A NON LINEAR ELASTODYNAMIC MODEL OF A CAMSHAFT SUPPORTED BY JOURNAL BEARINGS

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ABSTRACT
This paper deals with a non-linear model of a camshaft supported by hydrodynamic journal bearings, in order to analyse the elastodynamic behaviour of the desmodromic valve train of a motorbike engine. The model of the valve train takes into account the mass distribution, the link elastic flexibility, and the presence of several non-linearities. The camshaft is modelled as Timoshenko beams with effect of gyroscopic moment. The non-linear bearing forces are analytically obtained under the finite-length bearing assumption. The numerical results are compared with the experimental data in terms of valve acceleration and camshaft angular velocity. By means of this comparison, the effectiveness of the model is satisfactorily assessed. Moreover, the simulated camshaft centreline trajectories are examined and the effects due to the cam mechanism forces are analysed.

SOMMARIO
In questo lavoro viene presentato un modello non lineare di un albero a camme supportato da cuscinetti idrodinamici, al fine di studiare il comportamento vibratorio del meccanismo di distribuzione desmodromica impiegato nei motori da competizione Ducati. Il modello elastodinamico della distribuzione comprende diversi effetti non lineari e, in particolare, l’albero a camme viene modellato con elementi finiti di Timoshenko. Le forze non lineari dei supporti sono derivate con una formulazione analitica relativa al cuscinetto di lunghezza finita. I risultati della simulazione numerica sono confrontati con rilievi sperimentali in termini di accelerazione valvola e velocità angolare dell’albero a camme. Tale confronto mostra che il modello consente di prevedere adeguatamente il comportamento del sistema reale. Infine, mediante la simulazione vengono esaminate ed analizzate le orbite dell’albero a camme.

1. INTRODUCTION
This paper deals with a kineto-elastodynamic model of the desmodromic valve train of a Ducati motorbike engine. In comparison with the widely-used trains having a closing spring [1–3], the desmodromic valve train – i.e. a mechanism with positive-drive cams – presents
different dynamic behaviour, as shown in [4–6]. As a matter of fact, the dynamics of the classic valve train mechanisms is mainly influenced by the valve spring, which is not present in the desmodromic trains.

In order to get insight into the dynamics of the desmodromic cam system, and to help the development and validation of the elastodynamic model, valve motion measurements were retrieved from experimental tests carried out at the DIEM Laboratory of the University of Bologna on cooperation with Ducati [5].

Nowadays increasing attention is addressed to the elastodynamic analysis of the mechanisms operating at high-speed. Such a study allows the estimation of the actual dynamic forces, impacts, and mechanism performances, as well as the design optimisation and fault diagnostics [7–10]. 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 mechanism 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 may arise.

In the specific field of valve trains for engines that operate at very high speed, 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, 11].

The model presented in this paper describes 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, the elastic flexibility of all links (including the compliance of the driveline), and the presence of several non-linearities. The camshaft, which is supported by hydrodynamic journal bearings, is modelled as Timoshenko beams with effect of gyroscopic moment. The non-linear bearing forces are analytically obtained under the finite-length bearing assumption.

Conversely, in the literature, works dealing with cam’s mechanism modelling, usually consider the camshaft as modelled with few degrees of freedom; in addition, the camshaft is often mounted on rigid bearings or isotropic flexible bearings [1–4, 8, 10].

Firstly, the mechanical system and the measurement apparatus are described. Secondly, the model is presented and the numerical results are compared with the experimental data in terms of valve acceleration and camshaft angular velocity. By means of this comparison, the effectiveness of the model is satisfactorily assessed. In addition, in order to show the effect of both camshaft and bearing modelling, the results of the present model are compared with those obtained by a previous model developed by the authors [6], which had a lumped-parameter camshaft mounted on rigid bearing. Finally, the simulated camshaft centreline trajectories are examined and the effects due to the cam mechanism forces are analysed.

2. THE MECHANICAL SYSTEM AND THE TEST APPARATUS

2.1. The mechanical system

This work deals with the timing system of the fourcylinder ‘L’ engines of Ducati racing motorbike which has double overhead camshafts, desmodromic valve trains and four valves per cylinder. In particular, the fourcylinder engine has four camshafts having each four conjugate cams: two camshafts drive the intake valves and the other two the exhaust valves.
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. On the test bench is mounted only one cylinder head, and their two camshafts are moved by means of an electrically powered driveline. The driveline, that is schematically shown in Fig. 1(a), 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 – and, in the motorbike engine, it corresponds to the shaft moved by 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 the real motorbike.

