A MODEL FOR THE ELASTODYNAMIC ANALYSIS OF A DESMODROMIC VALVE TRAIN

G. Dalpiaz and A. Rivola
Department of Mechanical Engineering (DIEM), University of Bologna
Viale Risorgimento, 2, I-40136 Bologna, Italy
Phone: ++39 051 644 3461, Fax: ++39 051 644 3446, e-mail: giorgio.dalpiaz@mail.ing.unibo.it

Abstract

This paper presents a lumped-parameter model of a motorbike engine’s desmodromic valve train, as a tool for the kineto-elastodynamic analysis with the aim of simulating the main dynamic effects. The model is non-linear and highly time-varying. The parameter estimation is discussed; the model is then satisfactorily validated by comparison with experimental results, thus, it may be a useful tool for design optimization, as well as fault diagnostics. Finally, some applications to the detection of unacceptable dynamic phenomena are shown.

Cam, Desmodromic valve train, Kineto-elastodynamics, Non-linear lumped-parameter model.

1 Introduction

Many studies on modelling and dynamic response of flexible mechanisms are present in the literature [Erdman et al. 1972, Koster 1974, Sandor et al. 1984, Dresner et al. 1993]. They deal with mechanisms operating at such a high speed that the dynamic behaviour is deeply affected by link elastic flexibility and mass distribution, as well as the effects of backlash and friction in joints. As a consequence, motion alterations may occur, causing mechanisms to fail in the proper execution of their tasks; high accelerations and dynamic stress levels may also produce early fatigue failures, and high level of vibration and noise may arise [Dalpiaz et al. 1992, Dalpiaz et al. 1995]. In the particular field of valve trains for high-performance engines, these dynamic effects are particularly important, since they may cause serious functional troubles, such as jump and bounce phenomena, as well as faults. Thus, increasing attention is addressed to the elastodynamic analysis, used in design optimization and diagnostics, in order to predict the actual dynamic behaviour, forces and impacts, and to identify the causes of failures and poor performances. However, only works on the widely-used trains with closing spring were found in the literature [Pisano et al. 1983, Nagaya et al. 1993, Özgür et al. 1996]. In that case the valve spring plays an important role in the system dynamics and its accurate modelling is required. On the other hand, in the case of desmodromic valve trains - i.e. mechanisms with positive-drive cams - the dynamic effects are partly different, as studied in this paper.
2 Description of the model

This work concerns the timing system of the twincylinder “L” engines of Ducati racing motorbikes, having double over-head camshafts, desmodromic valve trains and four valves per cylinder. This system is shown in the general view of Fig. 1(a): two camshafts, driven by a timing belt, have each two conjugate cams; one camshaft drives the two intake valves and the other the two exhaust valves. 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 located at the tip of the valve; the two cam discs act on the adjuster, by means of the rockers, in opposite directions. Thus it is possible to identify two parts of the mechanism, each made up of one of the cam discs and the related rocker: they give valve acceleration respectively in positive and negative directions, where the positive direction is considered to be the one of the opening valve. Here the terms ‘positive’ and ‘negative’ cam disc/rocker are used; however these links are commonly, but improperly, called ‘opening’ and ‘closing’ cam disc/rocker, respectively. It is noteworthy that there is also a small helical spring mounted around the negative rocker pin and properly preloaded; its main function takes place during the dwell phase, when there is separation of the cam discs from the rockers: this spring applies to the rocker the force required for getting the proper contact force between valve-head and seat.

With respect to the more widely-used trains with closing spring, the desmodromic trains make it possible to give very high valve accelerations, preventing the follower from jumping off the cam, without employing a very stiff closing spring; on the other hand, the mechanical complexity of the desmodromic system is justified only in high-speed engines with single-cylinder heads, as Ducati engines.
In order to build an effective, but simple model of a flexible mechanism, the main difficulties are the proper choice of the number of degrees of freedom, appropriate modelling of non-linearities, estimation of the model parameters and choice of the solution method. In the case of the above-described valve train, a lumped-parameter model with eight degrees of freedom was developed, taking into account the mass distribution, the elastic flexibility of all links (including the bending and torsional compliance of the camshaft), the Hertzian contact stiffness, the variability of transmission ratios with mechanism position, the backlash in joints, the damping and squeeze effects. In particular, it is worth noting that the active cam-follower system is different in the phases of positive and negative acceleration and the system is consequently highly time-varying.

