A Camshaft Non-Linear Model for the Desmodromic Valve Train Simulation

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Abstract
In this paper the dynamic behaviour of the desmodromic valve train of a motorbike engine is simulated and analysed. The valve train model takes into account the mass distribution, the link elastic flexibility, and the presence of several non-linearities. The camshafts, which are supported by hydrodynamic journal bearings, are modelled by means of finite rotating elements based on Timoshenko beams with the effect of gyroscopic moment. The non-linear bearing forces are analytically obtained under the finite-length bearing assumption. The comparison between the numerical results and the experimental data – in terms of valve acceleration and camshaft angular velocity – shows that the effectiveness of the model is satisfactorily assessed. The paper mainly focuses on two points: the validation procedure of the finite element model of the camshafts, and the study of the simulated camshaft centreline trajectories.

1 Introduction

This paper deals with a kineto-elastodynamic model of the desmodromic valve train of a Ducati motorbike engine. The desmodromic train is a mechanism with positive-drive cams and – in comparison with the widely-used trains having a closing spring [1–3] – presents different dynamic behaviour, as shown in [4–7].

When operating at high-speed, a mechanism shows a dynamic behaviour which is affected by the link elastic flexibility and mass distribution, as well as the effects of backlashes and friction in joints. The resulting motion may be affected so deeply that the mechanism could fail to properly perform its task. In addition, high accelerations and dynamic stress may occur, causing early fatigue failure, and high vibration and noise. The development of an elastodynamic model of the mechanism allows the estimation of the actual dynamic forces, impacts, and mechanism performances, as well as the design optimisation and fault diagnostics. As a matter of fact, nowadays increasing attention is addressed to the elastodynamic simulation of high-speed mechanisms [8–11]. In the specific field of the engine valve trains, the above-mentioned dynamic effects are particularly important, since they may cause serious functional troubles, such as wear, fatigue loads and breakage of mechanical components, jump and bounce phenomena of the valve, and alteration of the engine’s fluid dynamics [4, 5, 12].

Experimental tests were carried out at the DIEM Laboratory of the University of Bologna on cooperation with Ducati, in order to get insight into the dynamics of the desmodromic cam system and help the development of the elastodynamic model. In particular, the valve motion and the camshaft angular velocity were picked up during the tests [5].

The model presented here simulates the desmodromic valve train and the mechanical transmission of the test bench; it is therefore possible to employ the experimental results for the model assessment and validation. The elastodynamic model takes into account the mass distribution and the flexibility of all links and the presence of several non-linearities. In particular, the camshafts, which are supported by
The organisation of the paper is as follows. Section 2 deals with the mechanical system and the measurement apparatus. Section 3 is devoted to the model description and validation: firstly is given a general description of the model; secondly the finite element model of the camshaft is shown; finally the validation procedure of the camshaft model is reported. In Section 4 the numerical results are compared with the experimental data in terms of valve acceleration and camshaft angular velocity. In addition, the simulated camshaft centreline trajectories are examined and the effects due to the cam mechanism forces are analysed. Section 5 is devoted to concluding remarks.

2 The valve train under test and the experimental apparatus

2.1 The tested desmodromic train

The tested system is a portion of the valve train of the fourcylinder ‘L’ engine of Ducati racing motorbike which has double overhead camshafts, desmodromic valve trains and four valves per cylinder. In the motorbike engine there are four camshafts having each four conjugate cams: two camshafts drive the intake valves while the exhaust valves are moved by the other two. The camshafts are supported by hydrodynamic journal bearings.

Experimental tests were carried out on a test bench developed at the DIEM Laboratory of the University of Bologna on cooperation with Ducati. Only one cylinder head is present on the test bench, and the two camshafts have each only one conjugate cam. In other words, the test bench makes it possible to run only two valves: one exhaust valve and one intake valve. In particular, the experimental results reported in this work are relative to tests conducted with the presence of the intake valve only. The system under test is clearly different from the real one: in particular, it is only a portion of the real valve train and it is worth noting that only the components required for operating the valve train are included into the test bench. The system response is therefore dissimilar from the actual one (i.e. the response of the motorbike engine system). However, such differences, as well as the inclusion (or exclusion) of the forces due to compressed gases, do not compromise the validity of the experimental data as a tool for model validation [12].

