J. Appl. Comput. Mech., 8(3) (2022) 1091-1102
DOI: 10.22055/jacm.2022.39795.3467

ISSN: 2383-4536
jacm.scu.ac.ir

Shahid Chamran University of Ahvaz

Journal of Applied and Computational Mechanics

Research Paper

Multibody Efficiency Analysis of Chain Drives in Racing Motorcycles

Lorenzo De Rossi[1], Alexander Schramm[2], Alessandro De Felice[3]

[1] Department of Engineering Enzo Ferrari, University of Modena and Reggio Emilia, Via Vivarelli 10, Modena, 41125, Italy, Email: lorenzo.derossi@unimore.it

[2] Department of Engineering Enzo Ferrari, University of Modena and Reggio Emilia, Via Vivarelli 10, Modena, 41125, Italy, Email: alexander.schramm@studio.unibo.it

[3] Department of Engineering Enzo Ferrari, University of Modena and Reggio Emilia, Via Vivarelli 10, Modena, 41125, Italy, Email: alessandro.defelice@unimore.it

Received January 18 2022; Revised February 20 2022; Accepted for publication February 28 2022.
Corresponding author: A. De Felice (alessandro.defelice@unimore.it)

Abstract. In racing motorcycles, the maximization of power transmission from the engine to the rear wheel is one of the critical aspects for improving the performance. Therefore, it is important to improve as much as possible the efficiency of the chain drive, consisting of a front sprocket on the output shaft of the transmission and a rear sprocket connected to the rear wheel, linked by a roller chain. In this study, a multibody model of a chain drive of a racing motorcycle involving high rotational speeds is developed and validated. The energy losses are analyzed, highlighting their dependency on working conditions, and the efficiency is studied as a function of number of teeth on the sprockets, mounting of the chain and sprockets, selected speed ratio, and chain pitch. As a result, it is found that the efficiency is improved by a larger number of teeth with an equal speed ratio, by reducing the chain pitch (while keeping the sprocket diameters constant), and by larger diameter sprockets (in every working condition, including those of high rotational speeds, in contrast to findings in previous literature). Variations in clearances in the chain influence the efficiency, while variations of center distance between sprockets is not influential if the clearances are kept constant.

Keywords: Chain drive; efficiency; multibody analysis; motorcycle dynamics; driveline.

1. Introduction

The most common final drive type found in contemporary motorcycles is the chain drive, consisting of a front sprocket on the output shaft of the transmission and a rear sprocket connected to the rear wheel, linked by a roller chain. The optimization of efficiency of these chain drives is especially important in racing motorcycles, where the maximal transmission of engine power to the rear wheel is critical.

Early studies on roller chain drives focused on approximating the real kinematic behavior at low rotational speed, understanding the importance of modeling sprockets as polygons [1, 2], but only recently a complete kinematic analysis has been provided, considering both chain branches [3, 4].

At high rotational speeds, dynamic analysis of roller chain drives increases in complexity considerably. In particular, studies focused on two topics: vibrations induced on the branches by polygonal effects (including string models [5, 6], multibody models [7, 8], simplified analytical models [9]), and impacts between rollers and teeth [10-12].

The contact between rollers and sprockets has particular importance for the dynamic analysis of roller chain drives. Among the first studies on this topic, in [13] both the geometry and the elasticity of contact bodies were considered, while in [14], the analysis was extended by taking into account the differences in pitch between chain and sprocket teeth due to wear. In [15] the load distribution on teeth was experimentally evaluated varying rotational velocity and tensile load in the slack branch. A method for computing the contact points between rollers and teeth was developed in [16, 17] and a dynamic model able to compute the contact and tensile loads in any working condition was proposed in [18].

One of the positive aspects of roller chain drives, especially for high power transmission, is their efficiency (usually more than 95%). Early contributions focused on meshing impacts and their dependency on speed [10], then the analysis was extended on impact forces due to meshing by taking into account the effects of link stress and chain preloads [12]. More recently, efficiency was evaluated as a function of sprocket tooth count, preload, clearances, lubrication [19, 20] and also link stress [21], showing that high tooth counts and a correct center distance values can improve efficiency in chain drives. A method was proposed in [22] for modifying the tooth profile to minimize meshing impact forces, and for reducing the periodic oscillation of rotational speed. In all mentioned studies, contact friction coefficients were simply estimated, until in [23] a more accurate method was proposed for experimental evaluation of friction at the link articulation level. The analytical models for efficiency estimation were further improved in [24], also including the effects of compliance and damping on the taut branch of the chain, according to [18], achieving a good fit with experimental data. While a different approach was proposed in [25], consisting of a semi-empirical model yielding a complete efficiency map for a roller chain drive, based on both experimental tests, and multibody simulations.

Published online: March 08 2022

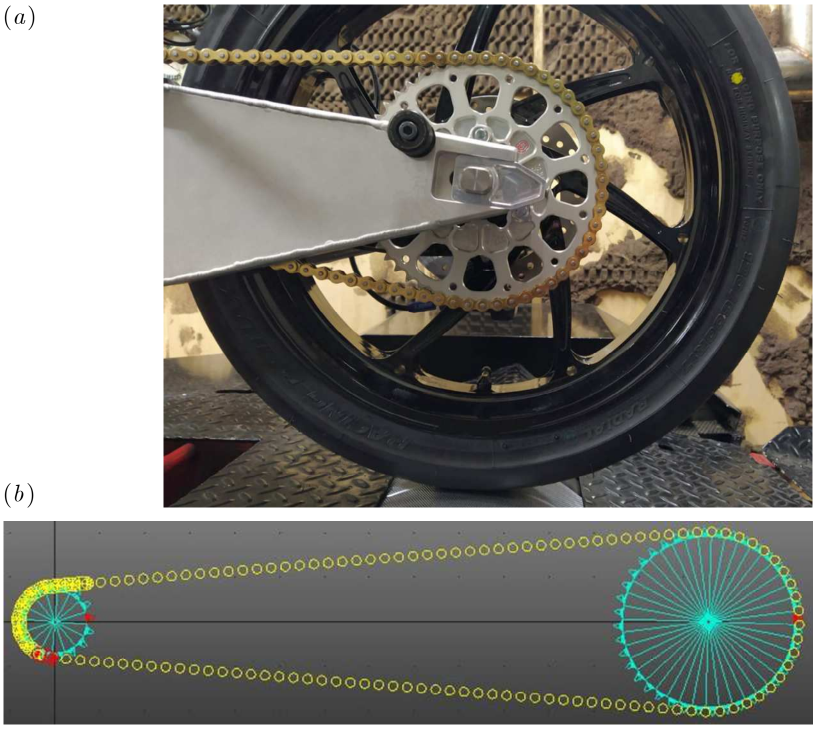

Fig. 1. Chain transmission of a motorcycle (*a*) and schematic of the multibody model (*b*).

The present contribution focuses on deepening the efficiency analysis of roller chain drives for racing motorcycles, characterized by very high rotational speeds. For this purpose, a multibody model of a high-speed roller chain drive has been created using *Adams-View*, providing further insights into the current developments of both efficiency analysis of chain transmissions and multibody analysis applied to motorcycle dynamics [26-28]. The efficiency of the system is studied paying particular attention to the effects of sprocket tooth count, mounting of the chain and sprockets (center distance, clearances, and number of links), selected speed ratio, and chain pitch. The energy losses in the model are analyzed, and their dependency on working conditions is investigated, in order to understand the variations in efficiency due to different configurations of the transmission system.

