MODELING AND SIMULATION OF AN OFFSHORE PIPE HANDLING MACHINE

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ABSTRACT

The objective of modeling the offshore machinery is to allow virtual prototyping and simulation of new designs which in turn makes it possible for engineers to test, redesign, adjust, and optimize a product before it is manufactured. An essential part of the modeling process is validation of the simulation results against real world measurements which in most cases is the limiting factor for achievable accuracy of the modeled system. In this paper we present a case study where a subsystem of an offshore drilling equipment is modeled and benchmarked with a full-scale machinery. Unlike some other works, the current study shows validation of the simulation results and friction identification process. We demonstrate that the model of the vertical pipe handling machine captures the important features of the real world system, and fundamentally improves computational effort of the simulation software as compared to a regular, multi-body model of the same equipment. Hence, the established model could be successfully applied in model based control system design as well as in real-time testing of control systems before commissioning of the equipment.
Keywords: Modeling and simulation, vertical pipe handling machine, electric drivetrain

NOMENCLATURE

c	Wire damping coefficient [N.s/m]
J	Mass moment of inertia [kg.m^2]
k	Wire stiffness coefficient [N/m]
m	Mass [kg]
T	Torque [Nm]
x	Translational displacement [m]
W	Wire force [N]
δ	Wire elongation [m]
$\dot{\delta}$	Wire rate of elongation [m/s]
θ	Rotational displacement [rad]
μ	Friction coefficient [-]

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INTRODUCTION

According to [1], there are strong signs that the hydrocarbon producing industry will tend to develop new fields in the Arctic in the near future. Oil and gas production in the Arctic depends on a complex set of variables. An increased demand for energy, high probability of finding resources and the fact that the Arctic ice is declining have fueled the race for resources. It is important to note, however, that harsh winters with extreme temperatures and year-round ice represent highly challenging conditions for the oil and gas industry. Therefore, it is observed that currently the petroleum enterprises invest in technologies which make exploratory drilling less difficult, more cost effective and environment friendly.

One of such fields is simulation based engineering. There exists a number of examples showing benefits of using modeling and simulation tools in the design, development, and analysis of offshore processes and machines - see for instance [3]-[7]. The overall tendency is not only to model a given phenomenon but also to utilize the model in either subsequent stages of product development or to gain better understanding of a considered problem. A good example is model predictive control (MPC) used in a closed loop simulation to improve process performance or model based optimization of a machine to provide for enhanced nominal characteristics.

Within the area of modeling and simulation, an increasing amount of research is carried out with particular attention to electrical actuation systems used in offshore conditions. In [8], an induction machine for wind power plants is modeled which enables efficient simulation of multi machine installations like offshore wind parks. In [9]-[11] special attention is given to selected new developments within the field of electric motors. These involve control of an induction generator using direct vector control technique implemented in a virtual simulation environment, application of direct vector control in positioning of servomotors or verification of suitability of linear permanent magnet machine for drilling purposes. Finally, in [12], a set of main design criteria that need to be addressed when choosing components of an electrical drivetrain to operate in offshore environment is specified.

In addition, offshore field work and experimental data gathering is covered in the literature as well - see [13]-[15]. Development and testing of large-scale machinery is presented together with simultaneous faults analysis performed using modeling techniques. Finally, modeling strategies are used in the areas that do not have a direct link with engineering, e.g. optimization techniques applied to lower workload of personnel operating on a drilling rig - see [16]. This wide set of references gives a clear indication that modeling and simulation tools are popular in the offshore drilling industry and find a broad range of applications.

In the current work we use a third party virtual modeling software to create a multi-body model of a subsystem of an offshore pipe handling machine. A mathematical model of the considered mechanism is initially formulated that allows for verification of the simulation results accuracy. Additionally, assumptions and simplification techniques are introduced to facilitate modeling process. Finally, simulation results are benchmarked against full-scale data measured on a real world machine to validate the virtual models.