2.1 Choice of the degrees of freedom

The model, shown in Fig. 2, describes the intake camshaft and the cam mechanism driving a single intake valve, namely the valve near the pulley [Fig. 1(a)]. In order to obtain a simple model, the other cam mechanism and valve driven by the same camshaft was not included; however this second mechanism is identical to the first and theoretically moves in phase; for this reason, the effects of the neglected mechanism are approximately introduced, assuming that this mechanism applies the same forces and torques to the camshaft as the first one and taking into account this assumption in the evaluation of model parameters.

With reference to Fig. 2, the known model input is the coordinate $\theta_0$, representing the angular displacement of the pulley fitted on the camshaft. In fact this pulley is assumed to rotate at constant speed, thus neglecting the dynamic effects of the transmission driving the pulley. The torsional compliance of the camshaft is taken into account by introducing the coordinates $\theta_1$ and $\theta_2$, representing the angular displacements of the positive and negative cam discs, respectively. As a matter of fact, for the considered valve, the positive cam disc is the nearest to the pulley. The moments of inertia $J_1$ and $J_2$, associated to these coordinates, are obtained by properly lumping the moment of inertia of portions of the camshaft together with the moments of inertia of the positive and negative cam discs, respectively. Consequently, torsional stiffness $k_{01}$ concerns the portion of the camshaft between the pulley and the positive cam, while torsional stiffness $k_{12}$ is relative to the portion between the two cam discs.

The camshaft bending compliance is also considered: the coordinates $x_7$ and $x_8$ are the deflections of the camshaft in correspondence of the conjugate cam and in two orthogonal directions, respectively coincident and perpendicular to the direction of the valve motion, as shown in Fig. 1(b). The deflections are assumed to be the same for the two cam discs, due to their small axial distance. The masses $m_7$ and $m_8$, associated to the bending coordinates, are the equivalent masses of the whole camshaft, computed by using the assumed-mode method with a shape function coincident to the static deflection; the camshaft bending stiffness are modelled by the springs $k_7$ and $k_8$; the camshaft bearing compliance is considered as negligible.

All the other coordinates are reduced to the direction of the valve motion. Thus the model contains the kinematical relationships of valve displacement vs. cam angular displacements and camshaft deflections: the cam angular displacements, $\theta_1$ and $\theta_2$, are reduced using the cam curve functions $x_1 = x_1(\theta_1)$ and $x_2 = x_2(\theta_2)$, which are theoretically identical and, in practice, very similar; the camshaft deflections, $x_7$ and $x_8$, are reduced using the appropriate
transmission ratios $\tau_{71}=dx_1/dx_7$, $\tau_{72}=dx_2/dx_7$, $\tau_{81}=dx_1/dx_8$ and $\tau_{82}=dx_2/dx_8$; these contributions are then combined. These operations are represented by a lever system in the model schematic of Fig. 2.
The angular displacements of the positive and negative rockers, respectively indicated by $\theta_3$ and $\theta_6$ in Fig. 1(b), after the reduction to the direction of the valve motion, become the linear coordinates $x_3$ and $x_6$. All the related model parameters are accordingly reduced to the same direction. The masses $m_3$ and $m_6$ are obtained reducing the moments of inertia of the rockers. The stiffnesses $k_{13}$ and $k_{34}$ are firstly computed by reducing and composing in series the stiffness of the positive rocker and the stiffness of the Hertzian contacts cam-rocker and rocker-adjuster, and then sharing out the global compliance between the two springs. Stiffness $k_{26}$ is obtained by reducing and composing in series the stiffness of the negative rocker arm in contact with the cam and the stiffness of the related Hertzian contact; similarly, $k_{46}$ concerns the stiffnesses of the negative rocker arm in contact with the adjuster and the related Hertzian contact; the bending compliance of the negative rocker pin is also included in the evaluation of $k_{26}$ and $k_{46}$. It is noteworthy that the Hertzian contact stiffness depends on the contact force \cite{Johnson1985}; in the simulation it is evaluated instantaneously. Moreover, stiffness $k_6$ is the reduced stiffness of the helical rocker spring. In order to appropriately model the effects of backlashes, the possibility of separation of the rockers from the cam discs and the adjuster is also included ($\delta_{13}, \delta_{34}, \delta_{26}, \delta_{46}$). The mass valve is lumped partly in mass $m_4$, in correspondence of the adjuster, and partly in mass $m_5$, in correspondence of the valve-head. The axial stiffness of the valve stem is $k_{45}$, while stiffness $k_5$ represents the stiffness of the valve-head in contact with the seat.