The paper is organized as follows: Section 2 introduces the multibody model, and discusses the assumptions and approximations made. Section 3 compares the modeled results with manufacturer-provided experimental data to validate the multibody model. Section 4 investigates the effects of center distance, sprocket diameter, drive ratio, and chain pitch on efficiency. Finally, Section 5 analyzes the different loss factors involved in the model, and how the efficiency gains are found.

2. Model Description

The chain drive (Fig. 1*a*) is modeled by means of MSC *Adams-Machinery* (Fig. 1*b*) for evaluating drive efficiency in stationary analysis. To cut long transients before the steady state conditions are reached, the simulation starts with taut branch already preloaded.

2.1 General Description

The model is represented by front and rear sprockets connected to the ground by revolute joints, and a chain with 120 to 128 links depending on the simulation. The swingarm (and the effects of its rotation) as well as all the out-of-plane displacements are not included. The rollers and lateral plates are considered as single rigid bodies, thus reducing the total number of bodies in the model and eliminating the contacts between rollers and bushings. The above assumptions allow a relevant reduction of computational load.

The front sprocket is driven by a constant or variable input torque, simulating torque delivered by an engine through a gearbox. The rear sprocket is loaded by a torque coming from the ground reaction, which is modeled as the rear sprocket rotational velocity squared multiplied by an appropriate gain to account for the effects due to the aerodynamic drag.

All the bodies are modeled as perfectly rigid, and all compliances are located at the interfaces between bodies. Compliance effects are introduced in the model at the contacts between rollers and teeth, and at the joints connecting each of the links. The model is parametrized using the following design variables: coordinates of the front and rear sprockets, longitudinal stiffness of a single link, longitudinal damping of a single link, rotational (viscous) damping between two adjacent links, static friction coefficient (contact between roller and tooth), dynamic friction coefficient (contact between roller and tooth), slip velocity of static friction between roller and tooth, transition velocity of dynamic friction between roller and tooth, contact stiffness between roller and tooth, contact damping between roller and tooth, and exponent of nonlinear stiffness relation between roller and tooth.

2.2 Modeling the Front and Rear Sprockets

The front and rear sprockets are coplanar, with parallel rotation axes. This is a simplifying assumption, which here is considered valid also during maneuvers involving high values of lateral forces, and at high values of roll angle of the motorcycle [29]. The front and rear sprockets are dimensioned conforming to ANSI B29.1 [30]. According to this standard the tooth profile is obtained by 3 circular arcs plus a straight segment, all tangent each other. Starting from the root of the tooth, there is a first circular arc called the “seating curve”, followed by a second circular arc called the “working curve”, then a straight segment and finally the last circular arc called the “topping curve”, as shown in Fig. 2. An approximate profile is adopted, consisting of only two tangent circular arcs (similar to ISO 606 [31]), to reduce computational load.

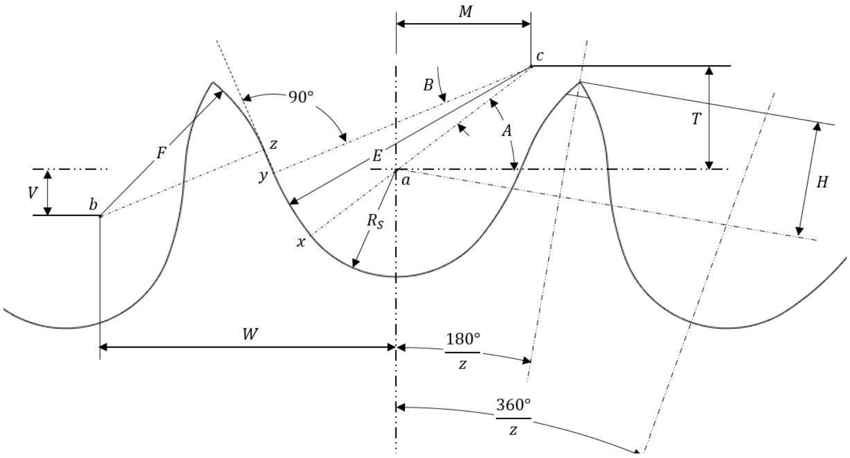

Fig. 2. Tooth profile according to the standard ANSI B29.

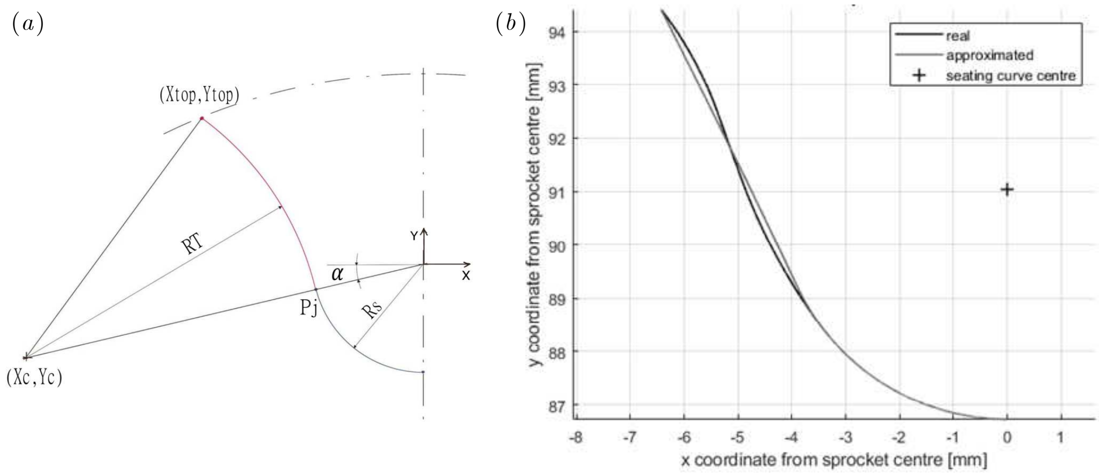

Fig. 3. Schematic for construction of the approximated tooth profile (*a*). Approximated versus real profile for a Z45 12.7 mm pitch profile (*b*).

The approximated profile is constructed using the following assumptions (with reference to Fig. 3*a*):

- The arc of the seating curve has its center on the nominal diameter of the sprocket, its radius equal to that given in ANSI B29.1.
- The two arcs describing the profile are tangent at point PJ.
- The point (X_{top}, Y_{top}) at top of the tooth profile is the same for both the actual and approximated profiles.

Given the above assumptions, with a fixed tooth-side-angle α, the problem reduces in finding the upper arc such that: it has its center on a line which passes through *PJ* and the center of the seating curve, inclined by an angle α; it passes through point (X_{top}, Y_{top}). The solution is unique and can be found after some geometrical manipulations. Referring to Fig. 3*a*, the coordinates of point (X_c, Y_c) are given by:

$$X_c = \frac{1}{2}\frac{\left(X_{top}{}^2 - X_{Pj}{}^2\right) + \left(Y_{top}{}^2 - Y_{Pj}{}^2\right)}{\left(X_{top} - X_{Pj}\right) + \tan\left(\frac{\pi}{2} - \alpha\right)\left(Y_{top} - Y_{Pj}\right)}, \quad Y_c = \tan\left(\frac{\pi}{2} - \alpha\right) X_c \tag{1}$$

and the radius of tooth side RT by:

$$RT = \sqrt{(X_{Pj} - X_c)^2 + (Y_{Pj} - Y_c)^2} \tag{2}$$

Therefore, the only parameter that needs to be selected is the angle α, which is chosen in order to get the best approximation of the actual profile. An optimization algorithm leads to the solution of the approximated profile. In Fig. 3*b* an example of a profile approximation is shown (Z45, 12.7 mm pitch).