The present study contributes in three areas. Most importantly, it presents a full process of modeling and simulation of an existing offshore drilling machine, establishment of its multi-body model and formulating its simplified, analytical version. Secondly, it gives an overview on how a complex, rigid multi-body model that is computationally demanding could be optimized for real-time performance. Finally, the accuracy of the simulation results is demonstrated on field data from a full-scale offshore drilling machine by identifying friction phenomenon in the system and comparing simulation results with reference, real world measurements.

OFFSHORE PIPE HANDLING EQUIPMENT

Vertical Pipe Handling Machine

The vertical pipe handling machine shown in Figure 1 is a machine which assembles and delivers stands or pipes to the well center as the drilling process continues. According to [2], it is a column supported at the top and bottom by a track and a rack and pinion system. The lower track is mounted directly on the drill floor, whereas the upper track is connected to the derrick structure. At both ends of the column there are located trolleys which allow for horizontal movement of the machine along the tracks. In total the column includes three arms. Their role is to deliver, handle, and assemble pipes. The upper and lower ones are designed to guide a pipe, i.e. to guide the top and bottom of the pipe being in operation. The middle arm is the so-called gripper arm which is responsible for holding a pipe in a secure grip. It is possible to hoist or lower this arm by using a winch located on top of the column. All three arms are equipped with hydraulic cylinders which allow for their extension / retraction in order to position pipes in the finger boards or well center. The whole machine can rotate about its vertical axis thanks to slew motors mounted at the lower trolley. In the current work, the gripper arm operated by a winch is selected as a case study. Real world torque and velocity profiles are used to validate its simulation model.

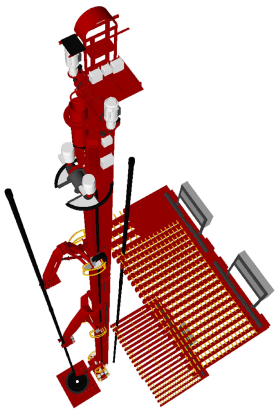

Figure 1: *Vertical pipe handling machine (MH VPR^{TM}) - courtesy of MHWirth.*

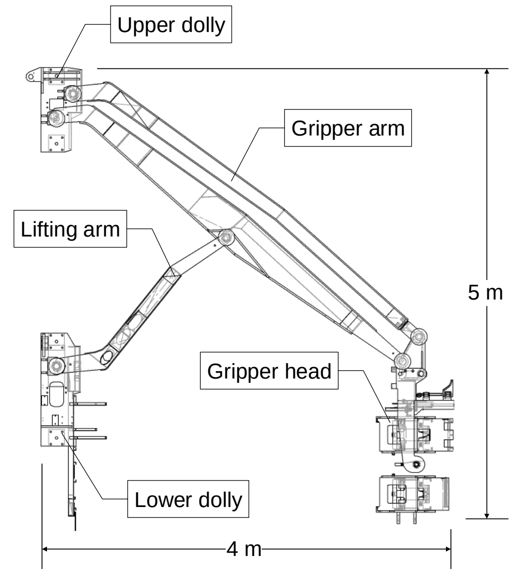

Figure 2: *The gripper arm structure.*

Gripper Arm - Functional Description

The gripper arm illustrated in Figure 2 is connected to the middle of the column through two dollies that are equipped with trolleys. Trolleys roll up and down on the column within the guide rails allowing, together with the winch mechanism, for vertical motion of the gripper arm. The geometry of the arms ensures that the head moves in a straight horizontal line from the column during extension / retraction cycles that are realized in a separate sequence. The winch drive consists of a motor, a hydraulic brake, a planetary gearbox and a winch drum directly connected to the hoisting wire. The hoisting wire is connected to the dead anchor at the top of the column via the sheave, located on the lower dolly. This arrangement provides for vertical motion of the gripper arm. In addition, there are two hydraulic cylinders installed between the lower and upper dollies that allow for horizontal extension / retraction of the arm by changing their stroke.

