

Coupling Between 1D-3D Simulation Results to Predict Cavitation in Motorcycle Forks

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ABSTRACT

Fork system is a primary component for motorcycles because it assures the contact between tires and road, therefore the safety and the driving feeling. Usually fork optimization and tuning are experimentally made involving the generation of a high large number of prototypes and an expensive experimental campaign. To reduce the design and the tuning phases of a generic damper system, the numerical simulation should be considered.

In this paper, a one-dimensional (1D) model of fore-carriage forks for road applications is presented. The model was built-up in AMESim code. In particular, the authors' attention was focused on the detection and analysis of cavitation phenomenon inside the fork. As well known, the cavitation is a complex three-dimensional (3D) phenomenon that implies the phase transition. Cavitation development and prediction are beyond 1D model capabilities, therefore the 1D fork model was improved by 3D Computational Fluid Dynamic (CFD) multiphase simulations performed by using Fluent v12 (beta version). 3D numerical results were integrated into the 1D model to define a new numerical test methodology able to reproduce the overall fork behaviour.

All 1D numerical results were compared to experimental damping force traces obtained for different fork rod axial velocities.

INTRODUCTION

Nowadays, virtual tests are mandatory to reduce the time to market of new products. Simulation tools are helpful to understand the behavior of single components and their interaction.

Surely, motorcycle front forks are good examples of fluid-mechanic systems that can get benefit from simulation design tools. In fact, the motorcycle forks are made by several components usually arranged inside a very narrow space. It implies the impossibility to place sensors inside the fork system to correctly understand its fluid dynamic behavior.

The state of the art of motorcycle forks modeling is relatively undeveloped. Liberati et al. [1] presented a paper describing the development of a grey-box modeling of a racing motorcycle single tube shock absorber. Martins et al. [2] developed a CFD shock absorber model. They focused their attention on the fluid dynamic analysis of the fluid flow through the clearance between shock absorber piston and tube. Choon-Tae Lee et al. [3] proposed a new mathematical model for simulating displacement-sensitive shock absorber. The vehicle dynamic characteristics of the model were evaluated in time and in frequency domains using a quarter car-simulation model. Falfari et al. [4] presented a 1D model of fore-carriage fork by using AMESim code. Numerical results were compared with experimental tests underlying the good agreement with test data and the prediction capability of the proposed 1D model about the fork performance.

In the present paper, the 1D model approach defined in [4] is adopted to study the fluid dynamic behavior of a new fork prototype designed by PAIOLI MECCANICA. The fork here analyzed presented the overall geometry previously presented in [4] but now it showed to be critical from the cavitation point of view.

Cavitation phenomenon involves the simultaneous presence of liquid and vapour. In a fork, cavitation implies decay of oil performances in terms of damping effect, resulting in a significant change of the damping force evolution during the ride (fast change of vehicle stability). Cavitation depends strongly not only on

geometry and liquid characteristics, but also on the actual operating conditions.

Generally, the cavitation onset and development prediction is beyond 1D model capabilities because of its 3D evolution. The major drawback of 1D modelling is that the critical cavitation number (K_{crit}), at which the transition from a single-phase flow to a two phase flow happens, must be set a priori. To overcome this problem, the 1D fork model set-up was integrated by the data obtained performing 3D CFD simulations of the fork by using Fluent v12 (beta version). In particular, the 3D simulations were useful to determine the value of the critical cavitation number depending on the actual geometry and operating conditions inside the fork and to define the evolution trend of the nozzle discharge coefficient (C_d) under cavitating conditions.

PAPER OUTLINE

This paper was organized in five main areas. In the first part, brief descriptions of the damping rod fork system and 1D fork model are provided.

In the second part, 1D simulation results are presented and compared to experimental results.

In the third part, 3D simulation set-up and cavitation model are presented.

In the fourth part, 3D simulation results are presented and discussed.

In the last part, the integration between 3D simulation and 1D model is presented and 1D results are newly compared to experimental ones.

In the present paper all the data are dimensionless because of PAIOLI MECCANICA discretion needs.

DESCRIPTION OF THE DAMPING-ROD FORK SYSTEM

In the present paper, a prototype fore-carriage fork designed by PAIOLI MECCANICA for motorcycle application was considered. Figure 1 shows the overall fork geometry. The fork layout here considered was the same as in [4] but it was designed to equip motorcycles having different performance. In particular, the fork assembly here considered guaranteed a damping action higher than the fork presented in [4].