2.3 Modeling of the Chain

A roller chain consists of a sequence of two alternating kinds of links: internal (inner) links and external (outer) links. Internal links are narrower and usually include two lateral plates, two bushings, and two rollers; the external links include two lateral plates and two pins.

In the model, internal and external links are assumed to have the same mass, but different moments of inertia, according to the different shape of the links (geometrical parameters reported in Table 1).

For modeling the compliance and elastic properties, the *Adams* “linear” option is selected. The parameters involved are the longitudinal stiffness K_{tx} of a link along the chain line, the longitudinal damping coefficient c_{tx} of a link along the chain line, and the torsional damping coefficient c_{rz} in the relative rotation between two adjacent links.

Table 1. Main parameters of the 520-pitch chain.

Pitch	15.875 mm	Height internal plate	15.00 mm
Roller diameter	10.32 mm (max)	Thickness external plate	2.00 mm
Internal link length	6.35 mm	Thickness internal plate	2.00 mm
Pin diameter	5.50 mm	Weight per meter	0.95 kg/m
Pin length	18.40 mm	Mass of one link	15.03 g/link
Height external plate	13.00 mm	Stiffness	8.55 kN/mm*

*With reference to 16 links. Stiffness of single link = 136.8 kN/mm

The stiffness K_{tx} is computed knowing the stiffness per unit length of the chain (kN/mm) experimentally measured (by the manufacturer) on a traction machine with a chain segment 10 inches (254 mm) long. Then an equivalent stiffness of a series of links is obtained (say K_{eq}) and the stiffness of a single link can be computed once the number of links involved in the test (say n) is known:

$$\frac{1}{K_{eq}} = \frac{1}{K_1} + \ldots + \frac{1}{K_n} = \frac{n}{K_{tx}} \quad \Rightarrow \quad K_{tx} = nK_{eq} \tag{3}$$

Generally, the chain pitch is a fraction of inch, and a segment 10 inches long contains an integer number of links.
The longitudinal damping coefficient between two links (representing the internal damping of the material of lateral plates and pins) is estimated as a reasonable fraction of the critical damping of the system (less than 3% of the critical value is considered). For a chain of the same length as that adopted for simulations the critical value is computed as:

$$c_{cr} = 2\sqrt{M_t K_{eq}} = 2\sqrt{mK_{tx}} \tag{4}$$

where M_t is the total mass of the chain and m is the mass of a single link. In the model, a chain is considered with pitch equal to 15.875 mm, 126 links, stiffness of single link K_{tx} = 136.8 kN/mm, and mass of single link m = 15.03 g, yielding a critical damping value c_{cr} = 361 Ns/mm. The value for the longitudinal damping coefficient of the link, after some tests for assessing the reliability of the results, is therefore selected as c_{tx} = 10 Ns/mm.

The last parameter to evaluate is the torsional damping associated with bushings. The articulation between two rings is given by an internal, lubricated contact between cylinders. In addition, there are elastomeric elements which contribute to friction. Moreover, it is known from data provided by the manufacturer that the relative rotation between loaded links requires a certain amount of torque. Even though a viscous damping model is not appropriate, in the literature is common practice modeling this contact by a rotational viscous damping. Therefore, a method is here adopted, as suggested in [8], to compute the friction torque in the articulation, having fixed the load on the chain and the friction coefficient. This dry friction torque must be equal to an equivalent viscous damping torque, from which the rotational damping coefficient can be obtained. Assuming a reasonable value for the dry friction coefficient (μ = 0.1 for a lubricated contact between steel surfaces), an equivalent torsional damping coefficient is found (c_{eq} = 75 Nms/°). This choice is then numerically assessed by comparison of the multibody simulations with experimental results.

The contacts between rollers and teeth are modeled as unilateral constraints. In the developed model the *Adams* function *impact* has been adopted. This sub-model is based on the elastic nonlinear contact according to Hertz theory, with an additional term to model the energy loss due to impacts. The normal force can be estimated according to:

$$F_N = k_c \delta^h + c_c \frac{d\delta}{dt} \tag{5}$$

in which k_c is the contact stiffness, h is an exponent determining the nonlinear elastic reaction, c_c is a viscous damping coefficient, δ and its first derivative with respect to time are the depth and the penetration speed between the two bodies (details on contacts involving linear viscoelastic materials are given for instance in [32]). The term related to the penetration speed is multiplied by a step distribution to model the discontinuous contact force during the impact between the two bodies.

Using the computational procedures suggested in [33-35] the adopted values for a 520-pitch chain are: $k_c = 10^5$ kN/mm, c_c = 1 Ns/mm, and n = 2.2.

In regard to the modeling of friction between rollers and the side of sprocket teeth, as usual in multibody modeling, a function is introduced for eliminating the discontinuity in the friction coefficient at zero velocity. To reduce the computational load, a singular value for the friction coefficient is adopted equal to 0.35 (static and dynamic).

3. Validation

The efficiency is estimated in several different working conditions of the roller chain drive using the multibody model. The simulated results are compared with experimental data produced by the manufacturer of the chain.

3.1 Experimental Data

The comparison is performed using a 520-pitch chain (in the following the pitch will be expressed in mm according to international standard). Its main parameters are reported in Table 1.

An efficiency map of the chain is obtained by applying different input values of torque and rotational speed at the front sprocket, which are representative of the actual working conditions of the chain.

3.2 Experimental and Simulated Efficiency

The simulated efficiency (ratio between front and rear sprocket mechanical power) is generally noisy, mainly due to polygonal effect. To reduce such noise a low-pass Butterworth filter of the third order is applied with a cutoff frequency of 10 Hz. If the model is not running in stationary conditions, the computation of efficiency would also be affected by errors. In numerical simulations, the moments of inertia of the sprockets are used as parameters for setting the time interval necessary for the system to reach stationary conditions. The simulated efficiency is compared with experimental results in Fig. 4, showing very good agreement. Nevertheless, it should be noted that the computed efficiency has an almost linear behavior with respect to rotational speed, while the experimental data have regressive curves with respect to decreasing rotational speed.

Table 2. Modeling data for front sprocket with Z = 16 and rear sprocket with Z = 40.

	Front Sprocket Z = 16	Rear Sprocket Z = 40
Radius of seating curve	5.14	5.14
Angle of tooth side α (°)	62.80	65.19
Radius of tooth side RT (mm)	43.27	102.27

Table 3. Parameters of modified teeth profiles.

Front Sprocket Z = 16	–4°	–1°	standard	+1°	+2°
Angle of tooth side α (°)	58.8	61.8	62.8	63.8	64.8
Radius of tooth side RT (mm)	840.12	57.78	43.27	34.30	28.22
Rear Sprocket Z = 40	–1,89°	–1°	standard	+1°	+2°
Angle of tooth side α (°)	63.30	64.19	65.19	66.19	67.19
Radius of tooth side RT (mm)	3072.90	213.06	102.27	66.42	48.70

3.3 Tooth Profile

The adopted tooth profile is an optimized approximation as explained in Section 2. The sensitivity of the results is tested with respect to slight variations in profiles. Since the radius of the seating curve is fixed equal to that of the standard, profile modifications are obtained by varying the tooth-side-angle (once the angle has been fixed, the radius is defined as well). A front sprocket with 16 teeth is considered, together with a rear sprocket with 40 teeth with data as reported in Table 2. Modified profiles with larger angle and with reduced angle are generated as reported in Table 3. The last configuration is the most extreme (beyond that, the profile becomes straight, and then concave).

