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Multibody Models and Simulations to Assess the Stability of Counterbalance Forklift Trucks

Michele Gardella and Alberto Martini

Abstract Assessing the stability of counterbalance forklift trucks is a problem of primary interest, since critical safety issues related to their operation are involved. Indeed, forklifts are the most common vehicles for material handling in industry. The stability limit is typically verified through experimental tests, which are costly and time consuming. This study aims at developing numerical tools to reliably predict the stability limit through simulations. Such tools should permit, on one hand, to partially reduce the amount of experimental tests necessarily required and, on the other hand, to support the design phase of new products since the early stages. A multibody model taking into account the compliance of the tires and of the mast assembly is developed. An experimental campaign to validate the model is ongoing. The preliminary numerical results confirm the model as a promising tool to estimate the stability limit of the forklift with satisfactory accuracy.

1 Introduction

Counterbalance forklift trucks constitute worldwide an essential equipment for material handling in manufacturing and warehousing. One of the most critical tasks to ensure the safety of forklift operation is assessing properly the stability conditions of the vehicle. Indeed, forklift tip over and roll over caused by, respectively, braking and cornering maneuvers, represent the most common accidents for this kind of vehicles [3].

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The stability analysis of forklifts is typically performed in accordance with the ISO 22915-2:2018 Standard, which defines four different test conditions, referred to as *Test 1-4* [1]: the vehicle is placed on a tilt-table that is driven in quasi-static conditions to a prescribed slope. The testing conditions of major interest for this study are the ones referred to as *Test 1* and *Test 3* [2]: the former simulates a breaking maneuver with the mast at maximum height carrying the rated load (Fig. 1), for evaluating the longitudinal stability; the latter simulates a turning maneuver at constant speed with the rated load lifted at maximum height (Fig. 2), hence assessing the lateral stability.

Fig 1. *Test 1*, (a) schematics, (b) view of the forklift during test.

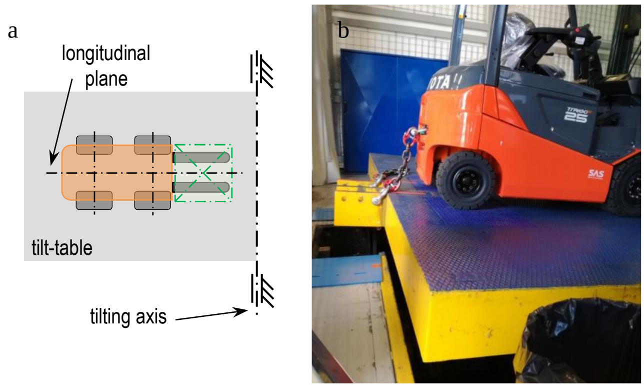
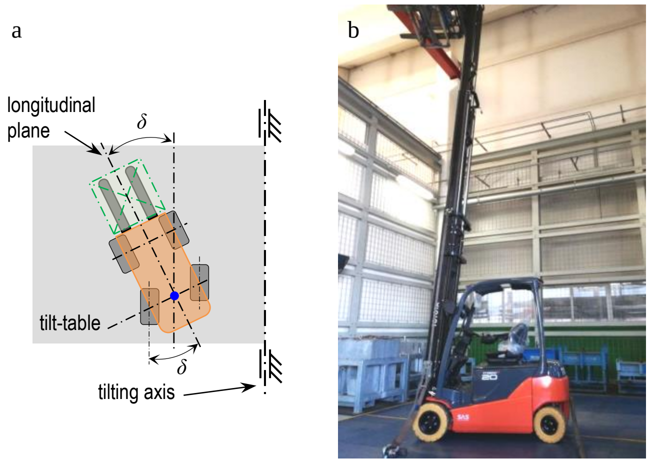


Fig 2. *Test 3*, (a) schematics, (b) view of the forklift during test.