On the test bench the camshafts are moved by means of an electrically powered driveline schematically shown in Fig. 1(a): it consists of a brushless motor that drives an intermediate shaft with ratio \( \tau_1 = 10/3 \) by means of the timing belt \( B_1 \). The intermediate shaft is fitted out with a flywheel in order to reduce fluctuations of torque and velocity; in the motorbike engine, it corresponds to the shaft linked with the crankshaft with a gear transmission having ratio 1/2. A second timing belt, denoted as \( B_2 \), drives the two camshafts [only one camshaft is shown in Fig. 1(a)] with ratio \( \tau_2 = 1/1 \): it exactly reproduces the engine belt loop of a real motorbike.

Figure 1(b) shows the schematic of the cam mechanism driving a single valve: the discs of a conjugate cam are each in contact with a rocker and the two rockers are in contact with the backlash adjuster. It is therefore possible to identify two parts of the mechanism: one part, made up of one cam disc and the related rocker, gives valve positive acceleration, while the negative acceleration is given to the valve by the other cam disc with the associated rocker (the positive direction is that of the valve opening). The action of the small rocker spring is mainly needed during the dwell phase, when rockers could separate from the cam discs and, consequently, the contact between the valve-head and the seat might be lost.
Figure 1: Schematic of the mechanical system: a) electrically powered driveline; b) cam mechanism driving a single valve.

2.2 The experimental apparatus

The test bench and the measurement apparatus can operate for different cylinder head type, at high camshaft speed, under high temperature of the lubrication oil, and reproducing the motorbike power belt transmission. In particular, the experimental apparatus includes a test stand, a cylinder head, an electrically powered driveline to operate the camshafts, a lubrication circuit, and the measurement instrumentation [5]. The maximum speed available at the camshaft is 10000 rpm. In order to properly lubricate the valve train, pressurized oil is fed into the cylinder head oil galleries; oil pressure and temperature are similar to those picked up from the motorbike during the racing.

The measurement equipment consists of a laser vibrometer and data acquisition apparatus. The laser equipment is a Polytec’s High Speed Vibrometer (HSV-2002), which can measure the absolute and relative velocity and displacement up to 30 m/s and 41 mm respectively. The centre of the valve-head plane surface was chosen as measurement point, thus making it possible to minimize possible valve’s flexional vibration effects, which may negatively affect valve motion measurement. An area close to the valve seat was selected as a reference surface. The differential measurement between valve surface and reference plane permits the elimination of raw vibration effects of the head cylinder support.

In order to refer time dependent valve motion measurements to the cam angular position, the intake camshaft was fitted out with an incremental encoder having 360 notches. This device makes available digital signals that allow the measurement of the camshaft angular velocity, thus providing insight into the dynamics of the timing belt transmission.

The signals were collected by means of a National Instrument PXI data acquisition system; the sampling frequency of the analogue signals was 102 kHz, while the digital signals were handled by 80 MHz counters. During the tests, velocity and displacement of the valve, and encoder signals were recorded. The signals were then processed and analysed with MATLAB.
3 Model description, validation and solution

3.1 General description of the model

The mechanical system described in Section 2.1 is modelled taking into account the presence of the electrically powered driveline, the two camshafts, and the cam mechanisms driving the two valves. Since the exhaust valve was not present during the experimental tests, in the course of model validation the cam mechanism driving that valve is turned off.

Seen as a whole, the model includes the lumped-parameter model of the power transmission, the finite element model of the camshafts, and the lumped-parameter model of the cam-valve mechanisms. The camshafts are modelled by means of finite rotating elements based on Timoshenko beams with the effect of gyroscopic moment. The hydrodynamic journal bearings, which support the camshafts, are introduced by means of a finite-length analytical model. The lumped-parameter portion of the model is developed with the aim to include all the important dynamic effects. In particular, it takes into account the mass distribution, the elastic flexibility of the links, the variation of rocker stiffness as a function of mechanism position, the damping effects, the variability of transmission ratios with mechanism position, and the presence of several non-linearities (e.g. the Hertzian contact stiffness, the backlashes in joints, and the lubricant squeeze effect).