The observed parameters to evaluate the effect of tooth profiles are efficiency, power loss due to link rotation, peak contact force on the sprocket, and average contact force on the sprocket.

To ensure robustness of the results, several simulations are performed at different input values of torque and rotational speed at the front sprocket. It is found that all the observed parameters remained nearly constant when varying the tooth profiles according to the data reported in Table 3. The plot in Fig. 5 shows the efficiency as a function of the tooth-side-angle of the front sprocket at different values of torque and rotational speed. Notice that the efficiency is not influenced by slight variations of the tooth profile (maximum differences in efficiency are about 0.05%).

4. Effects of Center Distance, Speed Ratio and Pitch

In this section, the efficiency is studied as a function of the main parameters characterizing the chain drive: center distance, number of links, clearance, number of teeth, speed ratio and pitch.

4.1 Center Distance

The center distance of the chain drive varies due to relative rotation between swingarm and frame. The total number of links (here 120), and therefore the total length of the chain, can be slightly varied depending on the desired speed ratio and on the length of the swingarm. It is important to assess if the estimated efficiency remains unchanged after slight variations in the number of links (keeping the transverse clearance constant), or after slight variations in the transverse clearance (keeping the total number of links constant). This assessment is important because during motion the transmission rarely works in purely stationary conditions. Vibration modes involving small variations in center distance are usually excited by external inputs, which under particular conditions are enhanced due to the occurrence of self-excited phenomena, such as chatter of front patter [36,37], both of which affect the in-plane dynamics of the motorcycle. The first analysis regards small variations in the number of links with constant clearance, therefore the center distance has been computed for each separate number of links. Starting from a configuration with 120 links and center distance equal to 727.6 mm, simulations are performed with 122 and 124 links, and their relative center distances. In all cases the transverse clearance is kept constant and equal to 20 mm. The results are shown in Table 4.

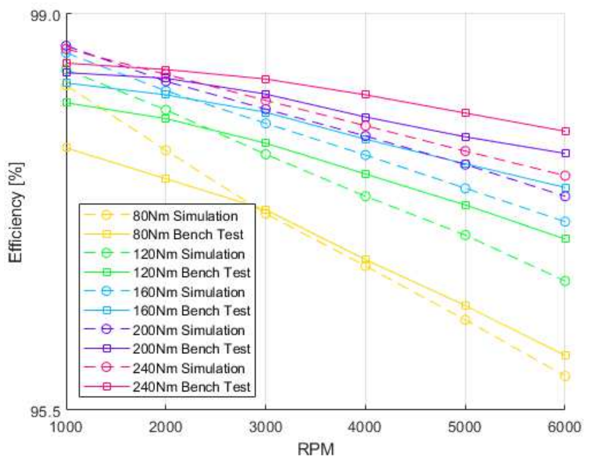

Fig. 4. Experimental and simulated efficiencies (pitch: 15.875 mm).

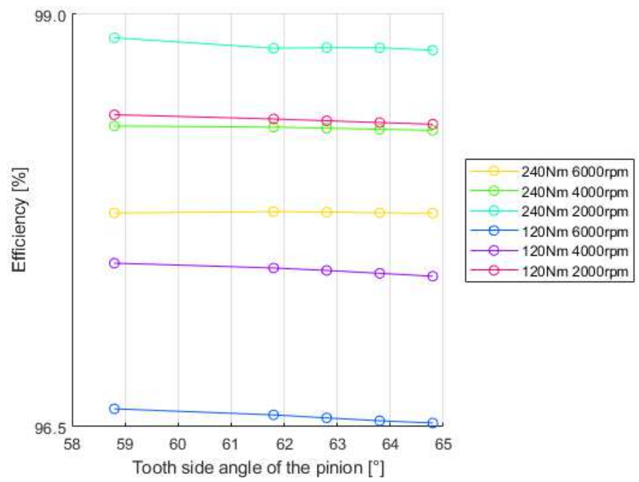

Fig. 5. Efficiency as a function of the tooth-side-angle.

Table 4. Effect of center distance on efficiency
(at constant clearance: 240Nm, 6000 rpm, transverse clearance 20 mm).

Number of Links	Center Distance [mm]	Efficiency [%]
120	727.6	97.69%
122	743.5	97.69%
124	759.5	97.70%

Table 5. Effect of center distance on efficiency (at constant number of links: 16/40, 120 links).

Center Distance [mm]	Preload [N]	Transverse Clearance [mm]	Efficiency [%] 240Nm 6000rpm	Efficiency [%] 240Nm 4000rpm	Efficiency [%] 120 Nm 4000rpm
725.6	-	78.3	97.75%	97.82%	97.62%
726.6	-	57.0	97.73%	98.09%	97.62%
727.6	-	18.8	97.69%	98.10%	97.53%
727.7	0	0	97.68%	98.09%	97.50%
728.0	631	-	97.66%	98.07%	96.68%
728.6	1994	-	97.56%	98.00%	96.63%
729.6	4267	-	96.66%	97.07%	94.43%

It can be seen that the efficiency remains practically unchanged with respect to the standard reference configuration. This is an expected result, since losses are concentrated in the relative rotation of meshing links, and in the contacts between rollers and teeth.

The effect of clearance is also investigated. A chain with 120 links is considered, and the center distance is modified in order to simulate clearances larger than the value of 20 mm, as used in the reference case, or with zero clearance and additional preload.

In motorcycles the chain is never actually mounted with preload, however it may happen that the rear hop produces an increase in center distance, so that the mounting clearance is reduced to zero, and the chain works in preload conditions. The results of these simulations are reported in Table 5. It clearly shows that for small variations of clearance, up until small values of preload, the efficiency remains nearly the same. Efficiency tends to reduce for excessive values of clearance, and this is probably due to the dynamics of the slack branch, involving large amplitude oscillations.

Increasing the preload produces a great loss of efficiency. This effect is due to dissipative effects in contacts between rollers and teeth, and a very high preload produces a strong increase in the contact forces and their related losses. It can be noted that for intermediate preloads, the efficiency does not vary significantly with respect to its value in case of chain with clearance. This can be explained by the fact that the chain is a compliant system, and its taut branch tends to stretch. Therefore, the clearance in working conditions (with applied torque) is larger than the mounting clearance.

4.2 Number of Teeth at a Fixed Speed Ratio

The question is if it is more efficient using sprockets with larger or smaller diameters once the speed ratio has been fixed. It is known that larger tooth counts improve the homokinetic behavior of the drive and reduce impacts and vibrations due to a reduced polygonal effect.

A comparison is made among 5 different front and rear sprocket setups, all of them with a speed ratio almost equal to 0.40, and with center distances and chain lengths adjusted to achieve the same mounting clearance. The selected setups are the following: ratio 16/40 (τ = 0.400, 120 links, center distance 727.6 mm); ratio 18/45 (τ = 0.400, 124 links, center distance 730.5 mm); ratio 15/37 (τ = 0.405, 118 links, center distance 728 mm); ratio 19/47 (τ = 0.404, 125 links, center distance 726.7 mm); ratio 19/48 (τ = 0.396, 126 links, center distance 730.1 mm). All setups are tested at 3 different values of applied torque, and several different values of rotational speed. Some results are reported in Fig. 6.