The company promoting this research activity, namely Toyota Material Handling Manufacturing Italy SpA (Bologna, Italy), conducts experimental tests to assess the stability of each new model of forklift. Moreover, many vehicles customized with masts bigger than the standard model and/or special purpose forks (e.g.

featuring additional actuators for handling the payload) are also manufactured, each one requiring additional tests for the stability analysis. This results in a large amount of experimental tests to be performed.

Moreover, the test are used by the company to gain further insights into the stability limits, to be exploited for guiding the design of new forklift models. Indeed, the analytical models adopted in the early design phases do not provide accurate enough results, due to the complex deformations affecting the forklift loaded with the nominal load. However, due to safety reasons, the actual tip over/roll over conditions are not generally reached in tests, particularly in *Test 3* (although the vehicle is secured to the tilt-table and/or to a fixed frame by means of chains and/or belts); hence, the stability limit must be extrapolated from the experimental results. This causes higher safety factors to be required in the design process.

The main objective of the research is developing numerical simulation tools based on multibody models to carry out the stability analysis of counterbalance forklift trucks. Firstly, they should be able to predict with high accuracy and reliability the forklift behavior in terms of stability limit, thus allowing to gradually reduce the need for experimental tests as well as to help the design of new products since the early stages. Secondly, such tools should be easy to implement by starting from the other numerical models (e.g. CAD and FE models) already implemented by the manufacturer, hence being well integrated in the design procedures of the company. To the authors' best knowledge, there is no established numerical approach already available in the literature to reliably assess forklift stability.

As a first stage of the research, this work aims at replicating the stability tests, in particular *Test 1* and *Test 3*, for achieving two main goals: the first one is to carry out a deeper investigation of the discrepancies between the experimental results and the analytical models; the second goal is to define the most convenient modelling strategy (i.e. satisfactorily accurate and cost effective) based on a multibody approach, which is reckoned as the main novel outcome of the presented activity. Indeed, although such tests are performed in quasi-static conditions, multibody modelling and simulations are deemed more suitable than FE analyses due to both a higher flexibility for rapidly switching between different configurations of the forklift (e.g. modifying the steering angle) and/or different test conditions, and potentially lower computational time.

2 Materials and methods

The study focuses on a four-wheeled forklift of the *Traigo 48* family, with electric propulsion (Fig. 3). In particular, the studied vehicle is characterized by a rated load of 1800 kg and a lift height of 5.5 m provided by a 3-stage mast. It is equipped with super-elastic solid tires. The mast tilt is controlled by two hydraulic cylinders; its lift is driven by a system of chains, pulleys and hydraulic actuators. The mast also includes an actuated attachment for fork positioning.

On the basis of the company know-how, the stability of the loaded forklift with fully extended mast is expected to be primarily affected by two contributions of deflection: (i) nonlinear deformations of the front tires; (ii) flexural deformations of the forks and the mast, with nonlinearities introduced by the translational guides (and additional torsional components in *Test 3*). The former effect should be preponderant in *Test 1*, where the mast deflection is compensated by imposing a proper backward tilt angle on the mast, in order to keep unaltered the longitudinal distance between the front axle and the Center of Mass (CoM) of the load at the beginning of the test. The latter is expected to be predominant in *Test 3*, where the vertical longitudinal plane of the forklift is inclined with respect to the rotational axis of the tilt-table, and maximum backward tilt angle is imposed on the mast (the corresponding angles being 16° and 6° for the studied forklift).

Fig 3. Side view of the studied forklift.

Approximate analytical models have been developed to reckon the stability limit. The model for *Test 1* takes into account the forklift pitch angle associated with the tire compliance (estimated on the basis of the datasheet provided by the tire manufacture) and the load CoM displacement caused by the deflection of the mast (approximated as a cantilever beam). The error on the stability limit predicted by such model is not deemed satisfactory, although not exceeding 10%. As for *Test 3*, only the tire compliance is considered, since reliable flexural-torsional deflections cannot be analytically estimated, due to the complexity of the mast assembly. With this rough approximation, the stability limit is overestimated by about 50%.