Figure 2: Schematic of the model.
With reference to Fig. 2, the known model input is the coordinate $\varphi_0$, representing the angular displacement of the brushless motor pulley, which is assumed as rotating at constant speed. The torsional stiffness $k_{b1}$ represents the stiffness of the first belt transmissions [i.e. the timing belt B1 of Fig. 1(a)]. The intermediate shaft is modelled by the moment of inertia $J_{s_j}$, ($j=1, 2, 3$) and the torsional stiffness $k_{s12}$ and $k_{s23}$. The symbol $k_{b2,i}$ ($j=1, 2, 3$) denotes the stiffness of the three branches of the belt transmissions B2 which connects the intermediate shaft and the two camshafts. The coordinate $\theta_{I(E)}$, represents the angular position of the pulley fitted to the intake (exhaust) camshaft. For the sake of simplicity, Fig. 2 schematically represents the camshaft as a rectangle having two cam profiles that are linked with the mechanism driving a single valve. In particular, this mechanism consists of the positive and negative rocker, the adjuster, and the valve. The coordinates of the mechanism, and the related model parameters, are reduced to the direction of the valve motion. Therefore, the linear coordinates $x_3$ and $x_6$, associated to the rockers, correspond to the rocker angular displacements [denoted $\theta_3$ and $\theta_6$ in Fig. 1(b)]. The coordinates $x_4$ and $x_5$ are associated to the adjuster and valve-head, respectively.

As a result, the lumped-parameter section of the model consists of eleven dofs: three of them are related to the power belt transmission, while the remaining eight dofs are devoted to modelling the two cam-valve mechanisms.

The rocker stiffness, represented by the parameters $k_{13}$, $k_{34}$, $k_{46}$, and $k_{26}$, are obtained by reducing and composing in series the stiffness of the corresponding rocker arm and the stiffness of the related Hertzian contact. It is noteworthy that the stiffness of the rockers is assumed as a function of the mechanism position [6]. Moreover, also the Hertzian stiffness is variable, as it depends on the contact force; in the simulation it is evaluated instantaneously. The possibility of separation of the rockers from the cam discs and the adjuster is taken into account by means of the parameters $\delta_{13}$, $\delta_{34}$, $\delta_{46}$, and $\delta_{6}$. A viscous damper is associated with each stiffness, in order to globally take account of structural damping, as well as other damping. In case there is no contact in joints with backlash, and links are approaching, the damper coefficient is computed in order to represent the lubricant squeeze effect [4, 9]. Coulomb friction forces have not been introduced into the model, due to their low value in comparison with the high dynamic loads present in the system. More details on the lumped-parameter portion of the model can be found in [4, 6].

### 3.2 Model of the camshaft supported by hydrodynamic journal bearings

The camshaft can be thought as a flexible rotor-bearing system that consists of a rotor composed of discrete discs (e.g. the cams), rotor segments with distributed mass and elasticity, and discrete bearings. Such a system is shown in Fig. 3(a) along with the fixed cartesian reference frame $XYZ$. The $XYZ$ triad has the $Z$ axis coincident with the undeformed shaft centreline, while the $Y$ axis is parallel to the direction of the valve motion. The location and orientation of a cross section of the rotor in a deformed state are defined by means of the translations $\nu_x$ and $\nu_y$ in the $X$ and $Y$ directions respectively, and by the rotations $\varphi_x$, $\varphi_y$, and $\theta$ about the $X$, $Y$, and $Z$ axes, respectively [see Fig. 3(b)].

A finite element model of the camshaft can be obtained by properly assembling mass, damping, stiffness and gyroscopic matrices of the finite elements that constitute the rotor system. In this work, the flexional and torsional equations of motion are formulated separately, while the axial contribution is omitted due to the absence of axial forces acting on the cam shaft. In particular, the equations of motion of a finite element are formulated as follows:

$$
[M]_f \{\ddot{u}\}_x + [C]_f \{\dot{u}\}_x + [G]_f \{\dot{u}\}_y + [K]_f \{u\}_x = \{F(t)\}_x
$$

$$
[M]_f \{\ddot{u}\}_y + [C]_f \{\dot{u}\}_y - \bar{G}[G]_f \{\dot{u}\}_x + [K]_f \{u\}_y = \{F(t)\}_y
$$

$$
[M]_\theta \{\ddot{\theta}\}_\theta + [C]_\theta \{\dot{\theta}\}_\theta + [K]_\theta \{\theta\}_\theta = \{F(t)\}_\theta
$$

(1) (2) (3)
where \([M]_{\theta}, [C]_{\theta}, \) and \([K]_{\theta}\) respectively represent the flexional (torsional) mass, damping, and stiffness matrices of the finite element; \([G]\) takes into account the element gyroscopic effects, \(\{F(t)\}_{xz(2)}\) is the flexional forcing term that acts on \(X-Z (Y-Z)\) plane, while \(\{F(t)\}_\theta\) is the torsional forcing term.