The results show that the use of sprockets with larger diameters yield higher efficiency values in every working condition. Such a result partially contradicts [20], which claims that at high speeds (like in racing motorcycles) it is more efficient to use sprockets as small as possible. The conclusion of that study, however, was drawn based on a simplified model for estimating efficiency, in which the only losses are those due to friction between bushings and pins.

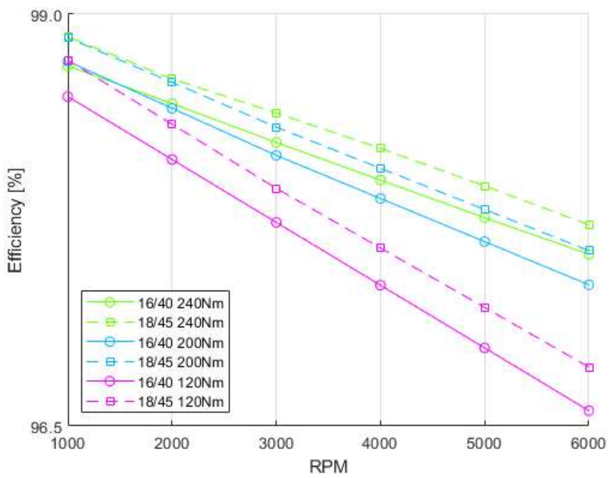

Fig. 6. Efficiency comparison: 16/40 and 18/45, with the same chain clearance.

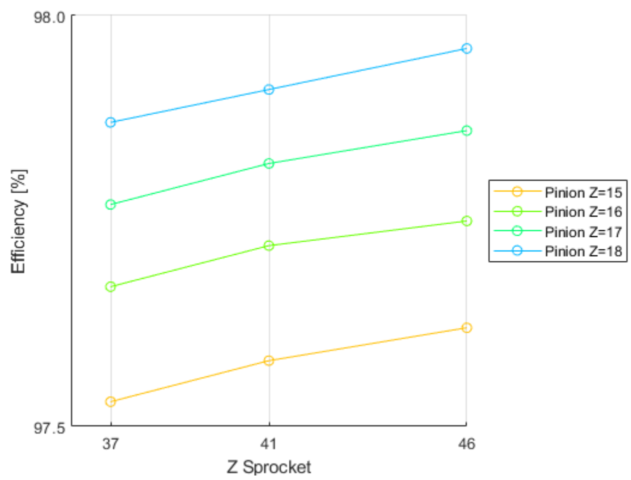

Fig. 7. Efficiency as a function of the number of teeth Z of the sprockets.

The use of larger diameter sprockets could be objected to since they have a larger inertia, however, it should be noted that the mass of sprockets in any case is very low and cannot significantly affect the dynamics. A final comment regards the load in the taut branch of the chain: chains in racing motorcycles are subjected to loads that often overcome their fatigue limits, so a larger diameter sprocket allows to reduce the load, and therefore the possibility of wear and breaks. In conclusion, the use of sprockets with larger diameters are always preferable in a racing motorcycle.

4.3 Speed Ratio

There is an efficiency associated with each speed ratio of the final drive (given by the selected tooth-counts of the sprockets). This value especially comes into play in those maneuvers in which power delivered by the engine is the limiting factor (the so called "engine limit" phase).

Having a complete set of front and rear sprockets, the choice in these numerical simulations is to use all available front sprockets, associating 3 different rear sprockets for each. As a simplifying assumption, a single point (torque, RPM) is identified as the characteristic of the "engine limit" for computing drive efficiency. The results are displayed in Fig. 7.

Once a reference speed ratio is fixed, adding one tooth to the front sprocket produces an increase of efficiency comparable to that which could be obtained by adding 10 teeth to the rear sprocket. Therefore, the dimension of the front sprocket is much more influential on efficiency than that of the rear sprocket.

4.4 Pitch

An increase in chain pitch always gives an increase in both strength and weight. Recently chains with a reduced pitch are available that have nearly the same weight and section of chains with larger pitch (for reaching the same overall strength). However, there are several advantages of reduced pitch (and a reduction of mass and/or section, among them, would be of almost negligible importance). The first important aspect is that the sprockets (with parity of teeth) are smaller if adopting a reduced chain pitch. For the same reason, comparing sprockets with parity of diameters, those with reduced pitch will have a larger number of teeth. Moreover, using a reduced pitch yields a larger number of possible speed ratios within a given range. A chain with pitch equal to 12.7 mm with the same strength properties as those of the chain under study (with pitch equal to 15.875 mm) will now be considered.

After modifying the model parameters for the reduced pitch chain (12.7 mm), the same steps for validating the model as in Sections 2 and 3 are performed. The only difference is the selected number of teeth: the speed ratio is held the same as the other chain with the diameters as close as possible to the previous ones, and the selected number of teeth is 20 for the front sprocket and 50 for the rear sprocket. In the selection of number of teeth, the dimensions (diameter) are kept constant, since reducing the pitch while keeping constant the number of teeth would have resulted in the sprockets being too small. The results of numerical simulations are also in this case in good agreement with the experimental data given by the manufacturer. They confirmed that the reduced pitch chain has higher efficiency with respect to the standard pitch chain in every working condition.

However, it must be noted that this comparison is made with very different tooth-counts, and it is already shown how deeply the number of teeth can influence drive efficiency. To verify that the above-mentioned difference in efficiency is only due to different numbers of teeth, a comparison between the two chains with different pitch is made keeping constant the number of teeth. For avoiding having the sprockets being too small, the selected number of teeth is 18 for the front sprocket and 45 for the rear sprocket (for both chains). The results show that the two drives have very similar efficiency values (though not exactly the same), with maximum differences about 0.15% (Fig. 8).

5. Efficiency Analysis

The loss factors are identified, and then separately analyzed for evaluating the impact of each on the total efficiency for different simulation conditions. The loss factors in the model are torsional damping between links, longitudinal damping between links, contact damping, and Coulomb friction in the contact between roller and tooth.

5.1 Analysis of Loss Factors

The damping torque between adjacent links (say T_r) and their relative rotational speed (ω_{link}) yield instantaneous power loss due to rotational damping at the articulation, which is given by:

$$P_{r_{1\,link}} = T_r \omega_{link} \tag{6}$$

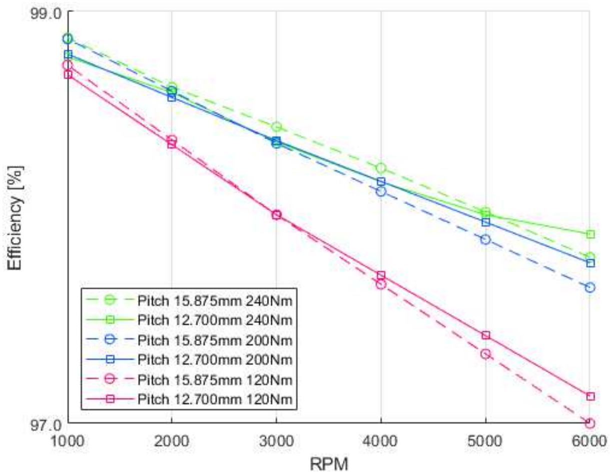

Fig. 8. Efficiency comparison between "short pitch" and "standard pitch" chains, with an equal ratio of 18/45.

Table 6. Drive efficiency for different loss factor settings (16/40, 240Nm, 6000rpm).