As mentioned before, each camshaft of the real engine has four conjugate cams; however, since on the test bench takes place only one cylinder head, the tested camshafts are different from the real ones. In particular, each of them has only one conjugate cam, that is, the test bench makes it possible to run only two valves: one exhaust valve and one intake valve. During the tests relative to the experimental results reported in this work, only the intake valve was present.

The schematic of the cam mechanism driving a single valve is shown in Fig. 1(b): the discs of a conjugate cam are each in contact with a rocker; the two rockers are then 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 separation of the cam discs from rockers takes place 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.

It is worth noting that only the components required for the operation of the valve train are included into the system. The system response is therefore dissimilar from the actual one (i.e. the response of the motorbike engine system). However, the inclusion (or exclusion) of the forces due to compressed gases does not compromise the validity of the experimental data as a tool for model validation [11].

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 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

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 mechanism driving a single intake valve. The cam mechanism driving the exhaust valve is not included because it was not present during the experimental tests.

In [6] the authors presented a lumped-parameter model of desmodromic valve trains having twelve degrees of freedom (dofs). That model included only one camshaft, while the presence of second camshaft was approximately introduced. In addition, in that study the camshaft was considered as mounted on rigid supports and the torsional and bending camshaft compliance were simply modelled by means of five dofs.

Conversely, as will be described in Section 3.2, in the present work a finite-length analytical model is employed for bearing modelling, in order to properly consider the presence of the hydrodynamic journal bearings that support the camshaft. In addition, both the camshafts are included; in particular, they are modelled as Timoshenko beams with effect of
gyroscopic moment. Obviously, the resultant model is more complicated than the previous one, but it leads to a better simulation of the dynamic behaviour of the valve train, as will be shown in Section 4.1.

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 (including the compliance of the driveline), 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).

With reference to Fig. 2, the known model input is the coordinate $\phi_0$, 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 $B_1$ 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 torsional stiffness $k_{b2}$ represents the stiffness of the belt transmissions $B_2$ between the intermediate shaft and the camshafts.

![Figure 2 – Schematic of the model.](image)

The coordinate $\theta_{I(E)}$, represents the angular position of the pulley fitted to the intake (exhaust) camshaft. As mentioned before, the camshaft is modelled by the finite element method (see Section 3.2) but, for the sake of simplicity, Fig. 2 schematically represents the camshaft as a rectangle having two cam profiles that are linked to 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 seven dofs: three of them are related to the power belt transmission, while the remaining four dofs are devoted to the modelling of the cam-valve mechanism.

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. Moreover, also the Hertzian stiffness is variable, as it depends on the contact force; in the simulation it is evaluated instantaneously.

In order to appropriately model the effects of backlashes, the possibility of separation of the rockers from the cam discs and the adjuster is taken into account by means of the parameters \( d_{13}, d_{34}, d_{26}, \) and \( d_{46} \).

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, 8]. Coulomb friction forces have not been introduced into the model, due to their low value.

More details on the lumped-parameter portion of the model can be found in [4, 6].

### 3.2. Model of the camshaft

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 \( v_x \) and \( v_y \) in the \( X \) and \( Y \) directions respectively (to locate the camshaft centreline), and by the rotations \( \phi_x, \phi_y, \) and \( \theta \) about the \( X, Y, \) and \( Z \) axes, respectively (to define the cross section orientation).

![Figure 3 – Schematic of the camshaft.](image)

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:

\[ [M]_F \{ \ddot{u} \}_{xz} + [C]_F \{ \dot{u} \}_{xz} + [K]_F \{ u \}_{xz} = [F(t)]_{xz} \]  
\[ [M]_T \{ \ddot{u} \}_{yz} + [C]_T \{ \dot{u} \}_{yz} - \Omega^2 [G] \{ \dot{u} \}_{yz} + [K]_T \{ u \}_{yz} = [F(t)]_{yz} \]  
\[ [M]_b \{ \ddot{\theta} \}_{b} + [C]_b \{ \dot{\theta} \}_{b} + [K]_b \{ \theta \}_{b} = [F(t)]_{b} \]  