![Schematic of the camshaft.](image)

By observing the Eqns. (1, 2), it can be noted that the flexural matrices are the same both for plane \(X-Z\) and \(Y-Z\), that is, the camshaft is assumed to be axisymmetric. In other words, the asymmetry due to the presence of the cams is neglected.

The coordinate vectors appearing in the Eqns. (1–3) represent the coordinates of the nodes of the \(j\)-th finite element:

\[
\{u_z(t)\}_j^T = [v_{x}^j(t) \quad \varphi_x^j(t) \quad v_{y}^{j+1}(t) \quad \varphi_y^{j+1}(t)] \tag{4}
\]

\[
\{u_z(t)\}_j^T = [v_{y}^j(t) \quad \varphi_y^j(t) \quad v_{x}^{j+1}(t) \quad \varphi_x^{j+1}(t)] \tag{5}
\]

\[
\{\theta(t)\}_j^T = [\theta^j(t) \quad \theta^{j+1}(t)] \tag{6}
\]

The flexional matrices \([M]_\varphi, [K]_\varphi, \) and \([G]\) are obtained according to [13]. In particular, a Timoshenko finite element was employed, thereby including transverse shear effects. The torsional matrices \([M]_{\theta}\) and \([K]_{\theta}\) are derived by assuming a linear shape function for the torsional finite element [14].

The presence of the damping has been introduced under the hypothesis of proportional damping, that is, the damping matrix is a linear combination of mass and stiffness matrices by means of coefficients \(\alpha\) and \(\beta\): \([C]_{\theta} = \alpha [M]_{\theta} + \beta [K]_{\theta}\) [15]. The estimation of the coefficients \(\alpha\) and \(\beta\) will be discussed in the Section 3.3.

The vectors \(\{F\}_{xz}, \{F\}_{yz}, \) and \(\{F\}_\theta\) in Eqns. (1–3) contain the forces, bending moments, and torques acting on the finite element nodes. In particular, on the nodes corresponding to the cams act the external forces transmitted by the lumped-parameter model of the cam-valve mechanism, while the external forces and torques transmitted to the camshaft by the lumped-parameter model of the power belt transmission take action on the node related to the camshaft pulley. In addition, the model takes into account the forces due to the cam static unbalance: they act on the nodes corresponding to the cams. Finally, the fluid dynamic forces operate on the nodes that correspond to the camshaft supports.

The camshaft supports consist of two plain circumferentially-symmetric fluid journal bearings. A non-linear description of the bearing forces is required for transient rotor dynamic analysis. Specifically, the description defines the bearing reaction as a function of the position and velocity of the rotor at the bearing location. In this work the bearing forces are computed on the basis of a finite-length bearing analytical model [16]. With reference to Fig. 3, each support is divided into two identical parts; the forces acting on the bearing node are expressed as:

\[
F_X = F_X(v_x, v_y, \dot{v}_x, \dot{v}_y, \dot{\theta}, D, L, \delta, \mu), \quad F_Y = F_Y(v_x, v_y, \dot{v}_x, \dot{v}_y, \dot{\theta}, D, L, \delta, \mu) \tag{7}
\]
where $\delta$ is the radial clearance, $\mu$ is the fluid viscosity, $D$ and $L$ are the bearing diameter and length, respectively.

### 3.3 Camshaft model validation

Details on the validation of the lumped-parameter portions of the model are given in [4, 6]. The aim of this Section is to describe the procedure used to validate the camshaft finite element model, that is, to show how the dynamic matrices $[M]_{\theta\theta}$ and $[K]_{\theta\theta}$ were validated and the coefficients $\alpha$ and $\beta$ of the proportional damping matrix $[C]_{\theta\theta}$ were estimated.

The camshaft shape is quite simple; in fact, except the cams, it consists of coaxial cylinders with circular cross section which have different diameters. Consequently, once the dimensions, density and structural properties of the material are known, the torsional matrices are almost confident. On the contrary, uncertainty involves the flexural matrices especially due to the low $L/D$ (length to diameter) ratio of the camshaft sections. Therefore, the validation procedure only deals with the flexural matrices $[M]_F$ and $[K]_F$, while the torsional ones (i.e. $[M]_{\theta\theta}$ and $[K]_{\theta\theta}$) are assumed to be confident.