		Normal	No friction	No rot. damping	No trans. damping	No cont. damping
Efficiency	%	97.69	98.55	98.58	97.71	97.74
Total power loss	W	3483.4	2191.0	2145.7	3459.0	3403.2
Difference	W	0	−1292.3	−1337.6	−24.4	−80.1
Difference percent	%	0	−37.1	−38.4	−0.7	−2.3
Power loss rotation	W	1501.6	1729.7	176.6	1531.6	1521.9
Power loss longitudinal	W	49.9	38.9	71.1	3.0	143.2
Power loss friction & contact	W	1931.9	422.4	1898.1	1924.3	1738.1

The stretching speed (say $d\delta/dt$), and the longitudinal damping coefficient (c_{tx}) are the factors of the instantaneous power loss in the links, which can be expressed as:

$$P_{t_{1\,link}} = c_{tx}\left(\frac{d\delta}{dt}\right)^2 \tag{7}$$

Integrating in time by considering a complete revolution of the chain, the dissipated energy of a link during a single revolution is obtained:

$$E_{i_{1\,link}} = \int_t^{t+T} P_i\, dt \tag{8}$$

where P_i is the instantaneous power loss on a single link due to a loss factor and T is the time period for a complete revolution of a link. The total power loss for each of the two loss factors can then be computed by multiplying Eq. (8) by the number of revolutions per second (RPS) of the sprocket, and its number of teeth (Z):

$$P_{i_{tot}} = E_{i_{1\,link}} RPS_{sprocket} Z_{sprocket} \tag{9}$$

To identify the contribution to the total power loss of contact damping and friction between rollers and teeth, each of the 4 loss factors are eliminated separately in simulations and compared to the results of the standard reference chain drive. This approach is applied to three different chain models:

1) 18/45 ratio, 15.875 mm pitch, 240 Nm applied torque, and 6000 RPM
2) 16/40 ratio, 15.875 mm pitch, 240 Nm applied torque, and 6000 RPM
3) 20/50 ratio, 12.7 mm pitch, 240 Nm applied torque, and 6000 RPM

For each chain model, five different simulations are performed, eliminating (or strongly reducing) a single loss factor in each:

1) standard model
2) no friction between rollers and teeth
3) rotational damping reduced by a factor of 1/80
4) longitudinal damping reduced by a factor of 1/1000
5) contact damping reduced by a factor of 1/100

The reduction factors of parameters are selected for preserving the numerical stability of the model, and at the same time for reducing as much as possible the loss factor of interest.

A single set of results is reported in Table 6, which is representative of the general trend of all other numerical tests performed. It is possible to see that the loss factors influence each other, and eliminating one causes an increase in one or more of the remaining. The simulation with “no friction” and the one with “no contact damping” do not allow to clearly distinguish the respective contributions in terms of power losses. All simulations, except for the one with reduced rotational damping, give consistent results about the power loss due to relative rotation between links. All simulations give consistent results about the power loss due to longitudinal damping.

The power losses due contact and Coulomb friction cannot be evaluated separately with precision and therefore it is advisable to evaluate them as a unique contribution by subtraction from the total power loss. It is also important to recall that the links are modeled with simplifying assumptions, as already discussed in Section 2.

In literature the larger loss factor has been identified as the relative rotation between links (especially if loaded), and some analytical models including this factor were developed, such as the one in [21]. On the other hand, here it is put in evidence that the loss factor due to contact between roller and tooth is very relevant as well.

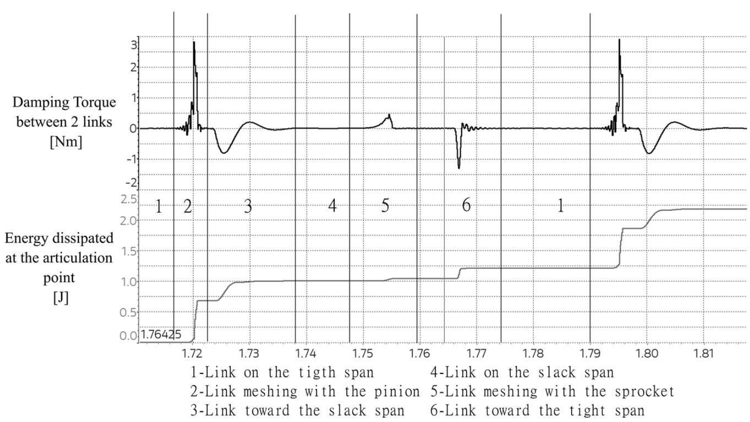

Fig. 9. Energy dissipated by one link (grey) and damping torque (black) during a single revolution.

The performed simulations show that the power loss due to relative rotation increases by increasing the rotational speed (with fixed applied torque); at a fixed rotational speed it is independent from the applied torque. It is slightly reduced by increasing the number of teeth, it is larger for the input and output links meshing with the front sprocket, and smaller for the analogous ones at the rear sprocket. Figure 9 shows the energy dissipated due to relative rotation.

Regarding the effects on efficiency due to variations of applied torque and rotational speed, the influence of working conditions is investigated on power loss due to different loss factors. Two sets are considered for the chain drive:

1) 16/40 ratio, 15.875 mm pitch, 120 links
2) 20/50 ratio, 12.7 mm pitch, 150 links

The conclusions are: the power loss factors due to relative rotation and to contact & friction represent more than 90% of total power loss and the power loss due to longitudinal damping is nearly negligible. At low rotational speed and high applied torque, the dominant loss factor is contact between roller and tooth, and at high rotational speed and low applied torque the dominant loss factor is the relative rotation between links. Then for motorcycle applications, the optimization of tooth profiles is not sufficient, and it is necessary to take into account the reduction of friction in the contact between pins and bushings.

5.2 Analysis of Differences in Efficiency

The differences in efficiency between a "long" and "short" pitch are investigated. A ratio of 16/40 with "long pitch" (15.875 mm) is compared with a ratio of 20/50 with "short pitch" (12.7mm). These two settings have almost identical diameters and speed ratios. Comparisons are made by setting several different values of applied torque and rotational speed as in the previous section. The results are displayed in Figs. 10 and 11.

The power loss due to rotation and that due to contact are very different in the two models.

The reduction in power loss due to rotation is explained by considering that the links at the ends of the chain branches rotate with smaller angles, given that the pitch of the 20/50 sprockets is a "short" pitch which is much lower than the 16/40 sprockets.

The reduction in power loss due to contact is mainly due to friction. It is clear that in all simulations of the model with a ratio of 20/50 the contact forces between roller and tooth are smaller than those in the model with ratio 16/40. A lower contact force between rollers and teeth in a "short pitch" chain can be explained by accounting for the different pressure angles of the teeth. It should also be considered that a shorter chain pitch yields to smaller teeth with respect to the standard pitch. Consequently, during meshing the distance which the roller slides on the side of the tooth is shorter. In conclusion, a reduction in chain pitch yields a reduction in power loss due to both relative rotation between links and contact between rollers and teeth. Therefore, keeping constant the diameters of the sprockets, the efficiency of a "short pitch" chain is always higher than that of a "long pitch" chain.