Simplified Gripper Arm

In order to create a computationally efficient virtual model of the analyzed gripper arm system, a simplified mechanical structure is considered as shown in Figure 3. Location of joints as well as overall dimensions and masses of particular elements of the mechanism are kept the same as in the original system. This arrangement is subsequently implemented in SimulationX software and its simulation results are verified with an analytical model introduced in the following Section. SimulationX is a commercially available multi-domain modeling software. It consists of a number of libraries from various physical domains (e.g. rigid body mechanics or hydraulic actuation systems). Applying such predefined components in modeling enhances design and analysis of complex mechatronic systems, as they are typically composed of many integrated sub-systems. In the current work, elements from a multi-body library are used to represent the gripper arm of MH VPR^{TM}.

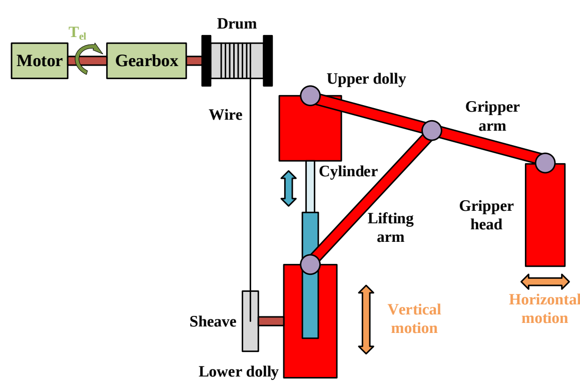

Figure 3: *A simplified gripper arm connected to the winch drivetrain.*

MATHEMATICAL MODELING

Modeling Assumptions

The scope of the current work is to analyze vertical motion of the gripper arm, without the effect of extension / retraction of the hydraulic cylinders. Therefore, it is assumed that the only degree of freedom in the machine is relative motion of the lower dolly (with all its attachments, e.g. upper dolly, gripper arm, etc.) with respect to the winch. In order to formulate a relatively computationally efficient model of such a system, the arrangement shown in Figure 3 is simplified and represented as a mass and drum / winch system, illustrated in Figure 4. Combined mass m_c includes masses of particular components of the gripper arm and mass of the payload (i.e. the pipe being handled). The following assumptions have been made to establish a mathematical model of the analyzed machinery:

1. Drum radius is constant. Normally, there are two layers of wire in use but in the current study we assume that there is no layer shift.
2. Mass of the wire is neglected since it constitutes only a small portion of the system total mass.
3. Inertia and mass of the drum are constant.
4. Wires are modeled as spring-damper systems: stiffness and damping are constant.
5. Gearbox inertia is neglected as it does not contribute significantly to the combined inertia of the modeled system.
6. Heave motion of the platform is neglected.

Wires

The wires are modeled as elastic spring-damper systems. Therefore, the elastic force in each wire is determined to be:

$$W_1 = k \cdot \delta_1 + c \cdot \dot{\delta}_1 \tag{1}$$

$$W_2 = k \cdot \delta_2 + c \cdot \dot{\delta}_2 \tag{2}$$

where $\delta_{1,2}$ - elongation of the wire [m], $\dot{\delta}_{1,2}$ - rate of elongation of the wire [m/s], k - stiffness of the wire [N/m], c - damping of the wire [Ns/m]. The following equations describe wire elongations ($x_c(t)$ - the combined payload displacement [m], θ_w - rotational displacement of a winch [rad], θ_{sh} - rotational displacement of a sheave [rad]):