where \([M]_F\), \([C]_F\), and \([K]_F\) represent flexional (torsional) mass, damping, and stiffness element matrices respectively, \([G]\) takes into account the element gyroscopic effects, \([F(t)]_{xz}\) is the flexional forcing term that acts on \(X-Z\) plane, while \([F(t)]_{yz}\) is the torsional forcing term. The coordinate vectors appearing in the Eqs. (1–3) represent the coordinates of the nodes of the \(j\)-th finite element:

\[ [u_{xz}(t)]^T = [v_i(t) \ \psi_i(t) \ \phi_{i+1}(t) \ \phi_{i-1}(t)] \]  
\[ [u_{yz}(t)]^T = [v_i(t) \ \phi_i(t) \ \psi_{i+1}(t) \ \psi_{i-1}(t)] \]  
\[ [\theta_r(t)]^T = [\theta_1(t) \ \theta_{i+1}(t)] \]  

The flexional matrices \([M]_F\), \([K]_F\), and \([G]\) are obtained according to [12]. In particular, a Timoshenko finite element was employed, thereby including transverse shear effects. As a result, all the flexional matrices are symmetric except the gyroscopic matrix \([G]\) which is skew-symmetric. The torsional matrices \([M]_b\) and \([K]_b\) are derived by assuming a linear shape function for the torsional finite element [13]. In addition, since the camshaft consists of circular cross elements, with different diameters, which are connected by circular fillets, the flexibility of the junction between two adjacent elements is taken into account by increasing of some amount the length of 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 \(\beta\): \([C]_{F(\theta)} = \alpha [M]_{F(\theta)} + \beta [K]_{F(\theta)}\) [15]. The coefficients \(\alpha\) and \(\beta\) were estimated by means of experimental vibration test carried out on the camshaft.

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

The 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. The adopted formulation is a combination of the asymptotic Ocvirk (short) bearing model and Sommerfeld (long) model,
which provides very accurate results for all L/D (length to diameter) ratios and eccentricity of
general interest [16]. With reference to Fig. 3(a), each support is divided into two identical
parts, that is, the model counts four finite-length bearings. By referring to the notation of Fig.
3(b), the bearing forces acting on the node that corresponds to the generic bearing are
expressed as:

\[
F_x = F_x (y, \nu, \kappa, \delta, D, L, \delta, \mu);
\]

\[
F_y = F_y (y, \nu, \kappa, \delta, D, L, \delta, \mu)
\]

(7)

where \( \delta \) is the radial clearance and \( \mu \) is the fluid viscosity.

### 3.3. Model solution

Seen as a whole, the model includes the lumped-parameter model of the power belt
transmission, the finite element model of the two camshafts, and the lumped-parameter model
of the cam-valve mechanism. The lumped-parameter portion counts seven dofs, while the
camshaft finite element models consist of about 370 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. In
addition, 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. In order to reduce the computation time of the numerical
integration, due to the high number of dofs, a ‘pseudo-modal’ reduction method was applied
[17]. In particular, only 15 mode shapes were considered for each camshaft, that is, only the
mode shapes corresponding to natural frequencies lower than 20 kHz were taken into account.
As a result of the reduction process, the model globally consists of 37 dofs.

### 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.

#### 4.1. Valve acceleration and camshaft angular velocity

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 (see Fig. 2). The
acceleration scale is made dimensionless with reference to the theoretical maximum value.

The experimental angular velocity of the camshaft is achieved by processing the digital
signal from the encoder fitted to the camshaft pulley. The numerical camshaft angular
velocity is therefore obtained from the torsional dof of the node which corresponds to that
pulley.

By comparing the experimental data with the numerical results, the effectiveness of the
model can be satisfactorily assessed. As a matter of fact, the comparison between Fig. 4(a)
and Fig. 4(c), shows good agreement between experimental and numerical valve acceleration.
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 negative acceleration phase, the simulated vibration frequency is slightly lower than the
real one. However, the model is able to reproduce the more important dynamic phenomena,
such as the impacts, and the level of the acceleration peaks is globally matched.

Figure 4 – Normalised valve acceleration: a) experimental; c) numerical; e) numerical. Camshaft angular velocity: b) experimental; d) numerical; f) numerical. Simulated intake (I) camshaft orbits: g) positive cam; h) negative cam; i) bearing 1-1; j) bearing 1-2; k) bearing 2-1; l) bearing 2-2. Simulated exhaust (E) camshaft orbits: m) bearing 1-1; n) bearing 1-2; o) bearing 2-1; p) bearing 2-2.
As a further example of the model accuracy, the actual and numerical camshaft motion are compared in Fig. 4(b) and 4(d) in terms of camshaft angular velocity. The comparison shows that the model is able to predict the camshaft motion very well. In particular, both the amplitude and frequency of the camshaft oscillation are correctly reproduced. It is noteworthy that it is essential to provide insight into the camshaft motion, so to supply further information on the dynamics of the timing belt transmission.