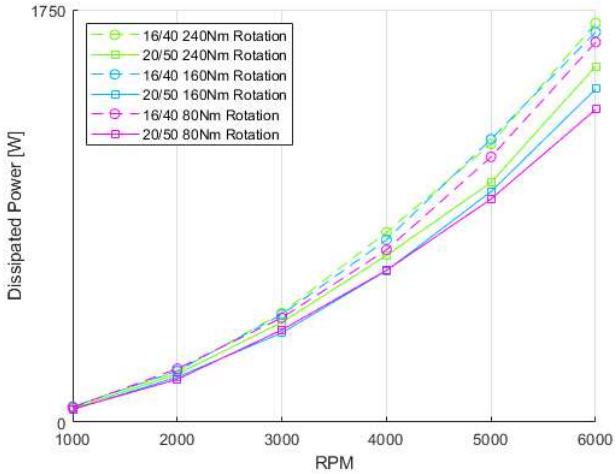

Fig. 10. Comparison of power loss due to relative rotation between links, for a ratio of 16/40 with 15.875 mm pitch and a ratio of 20/50 with 12.7 mm pitch (parity of applied torque and rotational speed).

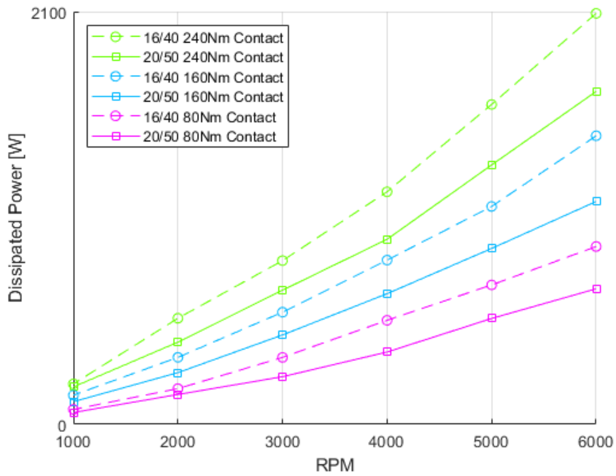

Fig. 11. Comparison of power loss due to contact between roller and tooth, for a ratio of 16/40 with 15.875 mm pitch and a ratio of 20/50 with 12.7 mm pitch (parity of applied torque and rotational speed).

6. Discussion

The developed multibody model allows the deepening of the efficiency analysis of chain drives with respect to the existing literature, focusing specifically on racing motorcycle applications. The efficiency is studied with respect to the relevant parameters of a chain drive, and the main sources of energy losses are determined under typical working conditions.

Regarding the effects on efficiency of each of the chain drive parameters, the following results were found:

- *Tooth profile*: the efficiency is not influenced by slight variations of the tooth profile, with maximum differences of about 0.05%.
- *Center distance*: for small variations in the number of links with constant clearance, the efficiency remains practically unchanged, as well as for small variations of clearance, up until small values of preload; efficiency tends to reduce for excessive values of clearance.
- *Number of teeth at a fixed speed ratio*: the use of sprockets with larger diameters yields higher efficiency values in every working condition, which partially contradicts [20]. Since chains in racing motorcycles are subjected to loads that often overcome their fatigue limits, a larger diameter sprocket allows the reduction of force and therefore the use of sprockets with larger diameters is always preferable for a racing motorcycle.
- *Speed ratio*: the dimension of the front sprocket is much more influential on efficiency than that of the rear sprocket.
- *Pitch*: the first important aspect is that the sprockets (with parity of teeth) are smaller if adopting a reduced chain pitch; for the same reason, comparing sprockets with parity of diameters, those with reduced pitch will have a larger number of teeth, yielding a larger number of possible speed ratios within a given range. A reduction in chain pitch yields a reduction in power loss due to both relative rotation between links and contact between rollers and teeth; therefore, keeping constant the diameters of the sprockets, the efficiency of a "short pitch" chain is always higher than that of a "long pitch" chain.

Regarding the analysis of energy losses in the model, the following loss factors are considered: torsional damping between links, longitudinal damping between links, contact damping, and Coulomb friction in the contact between roller and tooth, where the power losses due contact and Coulomb friction were evaluated as a unique contribution.

In literature the larger loss factor has been identified as the relative rotation between links, especially if loaded [21]; however, in this study it is shown that the loss factor due to contact between roller and tooth is very relevant as well.

The power loss factors due to relative rotation and to contact plus Coulomb friction are found to represent more than 90% of total power loss, and the power loss due to longitudinal damping is found to be nearly negligible. It is found that at low rotational speed and high applied torque the dominant loss factor is contact between roller and tooth, and at high rotational speed and low applied torque the dominant loss factor is the relative rotation between links. The performed simulations show that the power loss due to relative rotation increases by increasing the rotational speed (with fixed applied torque), and that at a fixed rotational speed it is independent from the applied torque. Therefore, for motorcycle applications, the optimization of tooth profiles is not sufficient, and it is necessary to consider the reduction of friction in the contact between pins and bushings.

7. Conclusion

The efficiency of a roller chain drive of a racing motorcycle was studied by developing a multibody model, and analyzed with respect to its relevant parameters, also investigating which are the main sources of energy losses. Variations of center distance are not influential on efficiency if the clearances in the chain are kept constant, while variations in clearances influence the efficiency: too much clearance or too much mounting preload reduces the drive efficiency. A larger number of teeth with an equal speed ratio yields higher efficiency and reducing the chain pitch (while keeping constant the diameters of the sprockets) always improves the efficiency. Regarding the last two points, it can be stated that: (1) the improvement in efficiency obtained by using sprockets with larger diameters is due to tension reduction in the taut branch of the chain; (2) using a chain with shorter pitch yields a higher efficiency because of a reduction in energy loss in both the articulations between the links, and the contacts between rollers and teeth. Another important conclusion is that it is found that larger diameter sprockets yield higher efficiency values in every working condition, including those of high rotational speeds, in contrast to findings in previous literature.

Author Contributions

L. De Rossi contributed to plan the scheme and developed the model and the calculations; A. Schramm contributed to suggest the experiments and to analyze the results; A. De Felice contributed to plan the scheme and to analyze the results. The manuscript was written through the contribution of all authors. All authors discussed the results, reviewed, and approved the final version of the manuscript.

Acknowledgments

The authors kindly acknowledge M. Sgamma for his valuable contribution in the development of the multibody model.

Conflict of Interest

The authors declared no potential conflicts of interest with respect to the research, authorship, and publication of this article.

Funding

The authors received no financial support for the research, authorship, and publication of this article.

Data Availability Statement

All data generated or analyzed during this study are available from the corresponding author upon reasonable request.

Nomenclature

c_c	Contact damping coefficient (Hertz theory) [Ns/m]	n	Number of links
c_{cr}	Critical damping coefficient [Ns/m]	P	Power [W]
c_{eq}	Equivalent torsional damping coefficient [Nms/°]	RPS	Number of revolutions per second
c_{rz}	Torsional damping between two adjacent links [Nms/°]	RT	Radius of tooth profile [m]
c_{tx}	Longitudinal damping of a single link [Ns/m]	T_r	Damping torque [Nm]
E	Energy [J]	T	Period for a complete revolution of a link [s]
F_N	Normal force [N]	t	Time [s]
h	Exponent determining the nonlinear elastic reaction	X,Y	Spatial plane coordinates
K_{eq}	Equivalent stiffness of a series of links [N/m]	Z	Number of teeth
K_{tx}	Longitudinal stiffness of a single link [N/m]	α	Angle of tooth side
k_c	Contact stiffness (Hertz theory) [N/m]	δ	Penetration depth between two bodies [m]
M_t	Total mass of the chain [kg]	μ	Dry friction coefficient
m	Mass of a single link [kg]	τ	Speed ratio