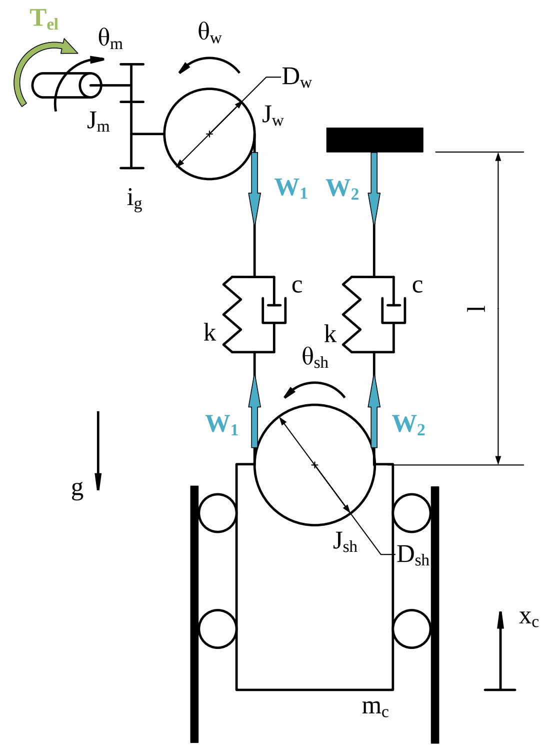

Figure 4: *Simplified representation of the gripper arm as the combined mass system.*

$$\delta_1 = \theta_w \cdot \frac{D_w}{2} + \theta_{sh} \cdot \frac{D_{sh}}{2} - x_c(t) \tag{3}$$

$$\delta_2 = -\theta_{sh} \cdot \frac{D_{sh}}{2} - x_c(t) \tag{4}$$

and wire elongation rates:

$$\dot{\delta}_1 = \dot{\theta}_w \cdot \frac{D_w}{2} + \dot{\theta}_{sh} \cdot \frac{D_{sh}}{2} - \dot{x}_c(t) \tag{5}$$

$$\dot{\delta}_2 = -\dot{\theta}_{sh} \cdot \frac{D_{sh}}{2} - \dot{x}_c(t). \tag{6}$$

What is more, since we do not model the wire compression, Equations 3 and 4 are valid only for $\delta_{1,2} \geq 0$. For $\delta_{1,2} < 0$ they become 0. Stiffness of the wire is determined by applying the following relationship (l - length of one wire segment [m], A_{eff} - effective cross-sectional area of the wire [m^2], E - Young's modulus [Pa]):

$$k = \frac{E \cdot A_{eff}}{l}. \tag{7}$$

According to [17], damping of the wire c is assumed to be equal to $c = 0.1 \cdot k$.

Winch and Sheave

Since the winch is fixed to the rig structure, it experiences no acceleration with respect to the overall system. Therefore, the equilibrium equation for the winch becomes:

$$N_w - G_w - W_1 = 0 \tag{8}$$

where N_w - reaction force of winch [N], $G_w = m_w \cdot g$ - gravity force of winch [N]. The sheave is connected to the combined mass, hence it moves together with it having its acceleration $\ddot{x}_c$. Therefore, the equilibrium equation for the sheave is given by:

$$-N_{sh} - G_{sh} + W_1 + W_2 = m_{sh}\ddot{x}_c \tag{9}$$

where: N_{sh} - reaction force of sheave $[N]$, $G_{sh} = m_{sh} \cdot g$ - gravity force of sheave [N]. To model the friction force between the bearing and pin of the sheave / winch, a simple Coulomb friction model is applied according to [17] (F_f - friction force [N], N - normal force [N]):

$$F_f = \mu \cdot N. \tag{10}$$

By substituting Equations 8 and 9 into Equation 10 we obtain the formula for the friction force on the sheave and winch, respectively (μ_{pin} - friction coefficient between bearing and pin):

$$F_{f,sh} = \mu_{pin}|(-G_{sh} + W_1 + W_2 - m_{sh}\ddot{x}_c)| \tag{11}$$

$$F_{f,w} = \mu_{pin}|(G_w + W_1)|. \tag{12}$$

It is assumed that the friction coefficient is the same for both sheave and winch. To mitigate the computational effort of the software during the detection of the change of direction of sheaves rotation, instead of using a typical *signum* function, the following piecewise $S(\dot{\theta})$ function was applied in the friction model:

