Comparative Analysis of Numerical Models of Pipe Handling Equipment Used in Offshore Drilling Applications

Witold Pawlus*, Morten K. Ebbesen, Michael R. Hansen, Martin Choux, and Geir Hovland

Department of Engineering Sciences, University of Agder, PO Box 509, N-4898 Grimstad, Norway, e-mail: *witold.p.pawlus@ieee.org

Abstract. Design of offshore drilling equipment is a task that involves not only analysis of strict machine specifications and safety requirements but also consideration of changeable weather conditions and harsh environment. These challenges call for a multidisciplinary approach and make the design process complex. Various modeling software products are currently available to aid design engineers in their effort to test and redesign equipment before it is manufactured. However, given the number of available modeling tools and methods, the choice of the proper modeling methodology becomes not obvious and – in some cases – troublesome. Therefore, we present a comparative analysis of two popular approaches used in modeling and simulation of mechanical systems: multibody and analytical modeling. A gripper arm of the offshore vertical pipe handling machine is selected as a case study for which both models are created. In contrast to some other works, the current paper shows verification of both systems by benchmarking their simulation results against each other. Such criteria as modeling effort and results accuracy are evaluated to assess which modeling strategy is the most suitable given its eventual application.

Keywords: Numerical analysis, mechanical system design, modeling and simulation, offshore drilling applications.
PACS: 45.80.+r, 45.40.-f, 07.05.Tp.

INTRODUCTION

The modeling and simulation tools are indispensable for the improvement of mechanical system design processes and reducing the effort of design engineers [1], [2], [3], especially in the offshore drilling industry, where such a high focus is on safety, efficiency, and performance of the produced equipment [4], [5], [6]. With the development of the commercial multibody modeling software, it has become more convenient to test and redesign various solutions in a virtual simulation environment before applying them in real world. However, the accessibility of such products and relatively low level of expertise required to create moderately complex physical models might be a potential source of errors and misinterpretations of simulation results. Especially, complex closed-loop mechanisms, such as parallel mechanisms or manipulators require special attention during modeling process to avoid achieving only seemingly correct outcomes [7] – one example is finding the reaction forces in joints for such systems [8]. Therefore, this paper presents a comparative analysis of modeling and simulation of an offshore vertical pipe handling machine using a commercial multibody software and analytical methods. Two scenarios are investigates: inverse dynamics – to verify that the geometry of both models is matching, and forward dynamics – to verify if both models handle the external loads in the same way. The contribution of this work lies in comparison of the numerical model of the offshore machine with a model created in an existing commercial multibody software. In addition, we identify possible bottlenecks in results interpretation when using a product available in the market and highlight pros and cons of each approach.

EQUATIONS OF MOTION FOR SYSTEM OF BODIES

A set of Cartesian coordinates defines the global position of a rigid body's i reference frame according to [9]:

$$\mathbf{q}_i = \begin{bmatrix}\mathbf{r}_i^T & \phi_i\end{bmatrix}^T \tag{1}$$

where $\mathbf{r}_i = [x_i \;\; y_i]^T$ are the translation coordinates and ϕ_i is the rotational coordinate of the body in planar motion. The velocities and accelerations of body i are the corresponding time derivatives given by the vectors:

$$\dot{\mathbf{q}}_i = \begin{bmatrix}\dot{\mathbf{r}}_i^T & \dot{\phi}_i\end{bmatrix}^T \tag{2}$$

International Conference of Numerical Analysis and Applied Mathematics 2015 (ICNAAM 2015)
AIP Conf. Proc. 1738, 370013-1–370013-4; doi: 10.1063/1.4952158
Published by AIP Publishing. 978-0-7354-1392-4/$30.00

$$\ddot{\mathbf{q}}_i = \begin{bmatrix}\ddot{\mathbf{r}}_i^T & \ddot{\phi}_i\end{bmatrix}^T. \tag{3}$$

The equations of motion of an unconstrained system of bodies are written as (vector $\ddot{\mathbf{q}}$ contains accelerations of all bodies present in a system):

$$\mathbf{M}\ddot{\mathbf{q}} = \mathbf{g} \tag{4}$$

where $\mathbf{M}$ is a mass matrix and $\mathbf{g}$ is a force vector that is composed of external and reaction forces, respectively: $\mathbf{g} = \mathbf{g}^{ext} + \mathbf{g}^{c}$. According to [10], the reaction forces are expressed by the product of the Jacobian matrix $\mathbf{\Phi}_q$ and the Lagrange multipliers vector $\boldsymbol{\lambda}$:

$$\mathbf{g}^{c} = \mathbf{\Phi}_q^T \boldsymbol{\lambda}. \tag{5}$$

In a constrained multibody system the kinematic joints are described by a set of algebraic constraints which depend on time:

$$\mathbf{\Phi}(\mathbf{q}, t) = \mathbf{0}. \tag{6}$$

Consequently, the velocity and acceleration constraints need to be satisfied as well:

$$\dot{\mathbf{\Phi}}(\mathbf{q}, t) = \mathbf{\Phi}_q \dot{\mathbf{q}} = \mathbf{0} \tag{7}$$

$$\ddot{\mathbf{\Phi}}(\mathbf{q}, t) = \mathbf{\Phi}_q \ddot{\mathbf{q}} = \boldsymbol{\gamma}. \tag{8}$$