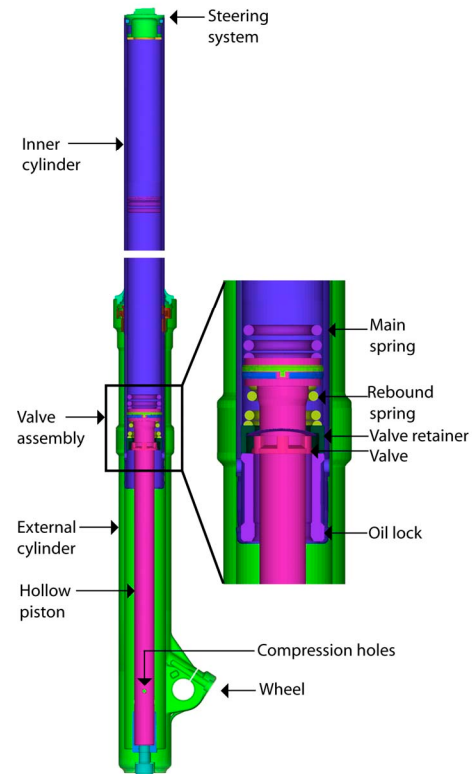


Figure 1: Schematic representation of the considered fork assembly and detail of the fork valve group.

In a typical damping rod fork (as the considered system) the damping action is guaranteed by the combined action of elastic and hydraulic elements:

- Elastic elements (springs) are responsible to maintain the target fork height with respect to the road;
- Hydraulic elements (oil flow through calibrated orifices) are responsible to control the fork axial velocity, damping the oscillations induced by the springs;

For a complete description of the fork behavior during compression and rebound strokes the authors referred to [4].

DESCRIPTION OF THE FORK 1D MODEL

The 1D fork model was built up using AMESim platform. AMESim code is suitable to perform one-dimensional (1D) lump simulation of hydraulic and pneumatic systems [5]. Figure 2 shows a sketch of the AMESim fork model.

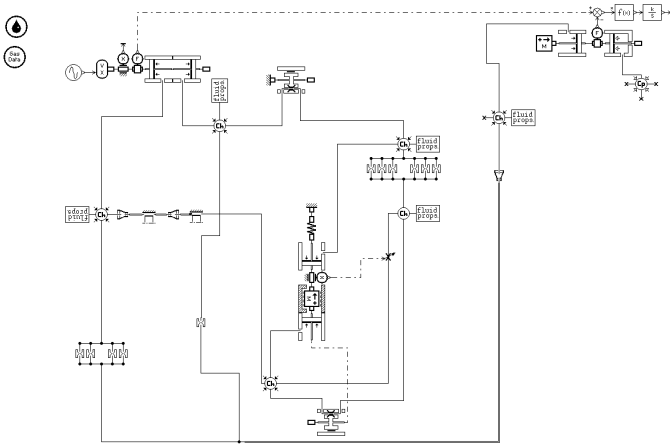


Figure 2: Fork model sketch built into AMESim code.

The overall 1D fork model was subdivided into three different sub-models: mechanical, hydraulic, and pneumatic.

Temperature dependency and friction forces were neglected as in [1-4].

MECHANICAL SUBMODEL

The mechanical damping-rod fork was modeled as a mass-spring-damper assembly applying the dynamic second law:

$$m\ddot{x} + c\dot{x} + k'x = F \quad (1)$$

where m is the mass of each component, c the viscous damping coefficient, k' the elastic constant, and F is the net force applied to the mass along the fork axis direction due to spring, friction, and pressure acting on fork internal surfaces.

HYDRAULIC SUBMODEL

A lump model was adopted to reproduce the flow behavior of elements as: chambers having variable or constant volumes, calibrated orifices (turbulent flow), and clearances characterized by laminar leakage. The basic equation of the hydraulic lump model was the mass conservation equation that for a generic capacity can be written as:

$$\frac{V}{B} \cdot \frac{dp}{dt} + \frac{dV}{dt} = \sum_i Q_i \quad (2)$$

where V is the volume of each chamber, B is the bulk modulus, Q_i is the inlet or outlet volumetric flow rate, and dV/dt is the pumping effect defined as:

$$\frac{dV}{dt} = A \cdot v \quad (3)$$

where A is the moving element cross-section area and v is its velocity.

The fluid leakage flow rate through annular section was defined as:

$$Q = \frac{b \cdot h^3}{12 \cdot \mu \cdot l} \cdot \Delta p \quad (4)$$

where b is the leakage perimeter, h is its height, and l is its length. μ is the dynamic viscosity.