In order to estimate the coefficients $\alpha$ and $\beta$, a simple modal test was carried out on the intake camshaft (the only one available): without the pulley, it was softly suspended to the ground and impacted by a hammer. In particular, see Fig. 4, the Frequency Response Function (FRF) was estimate by impacting the camshaft at the point A in the Y direction, and measuring the acceleration response by a piezoelectric transducer at the same point and in the same direction: it is denoted as $H_{A-A}$ hereafter. The impact excitation was also applied at other points, but the resulting FRFs presented a very low coherence function and were therefore unusable. The amplitude of the FRF $H_{A-A}$ is shown in Fig. 5(a): two pairs of resonant peaks clearly appear at about 4 and 13 kHz. The flexural natural frequencies result as uncoupled, because the camshaft is not axisymmetric due to the presence of the cams. Between 4 and 13 kHz other modes are possibly present, but they appear as over-damped and their contribution seems negligible.

![Figure 4: Set-up of the modal test on the intake camshaft.](image)

![Figure 5: Amplitude of the FRF $H_{A-A}$: a) experimental; b) numerical.](image)

The proportional damping coefficients $\alpha$ and $\beta$ were estimated by means of spectral modelling [15]. For the vibrational mode $j$, under the assumption of proportional damping, the following expression holds:

$$2 \omega_j \omega_{\sigma_j} = \frac{C_j}{M_j} = \frac{\alpha M_j + \beta K_j}{M_j} = \alpha + \beta \omega_{\sigma_j}^2$$

(8)
where $\xi_j$ and $\omega_{nj}$ are the damping ratio and the natural frequency of the mode $j$, while $M_j$, $K_j$, $C_j$ are respectively its modal mass, modal stiffness and modal damping. If $\xi_j$ and $\omega_{nj}$ are known for $N$ modes ($j=1, \ldots, N$), the equation (9) directly provides the coefficients $\alpha$ and $\beta$:

$$\{\lambda\} = \left(\begin{bmatrix} A^T A \end{bmatrix}^{-1} [A]^T \{b\}\right)$$

with: $\{\lambda\} = \begin{bmatrix} \alpha \\ \beta \end{bmatrix}$, $[A] = \begin{bmatrix} 1 & \omega_{n_1}^2 \\ \vdots & \vdots \\ 1 & \omega_{n_N}^2 \end{bmatrix}$, $\{b\} = \begin{bmatrix} \xi_1 \\ \xi_2 \\ \vdots \\ \xi_N \end{bmatrix}$.

By taking into account only the modes around 4 and 13 kHz, the modal parameters $\xi_j$ and $\omega_{nj}$ were estimated based on the FRF $H_{A-A}$ by means of the software MTS Engineering Office. The results of the modal parameter estimation are reported in Table 1.

<table>
<thead>
<tr>
<th>natural frequency [Hz]</th>
<th>damping ratio [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>3992.07</td>
<td>0.1349</td>
</tr>
<tr>
<td>4216.51</td>
<td>0.0741</td>
</tr>
<tr>
<td>13001.19</td>
<td>0.1182</td>
</tr>
<tr>
<td>13343.29</td>
<td>0.3618</td>
</tr>
</tbody>
</table>

Table 1: Results of the modal parameter estimation.

Based on the modal parameter of Table 1, Equation (9) provides: $\alpha = 9.46 \, \text{s}^{-1}$, and $\beta = 5.78 \times 10^{-8} \, \text{s}$. These values were also applied in the case of exhaust camshaft. In addition, although they were retrieved from an experiment related to the flexional vibrations, they were assumed to be valid for the proportional damping matrix of torsional vibrations too.

<table>
<thead>
<tr>
<th>non-validated model [Hz]</th>
<th>validated model [Hz]</th>
</tr>
</thead>
<tbody>
<tr>
<td>4209.8</td>
<td>3986.5</td>
</tr>
<tr>
<td>10016.8</td>
<td>9094.2</td>
</tr>
<tr>
<td>15392.2</td>
<td>13160.8</td>
</tr>
</tbody>
</table>

Table 2: Flexional natural frequencies of the finite element model of the intake camshaft.