2.1 Numerical multibody model

A numerical model of the complete forklift is implemented by using the FunctionBay RecurDyn multibody software package. All the components are modelled as rigid bodies derived from the CAD model, with the exception of the mast assembly. The steering system is also included, to set the longitudinal plane of the rear axle wheels parallel to the tilting axis in *Test 3* (Fig. 2). The mass distribution is adjusted to match the actual forklift global CoM, determined by measuring the tire normal forces through load cells.

The tire-ground interactions are modelled by using *solid-to-solid* contacts. In particular, the *Extended Surface* feature is adopted. Indeed, its formulation (namely a function of the maximum penetration depth) allows to define the contact stiffness straightforwardly by setting a spline that fits the static force-displacement curves provided by the tire manufacturer. Hence, it appears convenient in terms of efforts required to implement the model. By contrast, longer computational time may be experienced with respect to visco-elastic forces defined through analytical functions. Moreover, such contacts do not replicate stiction effects, which are actually experienced in tests, since the wheels are braked). Nonetheless, both drawbacks are deemed acceptable. In particular, relative tangential motion between the tire and the ground is expected to be negligible, the tests to be simulated generally not exceeding 30 s.

The mast assembly is modelled by means of a Finite Element (FE) mesh and a *full-flex* formulation (i.e. not reduced). In particular, the three mast sections (i.e. inner, central and outer mast), the attachment (i.e. the fork carrier) and the forks are meshed with shell elements (4-node structural shells, size of about 15 mm). The rollers, composed of a steel outer ring and a PTFE thrust bearing, are modelled as rigid parts. Each roller is connected to its (flexible) shaft through a rigid spider allowing the rotational DOF. The interactions between each roller and the corresponding mast guide are modelled by using two distinct *solid-to-flex* contacts: a *cylinder-to-surface* is implemented to take into account the radial loads; an additional contact models the axial loads. Contact patches with a finer mesh size (about 2 mm) are added to make the contacts work properly. The connections between the mast and the front axle assembly are modelled by means of ideal joints. At this stage, the relative motion between the mast sections and the forks is prescribed by using velocity functions that neglect the compliance of the hydraulic circuit and the transmission chains. The *full-flex* formulation permits to exploit the FE models already available to the company (developed in Ansys Workbench). The increment in the computational cost with respect to a Craig-Bampton reduced model is regarded as acceptable.

The load is a rigid body supported by contact forces exerted by the forks.

2.2 Experimental tests

Experimental measurements are performed for model updating and validation. In particular, *Test 1* and *Test 3* are carried out with a special setup featuring four load cells installed between the forklift tires and the tilt-table, in order to monitor the normal force acting on each wheel (Fig. 4). It is worth noting that the normal force is expected to become null, for the wheel(s) at the highest level on the tilt-table, when reaching the stability limit.

A further ad hoc experiment is arranged and conducted on the mast assembly to better characterize its flexural-torsional compliance. The test aims at measuring the actual displacements of the application point of the load (a dummy rigid pallet) and of other reference points belonging to the three mast sections, the fork carrier and the forks, with respect to the connections between the mast and the front axle assembly, for different lift heights. The mast assembly is separated from the vehicle and installed on a rigid support connected to the tilt-table, which replicates the orientation of the mast in *Test 3*. The tilt-table is rotated up to a slope of 7% (about 4°), in two loading conditions, namely loaded and unloaded mast. The 3D positions of the points of interest are detected through stereophotogrammetric measurements, performed by using a 8-camera VICON optoelectronic system. To this purpose, proper reflective markers are attached to the locations to be monitored (Fig. 5a).

Fig 4. Close up of the load cells in *Test 3*.

The comparison between the displacements of several reference points measured in the tested loading cases, for a lift height of 5.5 m, is reported as example in Fig. 5b (the actual values can not be shown, due to NDA). Although the experimental campaign is still ongoing, the first results confirms that remarkable combined flexural-torsional deformations can be experienced on the top of the extended mast, hence significantly affecting the stability limit.