A viscous damper is associated with each stiffness in the model, in order to globally take account of structural damping as well as other damping. When there is separation between links, the related damper coefficient is computed in order to represent the lubricant squeeze effect in the backlash \cite{Koster1974, Dalpiaz1992}. Moreover, Coulomb friction forces might easily be taken into account \cite{Dalpiaz1995}, but they have not actually been introduced.

This model leads to non-linear differential equations of motion that are numerically integrated, using the software Simulink®.

2.2 Estimation of the model parameters

Particular attention is paid to the estimation of the model parameters. The values of the inertial parameters, stiffnesses and squeeze coefficients were preliminarily computed on the basis of the dimensions of the links; the finite element method was used for computing the stiffness of the rockers and the valve-head. The rocker stiffnesses actually depend on the mechanism position, but each of them was computed in only one configuration - the position of zero and maximum valve displacement, for the positive and negative rocker, respectively - and assumed as constant. The coefficients of the dampers were preliminarily established based on literature data \cite{Dalpiaz1992, Dresner1993, Dalpiaz1995}.

Using experimental data of the valve motion, measured on a test bench, stiffnesses and damper coefficients were then adjusted in order to better match experimental results. In particular, some stiffnesses were reduced, as commonly occurs in modelling \cite{Dalpiaz1992, Dresner1993}, and the values of the proportionality constants between damper coefficients and stiffnesses were taken in the range $0.6 \times 10^{-5}$ to $3.5 \times 10^{-5}$ s. The values of the
model parameters are listed in Table 1. The values of $\delta_{13}$, $\delta_{34}$, $\delta_{26}$, $\delta_{46}$ refer to the amount of separation in the contacts during the dwell phase.

Table 1. Model parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Parameter</th>
<th>Value</th>
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<tbody>
<tr>
<td>$J_1$</td>
<td>$9.38 \times 10^{-6}$ kg m$^2$</td>
<td>$k_{01}$</td>
<td>$1.275 \times 10^4$ Nm/rad</td>
</tr>
<tr>
<td>$J_2$</td>
<td>$5.13 \times 10^{-5}$ kg m$^2$</td>
<td>$k_{12}$</td>
<td>$1.370 \times 10^5$ Nm/rad</td>
</tr>
<tr>
<td>$m_3$</td>
<td>$0.019$ kg</td>
<td>$k_{13}$</td>
<td>$214.4^*$ MN/m</td>
</tr>
<tr>
<td>$m_4$</td>
<td>$0.032$ kg</td>
<td>$k_{26}$</td>
<td>$8.5^*$ MN/m</td>
</tr>
<tr>
<td>$m_5$</td>
<td>$0.037$ kg</td>
<td>$k_{34}$</td>
<td>$214.4^*$ MN/m</td>
</tr>
<tr>
<td>$m_6$</td>
<td>$0.032$ kg</td>
<td>$k_{46}$</td>
<td>$15.5^*$ MN/m</td>
</tr>
<tr>
<td>$m_7$</td>
<td>$0.573$ kg</td>
<td>$k_{45}$</td>
<td>$76.0$ MN/m</td>
</tr>
<tr>
<td>$m_8$</td>
<td>$0.573$ kg</td>
<td>$k_5$</td>
<td>$172.0$ MN/m</td>
</tr>
<tr>
<td>$\delta_{13} = \delta_{26}$</td>
<td>$0.2$ mm</td>
<td>$k_7$</td>
<td>$83.4$ MN/m</td>
</tr>
<tr>
<td>$\delta_{34} = \delta_{46}$</td>
<td>$0.0$ mm</td>
<td>$k_8$</td>
<td>$83.4$ MN/m</td>
</tr>
</tbody>
</table>