With reference to the flexural matrices, the first column of Table 2 lists the flexional natural frequencies of the intake camshaft computed by means of $[M]_F$ and $[K]_F$ before the validation (only values below 20 kHz are reported). Since the flexural matrices are assumed to be the same in the planes $X-Z$ and $Y-Z$, the resonant frequencies have the same value in those two planes. Before the validation, the first resonance almost agrees with the experimental value, while the others do not. In order to arrange the frequencies, some modifications were made on the flexural matrices, especially on the mass matrix $[M]_F$. In particular, the mass moment of inertia of the cam discs, which have been considered as cylinders with circular cross section, were adjusted. The natural frequencies of the validated model are reported in the second column of Table 2. After, the model updating, the third natural frequency moved to about 13 kHz. Figure 5(b) shows the numerical FRF obtained by the validated flexional matrices $[M]_F$ and $[K]_F$, and the estimated proportional damping coefficients $\alpha$ and $\beta$. The comparison between Fig. 5(a) and Fig. 5(b) shows that both amplitude and frequency of the more important flexural modes are good enough.
As mentioned above, the torsional matrices are assumed to be confident. In order to prove their goodness, Figs. 6(a, c) show the comparison between the experimental and numerical angular velocity of the intake camshaft pulley. The experimental angular velocity is achieved by processing the digital signal from the encoder fitted to the camshaft pulley, while the numerical angular velocity is obtained from the node correspondent to the pulley. The signals refer to the camshaft speed of 6000 rpm and are reported over one camshaft revolution. The comparison shows that the model is able to predict the camshaft torsional motion very well. In particular, the low frequency oscillations (below the 10th camshaft order) are due to the dynamics of the belt transmission. Therefore, such a good simulation supplies further and interesting information on the belt dynamics.

Apart from that, the torsional modes of the camshaft are those related to the high frequency oscillations. Figures 6(b, d), show the results of the spectral analysis performed on the signals of Figs. 6(a, c). In particular, both the experimental and numerical spectrum reveal a high peak at about 32 camshaft rotation orders with a 30 dB amplitude. Since the camshaft speed is 6000 rpm (i.e. the rotational frequency is 100 Hz), the order 32 corresponds just to the first torsional frequency of the intake camshaft (3248.3 Hz) computed by means of the torsional matrices $[M]_\theta$ and $[K]_\theta$, (see Table 3 where the torsional natural frequencies below 20 kHz are reported for both the camshafts). In addition, in the numerical spectrum appear other two resonant bands around 110 and 160 camshaft rotation orders, which correspond to the second and third torsional natural frequencies of the intake camshaft, respectively. These resonant bands are not so evident in the spectrum of the experimental signal that, on the other hand, evidences how the main contribution to the camshaft angular motion is given by its first mode. As a conclusion, it is proved that the torsional matrices of the finite element model of the camshaft are confident, especially for the first vibrational mode. It can be noted from Table 3 that the exhaust camshaft has torsional frequencies higher than those of the intake one; in fact, the camshafts are quite similar, but the exhaust shaft is slightly shorter than the intake one and, above all, is not equipped with the encoder.

![Figure 6](image1.png)

Figure 6: Angular velocity of the intake camshaft pulley: (a, b) experimental; (c, d) numerical.

<table>
<thead>
<tr>
<th>Intake [Hz]</th>
<th>Exhaust [Hz]</th>
</tr>
</thead>
<tbody>
<tr>
<td>3248.3</td>
<td>3796.2</td>
</tr>
<tr>
<td>10825.2</td>
<td>12893.1</td>
</tr>
<tr>
<td>16203.6</td>
<td>17634.2</td>
</tr>
</tbody>
</table>

Table 3: Torsional natural frequencies of the camshaft finite element models.
3.4 Model solution

As mentioned above, the lumped-parameter portion of the model is composed of 11 dofs. Each camshaft consists of 36 finite elements which correspond to 37 nodes having each 5 dofs; as a consequence, the finite element portion of the model comprises 370 dofs. Seen as a whole, the model is therefore made of 381 dofs.

It is worth noting that the mechanical system is highly time-varying. As a matter of fact the active cam-follower system is different in the phases of positive and negative acceleration, the bearing forces are taken into account by a non-linear formulation, and several model parameters change with mechanism position. As a consequence, the differential equations of motion are strongly non-linear. They are numerically integrated by using the software Simulink.

Due to the high number of dofs, the computation cost might be heavy; the ‘pseudo-modal’ reduction method was therefore applied [17] to the finite element model of the camshafts. In particular, only the camshaft mode shapes corresponding to the natural frequencies lower than 20 kHz were taken into account. As a result, in the case of the intake camshaft 16 modes were considered (including the rigid modes): 12 flexional and 4 torsional. Conversely, for the exhaust camshaft only 14 modes were taken into account (including the rigid modes): 10 flexional and 4 torsional. In conclusion, after the reduction process, the model globally consists of 41 dofs (11 of them concern the lumped-parameter portion of the model).