The mass flow rate in rectilinear cylindrical shape orifices was calculated distinguishing two cases:

- Non cavitating orifice;
- Cavitating orifice;

NON CAVITATING ORIFICE

AMESim computes the mass flow rate by using the classic approach:

$$\dot{m} = A \cdot C_d \cdot \sqrt{2 \cdot \rho \cdot (p_1 - p_2)} \quad (5)$$

where:

p_1 is the hole upstream pressure;

p_2 is the hole downstream pressure;

A is geometrical hole area;

ρ is the local fluid density;

C_d is the hole discharge coefficient;

In particular, C_d is calculated as:

$$C_d = \frac{Re}{\lambda} = \frac{Re}{\frac{d_h}{v} \cdot \sqrt{2 \rho \cdot (p_1 - p_2)}} \quad (6)$$

where:

Re is the Reynolds number

λ is the flow number;

d_h is the hole hydraulic diameter;

CAVITATING ORIFICE

For evaluating the orifice discharge coefficient the following expression is used:

$$C_d = C_c \cdot \sqrt{\frac{p_1 - p_v}{p_1 - p_2}} = C_c \cdot \sqrt{K} \quad (7)$$

where:

p_v is the vapor pressure at the local fluid temperature;

K is the cavitation number;

C_c is the hole contraction coefficient evaluated by the empirical expression:

$$C_c = \left(\left(\frac{1}{C_{c0}} \right)^2 - 11.4 \cdot \frac{r}{d} \right)^{-0.5} \quad (8)$$

($C_{c0} = 0.61$ as proposed by Nurick [6] and r/d is the ratio between hole inlet radius and hole diameter).

Substituting Eq. (7) into Eq. (5) the mass flow rate of the cavitating orifice can be expressed as:

$$\dot{m} = A \cdot C_c \cdot \sqrt{2 \cdot \rho \cdot (p_1 - p_v)} \quad (9)$$

Eq. (9) states that the mass flow rate of a cavitating nozzle is independent from the downstream pressure p_2 .

In particular, AMESim uses the cavitation number K' to distinguish if a hole is cavitating or not:

$$K' = \frac{p_1 - p_2}{p_1 - p_v} = \frac{1}{K} \quad (10)$$

And so:

$$K'_{crit} = \frac{1}{K_{crit}} \quad (11)$$

When K' is higher than a critical value (K'_{crit}) the flow through the nozzle is cavitating and the AMESim cavitation model is activated. The main problem of this approach is to set *a priori* the K'_{crit} value. For example, as stated by Schmidt [7], the critical cavitation number K_{crit} for a typical Diesel nozzle under steady fluid dynamic conditions could range from 2 to 1.45 depending on parameters like percentage of non-condensable gases dissolved in the main liquid phase, surface roughness, rounding radius of the hole inlet edge, etc. The authors worked for many years about cavitation analysis into injectors for automotive applications [8,9] identifying the K_{crit} for these applications. However, only a small part of this experience can be applied to the fork analysis because geometry and operating conditions are completely different.

PNEUMATIC SUBMODEL

The fork is always filled of oil except for the upper part of the inner cylinder that is filled of air at the ambient pressure p_i (Figure 1). During the fork oscillations, the air pressure changes according to the fork movement because the fork is sealed. The air pressure decreases during the fork extension and increases during the fork compression. To find the right value of the air pressure at the operating starting point p_f (typically half stroke), an isentropic compression was supposed:

$$p_i \cdot V_i^k = p_f \cdot V_f^k \quad (11)$$

where k is the air isentropic index (equal to 1.4) and V_i and V_f are respectively the initial and final air chamber volumes.

The initial volume V_i of the air pressure was evaluated as:

$$V_i = V_{TOT} - V_{OIL} \quad (12)$$

where V_{TOT} is the total inner fork volume at its maximum extension, and V_{OIL} is the oil volume filling the fork volume.

The expression for the final volume V_f is:

$$V_f = V_i - \Delta V_{stroke} \quad (13)$$

where ΔV_{stroke} is the fork total volume variation related to the effective stroke of the fork itself.

LIMIT OF THE 1D FORK MODEL

The main limits of the fork 1D model previously described were:

- Oil viscosity was kept constant during the simulation. It involved that the foam onset inside the

liquid oil, which usually occurs at the highest axial velocities, was not taken into account. Increasing the foam formation a decay in the damping effect was expected because of the oil viscosity reduction;

- Simulations were isothermal;
- The fork valve lift was supposed to increase linearly with the rod movement, while oscillations around its equilibrium position could be expected;
- The model could not account for manufacturing and assembling tolerances. As a consequence it reproduced the "theoretical" fork;

The overall set-up of the 1D fork model was made on the basis of authors' previous work [4].

1D FORK SIMULATION: NUMERICAL RESULTS

First of all, the fork fluid dynamic behavior was analyzed by the proposed 1D model. To validate the 1D model, numerical results were compared to experimental ones in terms of fork hydraulic characteristic (damping force vs piston rod displacement).

Fork hydraulic characteristic was experimentally determined by PAIOLI MECCANICA using a MTS Shock Absorber Test System to acquire force, displacement, and velocity of the tested fork. Figure 3 shows a single fork vertically positioned on the MTS tester.