$$S(\dot{\theta}_{w,sh}) = \begin{cases} -1 & \text{if } \dot{\theta}_{w,sh} < -0.01 \\ \dot{\theta}_{w,sh}/0.01 & \text{if } -0.01 \leq \dot{\theta}_{w,sh} \leq 0.01 \\ 1 & \text{if } \dot{\theta}_{w,sh} > 0.01. \end{cases} \tag{13}$$

Hence, the equation of motion (EOM) for the sheave becomes (D_{sh} - sheave diameter, $D_{pin,sh}$ - pin diameter of sheave):

$$J_{sh} \cdot \ddot{\theta}_{sh} = \frac{D_{sh}}{2} \cdot (W_2 - W_1) - F_{f,sh} \cdot \frac{D_{pin,sh}}{2} \cdot S(\dot{\theta}_{sh}). \tag{14}$$

In order to take into account the external torque (T_{el}) supplied from the electrical system, winch inertia J_w is combined with motor inertia J_m. There is a gearbox between motor and winch (i_g - gear ratio), therefore, the EOM for this system is given by (θ_m - angular position of rotor, D_w - winch diameter, $D_{pin,w}$ - pin diameter of winch):

$$(2J_m + J_w i_g^2)\ddot{\theta}_m = T_{el} + i_g \frac{D_w}{2} W_1 - i_g F_{f,w} \frac{D_{pin,w}}{2} S(\dot{\theta}_w). \tag{15}$$

There are two motors connected to the gearbox shaft in the real system, therefore the rotor inertia is doubled in the above equation and the electric torque is the sum of torques delivered by each motor $T_{el} = T_{m1} + T_{m2}$.

Combined Mass

Combined mass is composed of masses of particular components of the gripper arm structure (including sheave) and the payload being handled (i.e. mass of the pipe). Its EOM is expressed as:

$$W_1 + W_2 - G_c - F_{f,c} = m_c\ddot{x}_c \tag{16}$$

where G_c is the weight of the combined mass and $F_{f,c}$ is the friction force acting on the combined mass along its vertical motion. It is related to the fact that rollers on the upper and lower dollies experience friction while moving inside the guide rails. According to [17], a Coulomb friction model is selected to represent this phenomenon. It was identified that in some cases Equation 13 does not provide for smooth enough detection of combined mass velocity direction switching. Therefore, it was decided to use more sophisticated *atan* function for this purpose. Finally, the vertical friction force acting on the combined mass becomes:

$$F_{f,c} = \begin{cases} F_{C,neg} \cdot \operatorname{atan}(k \cdot \dot{x}_c)/(\pi/2) & \text{if } \dot{x}_c \leq 0 \\ F_{C,pos} \cdot \operatorname{atan}(k \cdot \dot{x}_c)/(\pi/2) & \text{if } \dot{x}_c > 0. \end{cases} \tag{17}$$

Factor k adjusts the slope of the $atan$ function in the vicinity of zero speed so, in other words, it decides about how fast the friction model responds to changes in velocity direction. A comparative overview of the switching detection functions described by Equations 13 and 17 is illustrated in Figure 5.