The simulation of the valve train motion over one camshaft revolution takes about 50 s of CPU time on a Pentium III with 1 GHz clock speed.

4 Results and discussion

The results presented in this Section refer to the camshaft speed of 6000 rpm and are reported over one camshaft revolution.

The experimental acceleration of the intake valve is obtained by means of numerical derivative from the laser measurement of valve velocity, while the simulated valve acceleration is represented by the acceleration of the mass $m_5$ of the model of the cam-valve mechanism (see Fig. 2). The acceleration scale is made dimensionless with reference to the theoretical maximum value.

4.1 Simulation of the tested desmodromic train

Since the exhaust valve was not present during the experimental test, the simulation of the tested desmodromic train only deals with the intake cam-valve mechanism. In the course of model validation the cam mechanism driving the exhaust valve was therefore turned off.

Figure 7 compares the experimental [Fig. 7(a)] with the numerical [Fig. 7(b)] intake valve acceleration. Such a comparison shows good agreement between experimental data and the numerical results: thus, the effectiveness of the model can be satisfactorily assessed. The simulated acceleration is very similar to the actual one, inside of both the positive and negative acceleration phases, even if some discrepancies exist. In particular, the model overestimates to some extent the oscillation amplitude of the first acceleration phase, and, in the closing phase, shows a peak at about 200 degrees which is not present in the experimental acceleration. However, the model is able to reproduce the more important dynamic phenomena, and the level of the acceleration peaks is globally matched.

As a further evidence of the model accuracy, the actual and numerical angular velocity of the intake camshaft have been already compared in Figs. 6(a, c) (see Section 3.3). That comparison showed that the model is able to predict the intake camshaft motion very well. In particular, both the amplitude and frequency of the camshaft oscillation are correctly reproduced. Here, see Fig. 8, the simulated angular velocity of the exhaust camshaft is also reported. Since the exhaust valve is not active, the dynamic
behaviour of the exhaust camshaft results more regular: only low frequency oscillations take place, as demonstrated by the spectrum of Fig. 8(b). The camshaft torsional motion is mainly governed by the dynamic of the belt transmission. The different peak from those below the 10th order is the first torsional natural frequency of the camshaft, at about 38 rotation orders; in fact, as can be seen by Table 3, the exhaust shaft has its first torsional resonance at 3796.2 Hz.

Figure 7: Intake valve acceleration (normalised): a) experimental; b) numerical.

Figure 8: Simulated angular velocity of the exhaust camshaft (a) and its spectrum (b).

Figure 9: Orbit reference frames and resultants of the average belt loads.

Once the model is validated and its effectiveness is satisfactorily assessed, it can be employed to investigate and interpret the dynamics of the mechanical system. In particular, the camshaft axis orbits, which are not available from experimental measurements, are analysed hereafter. Figure 10 shows the polar plots of the camshaft axis at the pulley, bearing, and cam locations (for the notation also see Fig. 3). The angles 0 and $\pi/2$ of the polar coordinate system correspond to the rectangular coordinates $\xi$ and $\eta$ of Fig. 9, respectively. In addition, Fig. 9 shows the resultants of the average belt loads acting on the camshaft pulley axes, thus helping the interpretation of the whirl orbits.

Figures 10 (a–g) show the simulated orbits of the exhaust camshaft. The centreline of this shaft approximately moves on a conic surface with the vertex close to the bearing 1-2 at about 345 degree angle
[see Fig. 10(c)], and the axis which is not parallel to the Z axis. This can be clearly stated by observing Fig. 11(a), as well as that the orbit centres of the pulley and bearing 2-2 are not in the same quadrant [see Figs. 10(a, e)]. Such a quite regular behaviour is due to the absence of the cam-valve mechanism. The main forces that take action on the exhaust camshaft are the belt load resultant and the rotating forces due to the cam unbalance. As an additional remark, it is noteworthy that the orbit centres of the bearings 1-1 and 1-2 are at about 345 degree angle. On the other hand, the direction of the resultant of average belt loads is on the same quadrant at about 315 degree angle. As a consequence, it can be concluded that the bearings close to the pulley have a 30 degrees attitude angle. Moreover, the minimum radial gap is reached into the bearing 2-2 and, as the bearing radial clearance is $3.72 \times 10^{-2}$ mm, its value is about 8 $\mu$m [see Fig. 10(e)].