Fork tests were performed moving up the piston till the operating point (its half stroke) and then exciting the fork by different axial velocities with a sinusoidal displacement of ± 25 mm centred on the operating point. The same test methodology was used to perform 1D numerical simulations.

Table 1 summarizes all the experimental configurations tested and reproduced by AMESim fork model.



Figure 3: MTS Shock Absorber Test System adopted by PAIOLI MECCANICA to obtain the fork dynamic characteristics.

Case ID	Axial velocity [mm/s]
CASE 1	300
CASE 2	500
CASE 3	600
CASE 4	700
CASE 5	800
CASE 6	900
CASE 7	1000
CASE 8	1500

Table 1: Experimental configurations reproduced by the 1D model.

As previously underlined, the cavitation model was activated for all the 1D fork simulations. It implies that a K'_{crit} value had to be defined. In first instance a value of 0.5 was chosen for the K'_{crit} parameter. This value has to be considered as a first attempt value valid for holes on thin and flat plane.

Figures from 4 to 9 show the comparison between experimental and simulation results. Only some experimental configurations between them reported in Table 1 were reported for seek of brevity: they were believed to be the most representative.

These Figures depicted the asymmetric damping force – piston rod displacement trend of motorcycle forks: the maximum damping force during compression stroke is less than the maximum damping force in the rebound stroke. This asymmetry is due to the design of this particular fork but this kind of trend is not a general rule for forks, although it represents the 90% of motorcycle forks behaviour. Usually it is to consider that during the rebound stroke the damping force must be greater than during the compression stroke because the inner hydraulic system has to brake the main spring force (Figure 1), which is released during the rebound stroke after a previous compression. On the other side during the compression stroke the main work is made by the main spring, while the hydraulic damping acts only after a speed sinking stroke. In order to have more comfort on particular roads, sometimes the main spring stiffness has a low value, of consequence the global damping force – piston rod displacement trend is symmetric.

Figure 5 (fork axial velocity equal to 500 mm/s) shows a good agreement between experimental and simulation results during both compression and rebound strokes. This result could be expected because the 1D model set-up here adopted was derived from an authors' previous work performed on a similar fork geometry [4]. However, differently from the case presented in [4], now increasing the fork axial velocity a disagreement between experimental and simulation results was detected. In particular, Figures 6-9 show that increasing the fork axial velocity, the numerical underestimation of the maximum rebound damping force progressively increases becoming significant when the fork axial velocity exceeds the 1000 mm/s (Figure 8).

By a deep analysis of the 1D simulation results it was possible to identify cavitating conditions ($K' > 0.5$) for all the compression holes during the rebound phase.

Figure 10 shows the K' cyclic evolutions recorded during the fork 1D simulations. As could be noted, passing from 300 to 600 mm/s the K' value became higher than 0.5 during the rebound phase meaning that cavitation structures were detected by AMESim code.

It is important to underline that the information about cavitation obtained by AMESim code could be used only to extract generic fluid dynamic considerations because of the *a priori* definition of the K'_{crit} value.

On the basis of the previous considerations it was clear that 1D numerical set-up had to be improved in terms of:

- Critical cavitation number K'_{crit} ;
- Compression holes discharge coefficient (C_d). This parameter significantly changes from cavitating to non-cavitating condition, therefore a specific C_d profile has to be adopted when cavitation occurs;

To fill these gaps two possible ways could be used:

- Experimental approach;
- Numerical approach by performing 3D CFD multiphase simulations of the fork.

Due to the impossibility to perform a detailed experimental fork fluid dynamic analysis, the second approach was chosen.

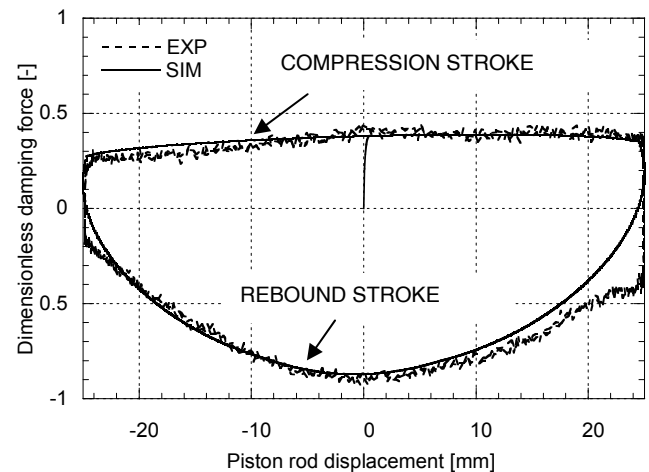


Figure 4: Experimental and simulated dimensionless damping force trace versus piston rod displacement – CASE 3: 300 mm/s.