In order to show the effect of modelling the camshafts by finite element method, and considering the presence of the hydrodynamic journal bearings, the results of the present model are compared to those from a model having lumped-parameter camshafts mounted on rigid supports. Such a simplified model counts 17 dofs (3 refer to the driveline, 5 dofs per camshaft, and 4 related to the cam-valve mechanism), and gives the results reported in Fig. 4(e) and 4(f). In terms of valve acceleration the numerical result is still similar to the experimental one but many discrepancies are present. In particular, during the positive acceleration phases, the acceleration peaks are highly overestimated [see Fig. 4(e)]. Moreover, the simulated camshaft angular velocity evidences how the high frequency content is nearly absent [see Fig. 4(f)]. Such a poor torsional dynamic behaviour is probably due to the inadequacy of the lumped-parameter camshaft model with respect to the finite element camshaft model.

4.2. Simulated camshaft orbits

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. From this point of view, this Section is dedicated to analysing the camshaft orbits, which are not available from experimental measurements.

Figures 4(g–p) show the polar plots of the camshaft centreline position at the cam and bearing locations. The angles $0$ and $\pi/2$ of the polar coordinate system correspond to the rectangular coordinates $\xi$ and $\eta$ of Fig. 5, respectively. In addition, Fig. 5 shows the resultants of the average belt loads acting on the camshaft pulley axes, thus helping the interpretation of the whirl orbits.

![Figure 5 – Orbit reference frames and resultants of the average belt loads.](image)

Figures 4 (g–l) report the simulated orbits of the intake camshaft. In particular, the Figs. 4(g) and 4(h) show the camshaft centreline positions at the cam locations. Since the two cam discs are very close together, their orbits are similar. With reference to the negative cam, the trajectory exhibits two phases: the cam axis draws an ellipse portion between 150 and 300 degree angles, which corresponds to the dwell phase of the valve motion, while the orbit
results more irregular between 300 and 100 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. Obviously, the cam axis orbits and the camshaft centreline trajectories at the location of the bearings 2-1 and 2-2 [Figs. 4(k) and 4(l)] – i.e. the bearings close to the cams – are similar in shape. The bearings close to the camshaft pulley, namely the bearings 1-1 and 1-2, exhibit a trajectory that approximately covers the third quadrant of the polar coordinate system [Figs. 4(i) and 4(j)]; this agrees with the direction of the resultant of average belt loads (see Fig. 5).

Figures 4 (m–p) 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. 4(n)], and the axis which is not parallel to the Z axis. This can be stated by observing that the orbit centres of the bearings 1-2 and 2-2 are not in the same quadrant [see Fig. 4(p)]. Such a dynamic behaviour of the exhaust camshaft is due to the absence of the cam-valve mechanism, that is, on this camshaft do not act those high dynamic forces that origin the complex whirl orbits of the intake camshaft. The main forces that take action on the exhaust camshaft are the belt load resultant and the rotating unbalance forces.

As a final remark, it is noteworthy that the orbit centres of the bearings close to the camshaft pulley (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, the bearings close to the pulley have an attitude angle, which has the shaft rotation direction, of 30 degrees.

5. CONCLUSIONS

This paper deals with a kineto-elastodynamic model of the desmodromic valve train of a Ducati motorbike engine. 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.

The model takes into account the mass distribution, the link elastic flexibility, and the presence of several non-linearities. The camshaft, which is supported by hydrodynamic journal bearings, is assumed as a flexible rotor-bearing system that consists of a rotor composed of discrete discs, rotor segments with distributed mass and elasticity, and discrete bearings. The non-linear bearing forces are analytically obtained under the finite-length bearing assumption. 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 mechanism.

The numerical results are compared with the experimental data in terms of valve acceleration and camshaft angular velocity. By means of this comparison, the effectiveness of the model is satisfactorily assessed. Moreover, the simulated camshaft centreline trajectories are examined and the effects due to the cam mechanism forces are analysed.

The results show that the model can be a very useful tool in order to predict and understand the actual dynamic behaviour of the system, and to estimate the magnitude of forces and impacts, so to be used in both design optimisation and diagnostics.
REFERENCES