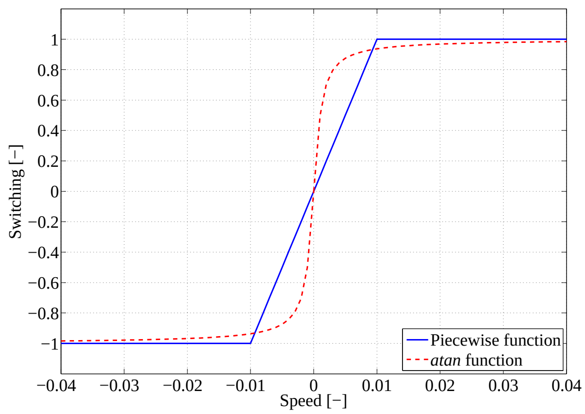

Figure 5: *Velocity switching detection functions.*

SIMULATION RESULTS
Reference Data Acquisition

In the full-scale Vertical Pipe Racker, there are two motors in the gripper arm coupled to one shaft that is actuating the winch through the gearbox (i.e. motors rotate with the same speed). A set of reference electric motors torques measured on a real world machine is illustrated in Figure 6. Both torque and speed profiles shown in this paper are normalized with respect to nominal torque T_n and nominal speed n_n of the motors used in the real machine. They have been recorded during regular machine operation when a pipe has been lowered and hoisted to certain positions. Hydraulic cylinders supporting the upper dolly have been immobilized during this sequence. The analyzed machine is controlled by a closed-loop control system: a reference speed profile is provided as a set point and the resulting torque profiles are recorded. These torque profiles are used in the current work to validate the simulation model of the gripper arm. The modeled system is excited with the real-world electromagnetic torque and the obtained velocity of motors is subsequently compared with the reference velocity recorded on a full-scale machine in order to assess model accuracy.

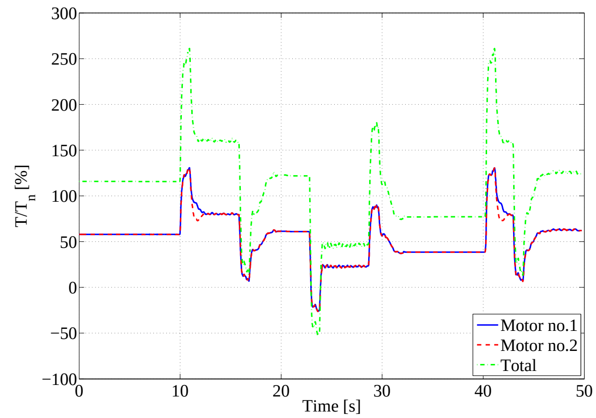

Figure 6: *Reference torque profiles measured on real world gripper arm winch motors.*

Validation of the Gripper Arm Models

Two friction parameters $F_{C,neg}$ and $F_{C,pos}$ are identified in an iterative process. Using Equation 16 it is possible to find the friction force profile $F_{f,c}$ that will fit to the reference electromagnetic torque T_{el} given the total system mass and wire properties. This operation is repeated a few times by adjusting values of $F_{C,neg}$ and $F_{C,pos}$ until the satisfactory system response is achieved. Since the obtained Coulomb friction model is not symmetric, it allows to capture the overall behavior of the gripper arm system depending if it is being hoisted or lowered by the winch. The overall shape of the friction model is depicted in Figure 7. The friction force is normalized with respect to the gravity force acting on the combined mass system.

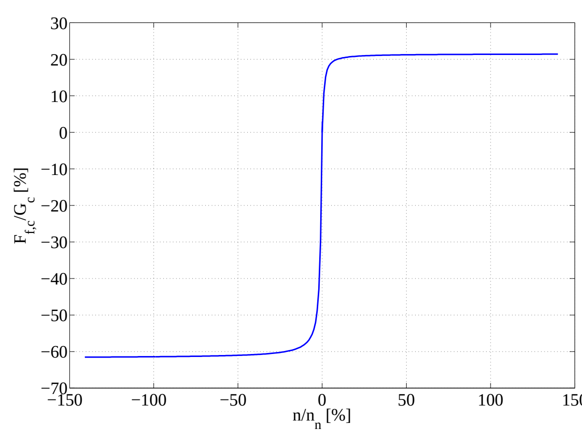

Figure 7: *Modified Coulomb friction model.*

Performed simulations involve validation of both the multi-body model of the gripper arm composed of a set of rigid bodies and joints as well as the analytical model represented by the combined mass and winch system. The total reference torque acquired from the real world machine is provided to both models as the input electromagnetic torque T_{el}. It is the sum of the torque signals measured on both drives of the drivetrain. The resulting velocity of the motors common shaft is then recorded in both simulations. The simulated velocity is compared with the reference velocity giving the basis to assess models accuracy. Results of this operation are shown in Figure 8.

