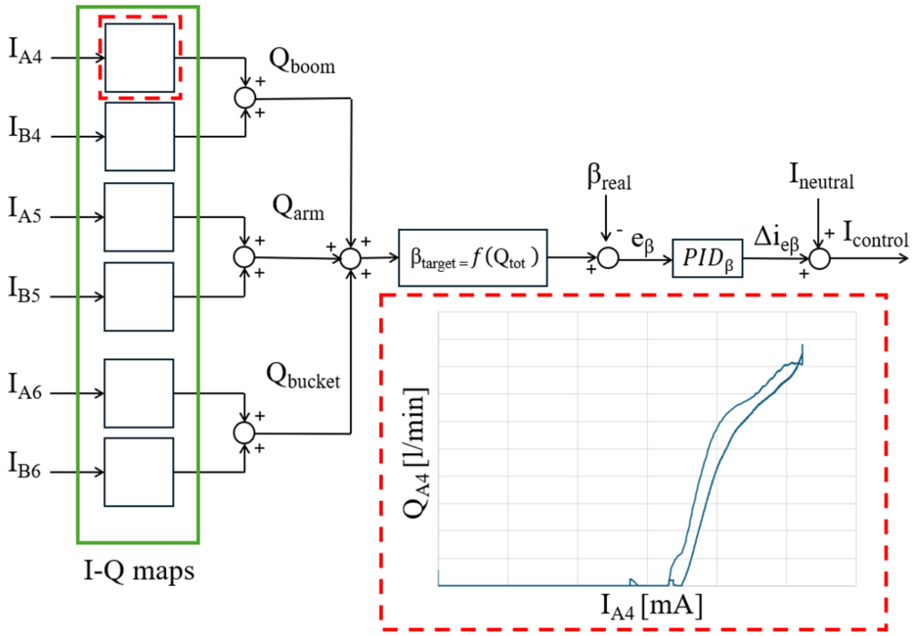

Figure 9: Flow matching control logic scheme with focus on boom down I-Q map

Although no direct displacement sensors were available to measure the actuator movements, the animation of the 2D mechanics visually confirmed that the motion performed by the numerical model corresponded to on-field movements. The alignment of the results presented, and the considerations made are deemed acceptable for the model validation.

6 Results and opportunities for control

Once we have obtained the complete model and verified its reliability, we present in this paragraph an example of its application achieved by comparing two different logics for the displacement control. This can be done by replacing the *AFC* controller with other types of valves or by modifying the control algorithm that operates on it. For example, the *AFC* can be replaced with the previously-mentioned *ALS* controller, which is an electro-hydraulic valve that allows the modification of the pump margin during system operation to achieve energy savings and optimized control [26, 11]. However, this option will not be further investigated in this paper.

The second strategy concerns the modification of the control logic that determines the operation of the *AFC* controller. With the aim of testing the robustness of the obtained simulation tool, we decided to modify the logic passing from pressure control to flow control, implementing a flow matching logic. Flow matching works by ensuring that the hydraulic pump supplies exactly the flow rate needed by the actuators at any given moment. This is achieved by dynamically adjusting the pump displacement based on real-time demand, in synchronization with the opening of the directional control valve. This dynamic adaptability minimizes the pressure losses and flow inefficiencies commonly encountered in load sensing systems, which struggle to maintain optimal flow rates under variable load conditions [32].

We first had to obtain maps linking the *DPX* opening current to the flow rate required by the actuators to implement this logic in the model equipped with the *AFC* controller. These maps were derived by extracting data from the simulations carried out on the complete model up to this point. Figure 9 shows the control logic and an example of these maps, where the input current, simulating the operator's control over the vehicle, is simultaneously converted into the flow rate required by the actuators. The new control logic then sums these flow rates to determine the total demand that must be supplied by the pump. The next step involved converting this flow demand into the required swash plate angle, allowing for correction based on the actual angle obtained in a closed-loop system. In this way, the angle is converted into the *AFC* regulation current following the just-presented logic and using the appropriate maps.

Figure 10 shows some of the results obtained by implementing this logic in the model, compared to the model working with the reference load sensing control. Note that Fig. 10 only provides the comparison for the arm actuator, as the other actuators show a totally similar behavior.

