Development and experimental validation of a numerical multibody model for the dynamic analysis of a counterbalance forklift truck

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Counterbalance forklift trucks are probably the most common type of material handling equipment in industrial applications. Typically, they do not have suspension systems with spring elements and/or shock absorbers; the vehicle is supported at three points (namely the two front wheels and the pivot point of the rear axle, which can freely swing) and tires constitute the most deformable components. Moreover, solid rubber tires or cushion tires are adopted for most applications, instead of pneumatic ones. Due to these specifications, the dynamic response of the vehicle can be significantly affected by small irregularities of the ground, like speed bumps and potholes. Indeed, impacts of the wheels on such obstacles during motion can cause instability and make the forklift tip over, thus representing a risk for safety [1]. In addition, the undamped energy of the impacts can generate transient overloads on the vehicle chassis and other components, hence inducing high vibration levels and durability issues [2].

The objective of this study is to investigate the dynamic behavior of a prototypal heavy-duty forklift. The final goal is implementing virtual testing tools to reliably assess the dynamic stresses acting on the specific family of counterbalance forklift trucks of interest during a standard working cycle defined by the manufacturer's testing protocols. As a first stage of the research, the focus is on developing and validating a numerical model to simulate the rigid body dynamics of the prototype for the case of passage over a speed bump, which is one of the most critical testing conditions.

The studied vehicle is an electric counterbalance forklift of about 16 tons of weight, with a load capacity of about 8 tons (Fig. 1a). Two motors with planetary gearboxes (one for each side) independently drive the front wheels (namely, twin wheels with solid tires). Two wheels with identical tires are mounted at the rear (steering) axle. The cabin is mounted on conical elastomeric bearings. The forklift features a highly asymmetric chassis, which is specifically conceived to permit automated fast replacement of the large battery (about 15% of the forklift mass). Due to this design (and the relevant dynamic loads), a careful evaluation of the actual strain/stresses characterizing the chassis in working conditions is required.

Experimental tests are performed by driving the unloaded forklift over a speed bump (namely, a 33 mm high steel obstacle with trapezoidal cross-section) at a constant velocity of 11 km/h. Twenty runs are carried out in order to ensure repeatability. About twenty strain gauges are attached to the chassis for monitoring its deformations (to validate flexible multibody models in the future steps of the research). Vibrations in a frequency range up to 400 Hz are detected by means of six triaxial piezoelectric accelerometers: one transducer is installed next to each wheel hub; one sensor is mounted on the chassis, at about half the wheelbase; one accelerometer is placed in the cabin, under the operator's seat.

The acquired acceleration signals exhibit two major peaks occurring, respectively, when the front and the rear wheels impact on the obstacle, and significant oscillations. The analysis in the frequency domain reveals a dominant contribution at about 4 Hz (Fig. 1b). Such frequency peak is ascribable to the pitch vibration mode of the vehicle, as resulting from a comparison of the phases of the narrow-band filtered signals.

A numerical model of the complete forklift truck is implemented within a multibody software environment (namely, MSC Adams). All the vehicle parts are modelled as rigid bodies. Their mass properties are defined on the basis of both CAD models and experimental measurements. The operator is considered as an additional mass rigidly attached to the cabin. Each wheel is connected to its hub by means of spring-damper elements (namely, three translational and two torsional elements) characterized by constant lumped parameters, to take into account the joint compliance. Each cabin bearing is modelled by using three translational spring-damper elements. Contacts between tires and ground/obstacle are modelled with solid-to-solid impact functions, including friction. The other kinematic constraints are modelled as ideal joints.

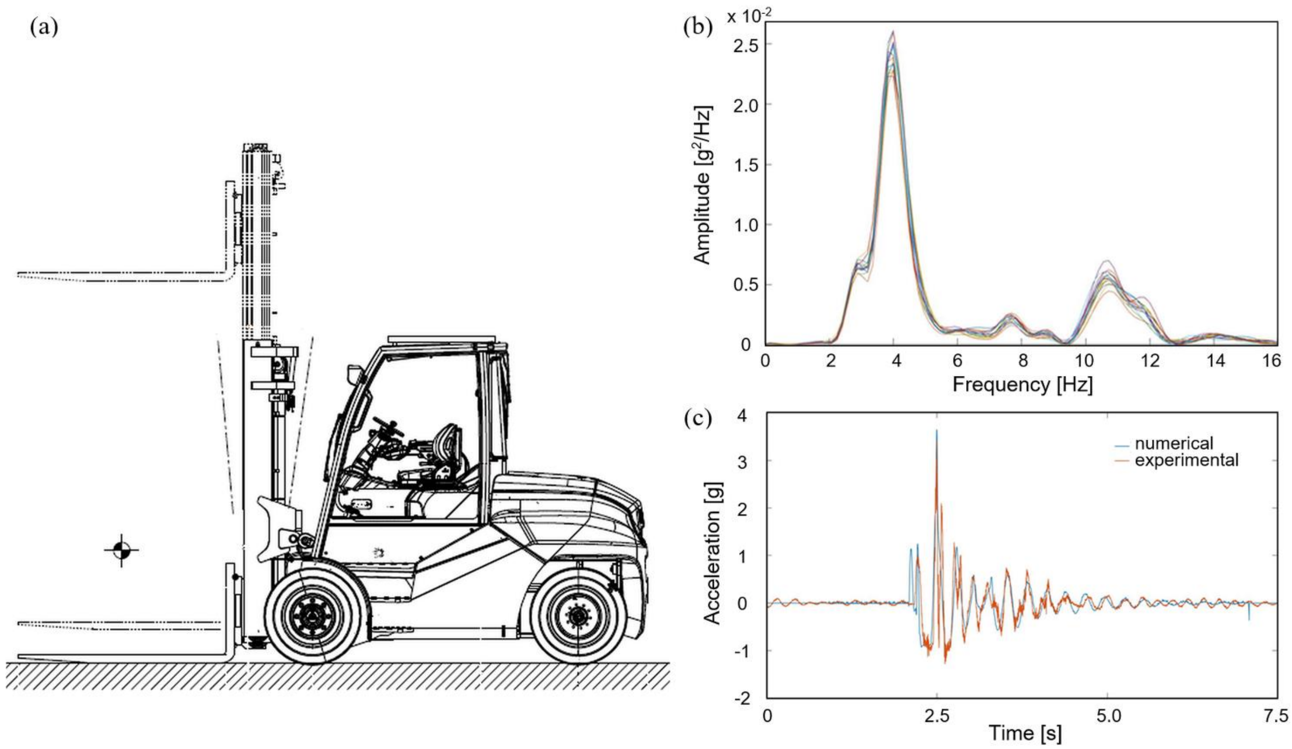

Fig. 1: (a) Schematics of the forklift, (b) power spectra of the acceleration signals of the left front wheel hub, (c) comparison between experimental measurements and updated numerical model.

Simulations are performed by imposing motion on the driving wheels by means of velocity functions, thus replicating the experimental conditions. The computed accelerations of the wheel hubs are compared with the experimental signals to update the model. In particular, the stiffness and damping parameters of tire contacts and wheel bushings are iteratively refined by using optimization tools that aim at matching the amplitude of the two major acceleration peaks and the frequency of the pitch vibration mode. After model updating, the simulation results show a satisfactory agreement with the experimental measurements, as reported in Fig. 1c. An improved model that includes flexible chassis is currently under development.

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References

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