The constraints (algebraic equations - AEs) should also be considered when solving equations of motion for the system (ordinary differential equations - ODEs). This is achieved by using the Lagrange multipliers technique. This way the ODEs are written together with the second time derivative (8) of the constraint equation (6). The motion of the multibody system is described as:

$$\begin{bmatrix}\mathbf{M} & -\mathbf{\Phi}_q^T \\ \mathbf{\Phi}_q & \mathbf{0}\end{bmatrix}\begin{bmatrix}\ddot{\mathbf{q}} \\ \boldsymbol{\lambda}\end{bmatrix} = \begin{bmatrix}\mathbf{g}^{ext} \\ \boldsymbol{\gamma}\end{bmatrix} \tag{9}$$

where $\boldsymbol{\gamma}$ is the vector consisting of all the terms of the acceleration constraint equations that depend on the velocities only. Solving (9) for accelerations and Lagrange multipliers yields:

$$\begin{bmatrix}\ddot{\mathbf{q}} \\ \boldsymbol{\lambda}\end{bmatrix} = \begin{bmatrix}\mathbf{M} & -\mathbf{\Phi}_q^T \\ \mathbf{\Phi}_q & \mathbf{0}\end{bmatrix}^{-1}\begin{bmatrix}\mathbf{g}^{ext} \\ \boldsymbol{\gamma}\end{bmatrix}. \tag{10}$$

In the inverse dynamics scenario, on the other hand, to describe motion of the mechanism it is sufficient to determine its positions, velocities, and accelerations by solving (6), (7), and (8) for a given kinematic preset (e.g. velocity).

OFFSHORE VERTICAL PIPE RACKER MODELING

The vertical pipe racker (VPR) of MHWirth shown in Fig. 1 is a full-scale offshore drilling machine that handles stands between finger boards and well center during the drilling process [4]. The 45 m tall column is connected to the upper and lower carriages that translate horizontally. In addition, the column features hydraulic arms which are used to guide and grip a tubular. Finally, the machine can rotate about its vertical axis thanks to slew motors installed on the lower carriage. In the current work, the gripper arm actuated by the winch hoisting system is investigated in detail to examine the effectiveness of two modeling approaches. The modeling workflow for the gripper arm of the VPR is illustrated in Fig. 1. The system is simplified by considering only the most important overall dimensions and masses, which, together with locations and types of joints, are kept the same as in the original equipment. In addition, only the planar motion is analyzed and friction is neglected in the system. The current work considers the vertical motion of the gripper arm with the effect of extension / retraction of the hydraulic cylinder which is installed between the lower and upper dollies as depicted in Fig. 1. Two input velocities used in the inverse dynamics scenario to excite the system are:

$$v_{cyl} = 0.2 \ \ [m/s], \qquad v_{winch}(t) = \frac{2\pi}{3} \cdot 0.7 \cdot \cos\left(\frac{2\pi}{3}t\right) \ \ [m/s]. \tag{11}$$

In addition, the revolute joint between bodies $3-5$ is assumed to have a zero velocity preset. In the forward dynamics scenario, the input forces are:

$$F_{cyl}(t) = 22000 + 15000 \cdot \sin(\pi t) \ \ [N], \qquad F_{winch}(t) = m_{tot}g + 10000 \cdot \cos\left(\frac{2\pi}{3}t\right) \ \ [N] \tag{12}$$