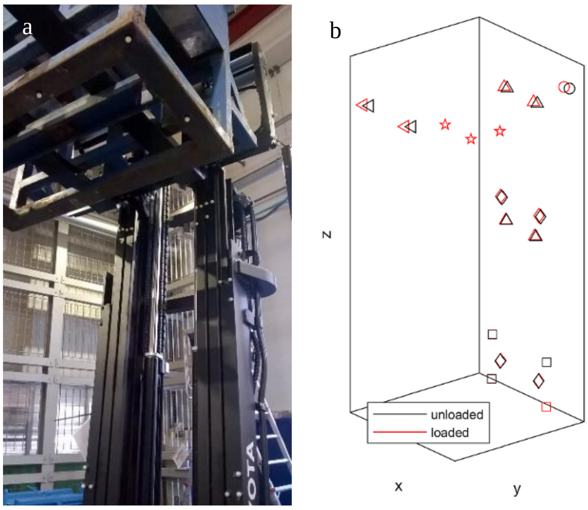


Fig 5. (a) Close up of the markers in the test setup; (b) comparison between unloaded and loaded mast, for test at 5.5 m lift height, of points of the outer (square), central (diamond), inner (upward triangle), carrier (circle), forks (leftward triangle) and load (star).

3 Preliminary results and model verification

The reliability of the implemented multibody model is assessed by evaluating the percentage error (*ε%*) affecting the stability limit predicted by the model with respect to the measured data, computed as:

$$\varepsilon\% = \frac{stability_{sim} - stability_{exp}}{stability_{exp}} \cdot 100 \tag{1}$$

where the stability limit is defined as the tilt-table slope corresponding to a condition of null normal force acting on the highest wheel of the forklift, and the subscripts *sim* and *exp* indicate numerical and measured results, respectively. The models are expected to overestimate the stability limit, hence *ε%* being typically positive.

The model has been verified in two steps, on the basis of the preliminary results currently available.

3.1 Test 1

As mentioned in Section 2, a minor influence of the mast deflection on the results of *Test 1* is expected, since the initial static deflection is compensated before the test starts by tilting the mast itself. Hence, the experimental results obtained in *Test 1* are exploited to partially validate the contact model describing the tire-ground interaction, whereas, at this stage, the mast assembly is assumed to behave like a rigid body (thus temporarily suppressing the features describing the compliance of the mast).

The spline curve describing the contact stiffness for a normal load between 0 and 5 kN is adjusted to minimize the parameter *ε%*, through numerical optimization. Indeed, the tire manufacturer does not provide any data for this load range.

After model updating, the stability limit error provided by the simulations is reduced below 5%. A comparison between the results of the approximated model (§2) and the updated multibody model for *Test 1* is reported in Table 1.

3.2 Test 3

Since the experimental campaign on the mast assembly is still ongoing, only a partial verification of the model for *Test 3* is performed. In particular, the position of the load CoM is updated on the basis of the first experimental results, whereas modifications in the mass distribution of the mast assembly caused by its deflection are neglected (i.e., the mast assembly is again considered as rigid).

The simulation results obtained from this model exhibit a significant improvement in the stability limit prediction, *ε%* being lower than 20%. A comparison with the results of the approximated model (§2) is shown in Table 1.

Table 1 Results provided by the numerical models.

Test #	Stability limit error (ε%)	
	Approximate model	**Multibody model**
Test 1	8.3 %	3.5 %
Test 3	47.0 %	18.7 %

4 Conclusion

A numerical multibody model for the stability analysis of a forklift truck has been developed and its reliability has been partially verified by using the preliminary results of an experimental campaign.

The results provided by the current model are promising and reasonably, after completing its validation, it will permit to predict the stability limit of the studied forklift with high accuracy.

On the basis of the results of this study, a standardized modelling procedure will be defined. Then, a larger set of numerical models will be generated for comparing the results with the company database of the stability test measurements, in order to confirm the reliability of the developed approach.

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