![Simulated camshaft axis orbits](image)

Figure 10: Simulated camshaft axis orbits: (a) exhaust pulley; (b–e) exhaust camshaft bearings; (f, g) exhaust cams; (h) intake pulley; (i–l) intake camshaft bearings; (m, n) intake cams.
Figures 10 (h–n) report the simulated orbits of the intake camshaft. In particular, the Figs. 10(m) and 10(n) show the camshaft axis orbits at the cam positions. Since the two cam discs are close together, their orbits are similar. With reference to the negative cam, its axis orbit exhibits two phases: an ellipse portion between 170 and 310 degree angles, which corresponds to the dwell phase of the valve motion, and a more irregular pattern between 30 and 130 degree angles. As a matter of fact, during the valve dwell phase, the most important forces acting on the camshaft are the belt load resultant, which has approximately constant direction and magnitude, and the rotating forces due to the static unbalance of the cams. Conversely, when the valve leaves its seat the dynamics of the camshaft is mainly influenced by the high contact forces between the cams and the rockers. The complex whirling behaviour of the intake camshaft centreline is reported in Fig 11(b). Obviously, the cam axis orbits and the camshaft centreline trajectories at the location of the bearings 2-1 and 2-2 [Figs. 10(k) and 10(l)] – i.e. the bearings close to the cams – have a similar pattern. The pulley axis, as well as the axes of the bearings close to the pulley (namely the bearings 1-1 and 1-2) exhibit a trajectory that approximately covers the third quadrant of the polar coordinate system [see Figs. 10(h, i, and j)]; this agrees with the direction of the resultant of average belt loads (see Fig. 9).

Figure 11: Simulated motion of the camshaft centreline (the blue line represents the undeformed shaft centreline, while the red lines represent the orbit of the cams): a) exhaust camshaft; b) intake camshaft.

4.2 Effect of the exhaust cam-valve mechanism.

This Section aims to show the dynamic behaviour of the desmodromic train when the exhaust cam-valve mechanism is activated.

The motions of the intake and exhaust valve differ of some amount in the theoretical pattern and present a 90 degrees shift. The simulated acceleration of both exhaust and intake valve is reported in Figs. 12(a, b); the exhaust valve exhibits higher negative acceleration peaks. The dynamic of the intake valve seems to be hardly influenced by the presence of the exhaust valve, as demonstrated by the comparison between Fig. 12(b) and Fig. 7(b). On the contrary, the behaviour of the belt transmission shows evident modifications. In fact, the inertia forces which take part in the exhaust cam-valve mechanism, alter the pattern of torsional motion of the camshafts, as can be seen by examining their angular velocity before the exhaust valve is activated [compare Fig. 12(c) to Fig. 8(a) and Fig. 12(d) to Fig. 6(c)]. In addition, Fig. 12(c) and the related spectrum [Fig. 12(e)] show that the forces acting on the exhaust camshaft are able to excite its higher torsional resonances. Figure 13 is a further proof of the changes in the motion of the exhaust camshaft originated by the activation of the exhaust cam-valve mechanism: the orbits of the exhaust cam axis are completely different from those depicted in Figs. 10(f, g). Conversely, the intake cam orbits are scarcely influenced by the presence of the exhaust mechanism [compare Figs. 13(c, d) to Figs. 10(m, n)].
5 Conclusions

In this paper the dynamic behaviour of the desmodromic valve train of a motorbike engine is simulated and analysed. In order to validate the model, experimental tests were carried out on a test bench at the DIEM Laboratory of the University of Bologna, on cooperation with Ducati.

Seen as a whole, the model includes the lumped-parameter model of the power belt transmission, the finite element model of the camshafts, and the lumped-parameter model of the cam-valve mechanisms. The camshafts are supported by hydrodynamic journal bearings; the non-linear bearing forces are analytically obtained under the finite-length bearing assumption. The paper mainly deals with two points: the validation procedure of the finite element model of the camshafts, and the study of the simulated camshaft centreline trajectories. In particular, the effects on the camshaft motion due to the cam mechanism forces
are examined and discussed.

The comparison between the numerical results and the experimental data – in terms of valve acceleration and camshaft angular velocity – show that the effectiveness of the model is satisfactorily assessed. The results show that the model can be a very useful tool in order to predict and understand the actual dynamic behaviour of the desmodromic valve train system.

References