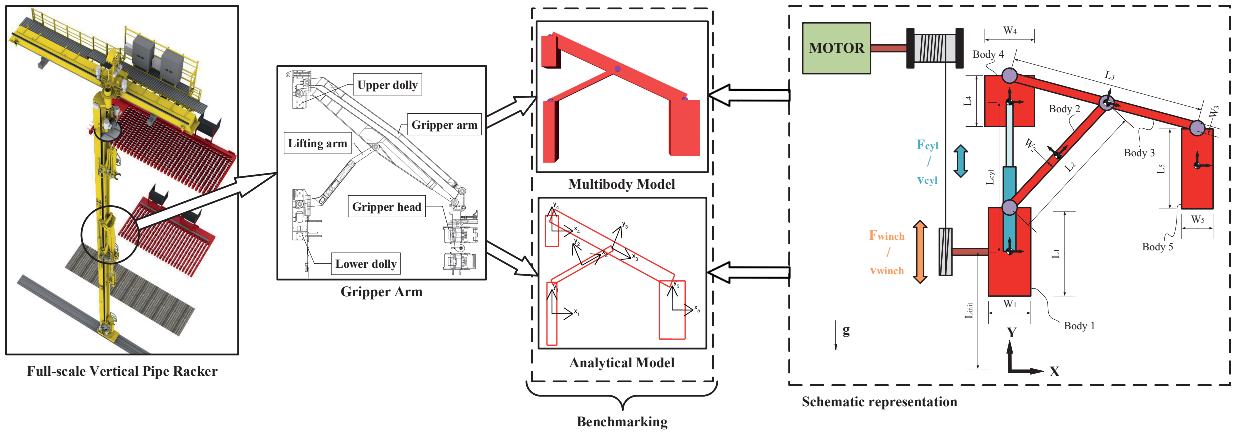

FIGURE 1. Workflow for VPR gripper arm modeling.

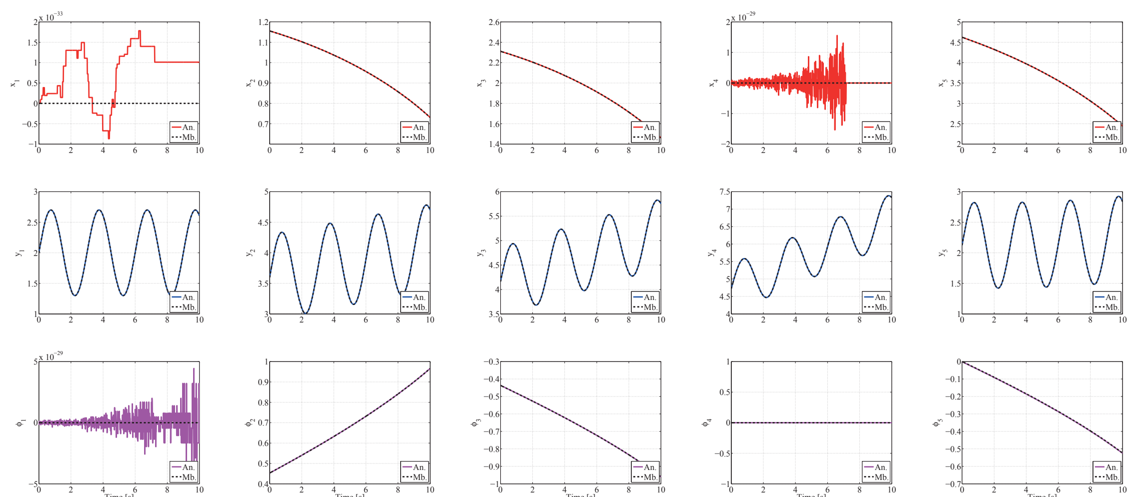

FIGURE 2. Comparison of simulated positions in the inverse dynamics scenario for analytical (An.) and multibody (Mb.) models (units are $[m]$ for x_i, y_i and $[rad]$ for ϕ_i coordinates).

where m_{tot} is the total mass of the mechanism and g is the gravity constant. Also, the initial cylinder extension L_{cyl} is increased as compared to the inverse dynamics case and a rotational spring and damper is applied between bodies $3-5$. Numerical data (dimensions and masses) of the system is selected to correspond to the realistic parameters of the reference machine.

SIMULATION RESULTS

The Cartesian coordinates of each body in planar motion under conditions specified in (11) and (12) are illustrated in Fig. 2 and Fig. 3. For the inverse dynamics case only the positions are shown to verify that the models geometry, constraint vector $\mathbf{\Phi}(\mathbf{q},t)$, and the Jacobian matrix $\mathbf{\Phi}_q$ are correct. For the forward dynamics, we plot only the resulting accelerations, since the velocities and positions could easily be obtained by performing the time integration of the acceleration signals. The results of the analytical modeling closely correspond to the simulation data generated in a commercial modeling software. Oscillations of analytical positions and accelerations close to zero value are related to the type of the solver used and its step size (Runge-Kutta solver, $\Delta t = 1 \cdot 10^{-3}$ s). It is expected that this discrepancy would disappear if considerably smaller step size were applied in the analytical method.