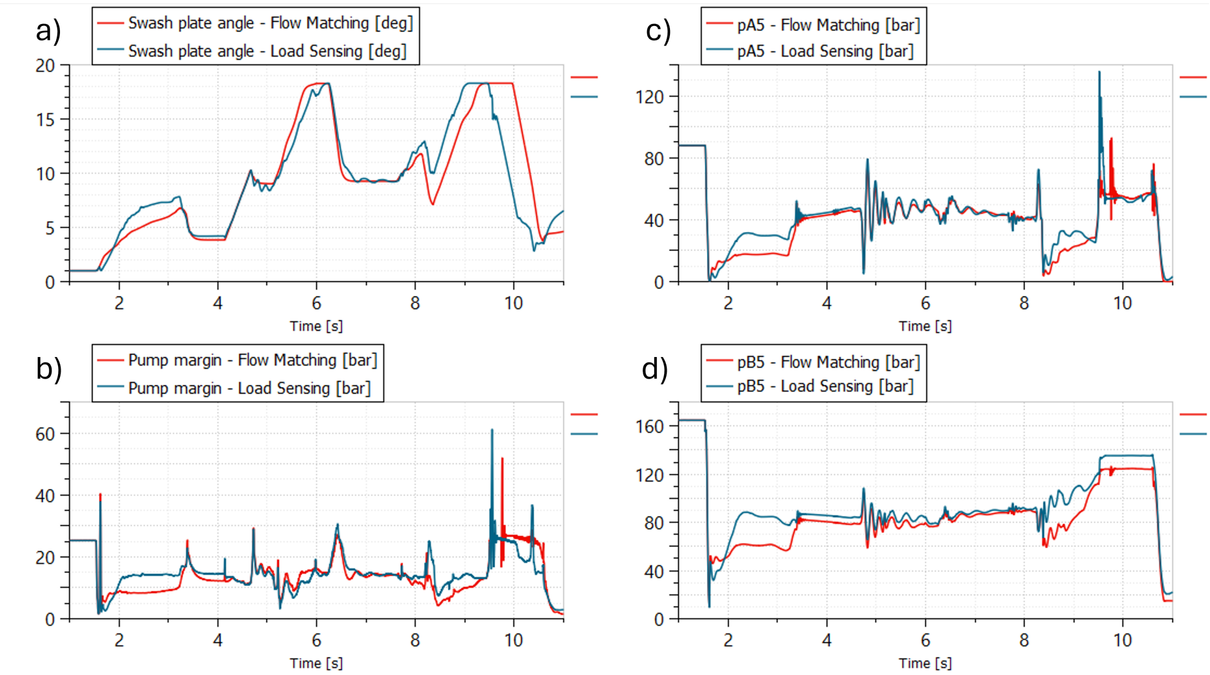

Figure 10: Result comparison between system with flow matching (red) and load sensing logic (blue): plate angle (a), pump margin (b), and pressure in arm actuator chambers, piston (c) and rod side (d)

It is easy to note how the system with the new control replicates the reference one, controlling the displacement in a similar manner. The main visible deviations are due to some differences between the two control methods. Firstly, flow control allows for a faster response compared to pressure control, so certain delays or advances in the swash plate angle response may be attributed to this reason. The slight variations in the pump margin between seconds 2 and 4, and between seconds 8 and 10 (graph *b* of Fig. 10) are attributed to increased regeneration during the boom-down movement, which was considered in the creation of the corresponding I-Q map. The spool detail at the bottom of Fig. 2, in fact, highlights the possibility of having a regenerative flow in the boom from the piston side to the rod one, with a check valve connecting the two actuator ports. Of course, this regenerative flow only occurs under certain conditions, but it can help to improve the overall efficiency. In conclusion, the flow matching strategy is capable of replicating the same reference duty cycle for the hydraulic system. Moreover, this different control logic may also be more efficient, since it promotes more fluid regeneration in some working phases.

This application example thus confirms the robustness and flexibility of the numerical tool just presented in this article.

7 Conclusions

We achieved a robust and flexible simulation tool that can support future developments, not only for the mini excavator application presented but also for component testing and control logic evaluation in this and other work vehicles. The model was first validated through comparison with experimental data, and subsequently tested with an alternative control logic, demonstrating its reliability and accuracy.

However, while the lumped parameter model is useful for simulating the hydraulic system, it also presents challenges, such as complex parameterization and unknown parameters (e.g., friction, discharge coefficient and flow characteristics). Even with model simplification, computational costs remain high, and it is still difficult to identify appropriate simplifications without misrepresenting the system. A linear analysis approach can be used to address these issues in future research. Adopting a linear approximation may help identify key parameters of the main components of the system and optimize their design. In fact, using linear analysis can also facilitate the application of optimization methods to enhance system performance by exploring different control algorithms.

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