References

[1] Bartlett, G.M., Roller chain drives in theory and practice, *Prod. Eng.*, 2(4), 1931, 253-255.
[2] Radzimovsky, E.I., Eliminating pulsations in chain drives, *Prod. Eng.*, 26(7), 1955, 153-157.
[3] Fuglede, N., *Kinematics and dynamics of roller chain drives*, Ph.D. Thesis, Mechanical Engineering, Technical University of Denmark, Kongens Lyngby, Denmark, 2014.
[4] Fuglede, N., Thomsen, J.J., Kinematics of roller chain drives. Exact and approximate analysis, *Mech. Mach. Theory*, 100, 2016, 17-32.
[5] Mahalingam, S., Transverse vibration of power transmission chains, *Br. J. Appl. Phys.*, 8, 1957, 145-148.
[6] Ariaratnam, S.T., Asokanthan, S.F., Dynamic stability of chain drives, *J. Mech. Transm-Trans. ASME*, 109(3), 1987, 412-418.
[7] Kim, M.S., Johnson, G.E., *A general multi-body dynamic model to predict the behaviour of roller chain drives at moderate and high speed*, Report, University of Michigan, Ann Arbor MI, USA, 1993.
[8] Pedersen, S.L., *Simulation and analysis of roller chain drive systems*, Ph.D. Thesis, Mechanical Engineering, Technical University of Denmark, Kongens Lyngby, Denmark, 2004.
[9] Zhao, J., Wang, S., Hu, S., et al., Dynamic analysis and simulation of a roller chain drive system on RecurDyn, *J. Appl. Sci. Eng. Innov.*, 1, 2014, 71-76.
[10] Archibald, F.R., Energy losses in chain-belt problem, *Mech. Eng.*, 68(2), 1946, 139-142.
[11] Fawcett, J.N., Nicol, S.W., The influence of lubrication on tooth-roller impacts in chain drives, *P. IMech. Eng.*, 191(21), 1977, 271-275.
[12] Conwell, J.C., Johnson, G.E., Experimental investigation of link tension and roller-sprocket impact in roller chain drives, *Mech. Mach. Theory*, 31, 1996, 533-544.
[13] Marshek, K.M., On the analyses of sprocket load distribution, *Mech. Mach. Theory*, 14, 1979, 135-139.
[14] Ryabov, G.K., The engagement of a worn chain with a sprocket, *Russ. Eng. J.*, 60(4), 1980, 31-34.
[15] Eldiwany, B.H., Marshek, K.M., Experimental load distributions for double pitch steel roller chains on steel sprockets, *Mech. Mach. Theory*, 19, 1984, 449-457.
[16] Naji, M.R., Marshek, K.M., Analysis of roller chain sprocket pressure angles, *Mech. Mach. Theory*, 19, 1984, 197-203.
[17] Naji, M.R., Marshek, K.M., The effect of the pitch difference on the load distribution of a roller chain drive, *Mech. Mach. Theory*, 24, 1989, 351-362.
[18] Troedsson, I., Vedmar, L., A method to determine the dynamic load distribution in a chain drive, *P. IMech. Eng. C-J. Mech.*, 215(5), 2001, 569-579.
[19] Lodge, C.J., Burgess, S.C., Experimental measurement of roller chain transmission efficiency, In: Su, D. (ed) *Proceedings of the International Conference on Gearing, Transmissions, and Mechanical Systems*, John Wiley and Sons, New York, 2000.
[20] Burgess, S.C., Lodge, C.J., Optimisation of the chain drive system on sports motorcycles, *Sports Engineering*, 7, 2004, 65-73.
[21] Lodge, C.J., Burgess, S.C., A model of the tension and transmission efficiency of a bush roller chain, *P. IMech. Eng. C-J. Mech.*, 216(4), 2001, 385-394.
[22] Wang, Y., Ji, D., Zhang, K., Modified sprocket tooth profile of roller chain drives, *Mech. Mach. Theory*, 70, 2013, 380-393.
[23] Wraggle-Morley, R., Yon, J., Lock, R., et al., A novel pendulum test for measuring roller chain efficiency, *Meas. Sci. Technol.*, 29, 2018, 075008.
[24] Zhang, S.P., Tak., T.O., Efficiency estimation of roller chain power transmission system, *Applied Science*, 10, 2020, 7729.
[25] Sgamma, M., Bucchi, F., Frendo, F., A phenomenological model of chain transmission efficiency, *IOP Conf. Ser. Mater. Sci. Eng.*, 1038, 2021, 012060.
[26] Pappalardo, C.A., Lettieri, A., Guida, D., A general multibody approch for the linear and nonlinear stability analysis of bycicle systems. Part I: methods of constrained dynamics, *J. Appl. Comput. Mech.*, 7(2), 2021, 655-670.
[27] Pappalardo, C.A., Lettieri, A., Guida, D., A general multibody approch for the linear and nonlinear stability analysis of bycicle systems. Part II: application to the Whipple-Carvallo bycicle model, *J. Appl. Comput. Mech.*, 7(2), 2021, 671-700.
[28] Manrique-Escobar, C.A., Pappalardo, C.A., Guida, D., On the analytical and computational methodologies for modelling two-wheeled vehicles within the multibody dynamics framework: a systematic literature review, *J. Appl. Comput. Mech.*, 8(1), 2022, 153-181.
[29] Romualdi, L., Mancinelli, N., De Felice, A., Sorrentino, S., A new application of the Extended Kalman filter to the estimation of roll angles of a motorcycle with Inertial Measurement Unit, *FME Transactions*, 48(2), 2020, 255-265.
[30] *Precision Power Transmission Roller Chains, Attachments, and Sprockets*, ASME B29.1, 2011 (revised 2016).
[31] *Short-pitch transmission precision roller and bush chains, attachments and associated chain sprockets*, ISO 606, 4th edition, 2015.
[32] Popov, V.L., Willert, E., Hess, Markus, Method of dimensionality reduction in contact mechanics and friction: a user's handbook. III. Viscoelastic contacts, *Facta Universitatis, Series: Mechanical Engineering*, 16(2), 2018, 99-113.
[33] Dubowsky, S., Freudenstein, F., Dynamic analysis of mechanical systems with clearances, part 1: Formation of dynamic model, *J. Eng. Ind-Trans. ASME*, 1971, 305-309.
[34] Dubowsky, S., Freudenstein, F., Dynamic analysis of mechanical systems with clearances, part 2: Dynamic response, *J. Eng. Ind-Trans. ASME*, 1971, 310-316.
[35] Pedersen, S., Hansen, J.M., Ambrosio, J.A.C., A roller chain drive model including contact with guide-bars, *Mul. Syst. Dyn.*, 12, 2004, 285-301.
[36] Sorrentino, S., Leonelli, L., A study on the stability of a motorcycle wheel-swingarm suspension with chain transmission, *Veh. Syst. Dyn.*, 55(11), 2017, 1707-1730.
[37] Cattabriga, S., De Felice, A., Sorrentino, S., Patter instability of racing motorcycles in straight braking manoeuvre, *Veh. Syst. Dyn.*, 59(1), 2021, 33-55.

ORCID iD

Lorenzo De Rossi https://orcid.org/0000-0001-8226-7160
Alexander Schramm https://orcid.org/0000-0001-6885-9848
Alessandro De Felice https://orcid.org/0000-0001-9142-1334

How to cite this article: De Rossi L., Schramm A., De Felice A. Multibody Efficiency Analysis of Chain Drives in Racing Motorcycles, *J. Appl. Comput. Mech.*, 8(3), 2022, 1091–1102. https://doi.org/10.22055/jacm.2022.39795.3467