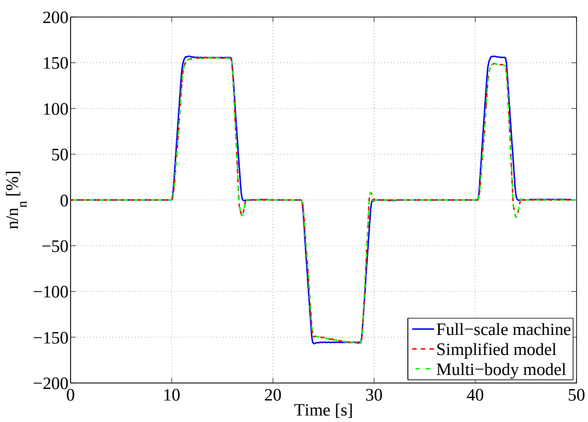

Figure 8: *Simulation results - velocity profiles.*

Figure 8 shows that both simulated models yield results that closely correspond to the reference velocity profile. Root-mean-squared error (RMSE) for reference and simulated velocities is equal to an acceptable low value of $RMSE = 7.49$ %. The analytical model is also implemented in a virtual simulation software making it possible to compare computational effort needed by a solver to run the simulation. Both multi-body and analytical models are simulated on a commercial PC: Intel® Core^{TM} $i7-3770$ CPU @ $3.4GHz$ and $8GB$ RAM. The solver used is BDF (Backward Differentiation Formula) with minimum calculation step size being equal to $dtMin = 1e-10$ [s]. Time intervals needed by the computer to finish both simulations (each lasting 49.76 [s]) are shown in Table 1.

Multi-body model	**Analytical model**
91.53 [s]	3.46 [s]

Table 1: Simulation time required by the solver to finalize computations

From Table 1 it follows that the analytical model could be run approximately 26 times faster than the multi-body model. Simultaneously, simulation accuracy remains at the same level as for the multi-body (i.e. more detailed) model - see Figure 8.

CONCLUSION

The current paper presents a case study of modeling and simulation of an offshore machine. Contrary to prior works, this study is specifically devoted to vertical pipe handling machine and identification of friction phenomenon appearing in this system. Comparative analysis clearly shows that both analytical and multi-body models of the considered equipment yield simulation results that stay in close accordance with the reference, real world measurements. In addition, it is verified that the simplified, analytical model runs significantly faster on a commercial PC than its more detailed, multi-body equivalent. The difference is so remarkable that the analytical model could easily be used in real-time testing of this equipment allowing to evaluate its control system before commissioning. This has a huge impact on design efficiency and project management, facilitating communication of different teams of engineers at early stages of product development.

It is expected that in the future these types of models could also find different applications. One example is fault detection and diagnostics. Having a validated model of a machine that operates offshore on a full scale drilling rig will allow to detect unexpected deviations in its performance. This will help to identify sources of measurements discrepancies between a reliable model and a real machine and consequently improve maintenance and service tasks. It will be beneficial to extend the current study into additional areas as well. Including a control system in the virtual model will help to improve its performance and robustness by allowing to spend more time on testing different control algorithms. Additionally, it is advisable to use smooth velocity profiles (instead of trapezoidal ones) as reference signals to a controller. It will decrease peaks in torque profiles resulting in a more stable operation of the machine. Finally, testing the established simulation models against different sets or real world measurements (e.g. various payload masses) will result in more robust models being able to yield accurate results under different load conditions.

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