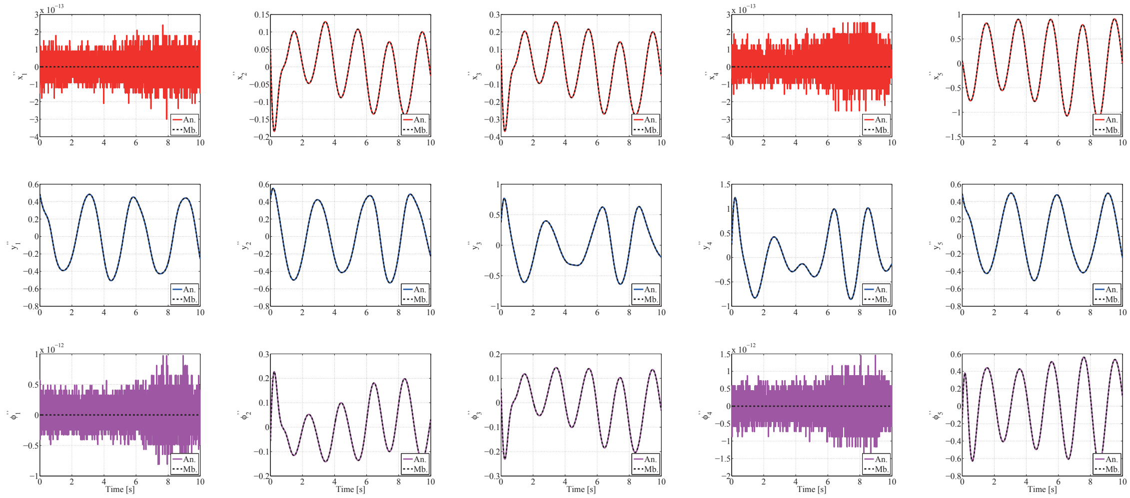

FIGURE 3. Comparison of simulated accelerations in the forward dynamics scenario for analytical (An.) and multibody (Mb.) models (units are $[m/s^2]$ for $\ddot{x}_i$, $\ddot{y}_i$ and $[rad/s^2]$ for $\ddot{\phi}_i$ coordinates).

CONCLUSION

In this paper we present modeling and simulation of an offshore pipe handling machine using both the commercial multibody software and analytical methods. Both inverse and forward dynamics scenarios are examined and simulation results produced by these two different approaches are confirmed to be identical. The modeling approach aided by the existing software seems to be more practical from the industrial perspective, since it allows rapid prototyping of new designs, saves both time and engineering effort, and does not require expert knowledge to use it successfully. However, more thorough analysis makes it possible to check if the modeling process based on commercial products is executed correctly and to avoid (or at least identify) mistakes by benchmarking the simulation outcomes of both modeling methods at the cost of rather complex and time-consuming analytical model derivation. Such comparative analysis should be performed not only if simulation results do not appear to be realistic but also to double-check if the design choices made based on the virtual model would be the same if another modeling strategy is employed, as for complex systems an assessment of results feasibility is not always straightforward. Hence, it is recommended to use an analytical modeling approach in these cases to be sufficiently conservative. Interestingly, the discrepancies between the simulated and real results might not only be caused by human errors of design engineers. There are indications that some commercially available multibody modeling software products do not handle sufficiently well determination of reaction forces in joints of closed-loop mechanisms. Therefore, in the future it is advisable to investigate this and compare reaction forces in joints for analytical and software-based models.

REFERENCES

1. T. O. Andersen, and M. R. Hansen, "Automated sizing procedure of servo-driven robot for pallettes handling," in *Mechanical Engineering Congress and R & D Expo, 2004 ASME International*, 2004, pp. 1–6.
2. T. H. Langer, T. K. Iversen, O. Ø. Mouritsen, M. K. Ebbesen, and M. R. Hansen, *Structural and Multidisciplinary Optimization* **47**, 621–630 (2013).
3. Y. Tazaki, and T. Suzuki, *Robotics, IEEE Transactions on* **30**, 1227–1234 (2014).
4. W. Pawlus, M. Choux, G. Hovland, S. Øydna, and M. R. Hansen, "Modeling and simulation of an offshore pipe handling machine," in *Simulation and Modeling, 2014 SIMS 55th Conference on*, 2014, pp. 277–284.
5. J. Hao, L. Yancong, Z. Yanting, L. Zhendong, and W. Guangbin, "Research on new drill string heave compensation system for floating drilling platform," in *Fluid Power and Mechatronics (FPM), 2011 International Conference on*, 2011, pp. 791–795.
6. A. Albert, O. Aamo, J.-M. Godhavn, and A. Pavlov, *Control Systems Technology, IEEE Transactions on* **23**, 813–819 (2015).
7. W.-H. Ding, H. Deng, Q.-M. Li, and Y.-M. Xia, *Robotics and Computer-Integrated Manufacturing* **30**, 421–431 (2014).
8. J. G. de Jalón, and E. Bayo, *Kinematic and Dynamic Simulation of Multibody Systems*, Springer Verlag, New York, 1994.
9. P. Flores, and J. Ambrósio, *Computers & Structures* **82**, 1359–1369 (2004).
10. P. E. Nikravesh, *Computer-Aided Analysis of Mechanical Systems*, Prentice Hall, Eaglewood Cliffs, New Jersey, 1988.