* Neglecting the Hertzian stiffness (variable)

3 Results and discussion

Figure 3 shows the comparison between experimental [Fig. 3(a)] and numerical [Fig. 3(b)] head-valve acceleration in the case of crankshaft speed equals 8000 rpm (corresponding to 4000 rpm of the camshaft) and for one camshaft revolution. Good agreement between simulated and experimental acceleration was generally attained. More specifically, the dynamic behaviour of the mass $m_5$ of the model is very similar to the actual motion of the valve during the first phase of positive acceleration; in addition, in the second positive acceleration interval, the model matches the highest peaks of experimental acceleration even if it is not able to satisfactorily describe the trend of the actual oscillations. On the other hand, during the negative phase of acceleration, the model does not adequately simulate the experimental valve motion; however, the value of the minimum acceleration peak is rather well reproduced. In order to interpret the discrepancies between the numerical and experimental results, one can remember that in this work the rocker stiffnesses were assumed as constant. On the contrary they actually depend on the mechanism position; this variation obviously affects the dynamic response of the system. By observing the Fig. 3, the disagreement is more evident in the negative phase of acceleration; thus, it is possible that the negative rocker (which acts in this phase) has a highly variable stiffness. Generally, in order to improve the numerical results it seems reasonable to spend more effort in taking into account the stiffness variability.

The model is a very useful tool in understanding the dynamic behaviour of the actual system and, as such, can be used in design optimization and diagnostics. As a matter of fact, it makes it possible to relate the valve acceleration pattern to the dynamic phenomena occurring in the
mechanism. As an example, Fig. 4(a) reports the contact force between the inertia element $J_2$ (the negative cam) and the mass $m_6$ (the negative rocker); in addition, the contact force in the kinematic pair connecting the negative rocker (mass $m_6$ of the model) and the backlash adjuster (mass $m_4$) is shown in Fig. 4(b). The first force is reduced to the direction of the valve motion; its actual value can be obtained by multiplying the force by the ratio of the distances of the rocker-adjuster force and the cam-rocker force from the rocker axis. Since this ratio has unitary mean value, the amplitude of the force shown in Fig. 4(a) is approximately equal to the actual one. The contact forces can be used, for example, in order to verify the structural strength of the rockers or the contact pressure. Moreover, Fig. 4 shows that the contact between negative cam and rocker is mainly present during the phase of negative acceleration, as is expected. On the other hand the contact between rocker and the backlash adjuster is assured over the whole camshaft cycle. This is due to the action of the rocker spring.

Figure 3. Head-valve acceleration corresponding to one camshaft revolution; crankshaft speed 8000 rpm: (a) experimental results; (b) numerical results.

Figure 4. Numerical contact forces between: (a) negative cam disc and rocker, reduced to the direction of the valve motion; (b) negative rocker and backlash adjuster.
As a further example of the model application, the contact between valve-head and seat is investigated. The first impact of the valve against the seat occurs at about 10 ms, as shown in Fig. 5 that represents the numerical valve displacement. This dynamic effect is particularly evident in Fig. 3(b); in fact a high acceleration peak appears at the same time position. Actually, in the experimental results [see Fig. 3(a)], the peak occurs slightly later but the source of the phenomenon is clearly the same. Bounce phenomena are also present, both in the experimental and numerical results: the first and highest bounce, as well as the consequent impact, are satisfactorily simulated. The amount of the separation due to the bounce may be estimated by means of the model.

![Figure 5. Numerical valve displacement: bounce phenomena and impacts.](image)

4 Conclusions

A kineto-elastodynamic model of a desmodromic valve train - whose typology is essentially different from the commonly-used trains with closing spring - was developed with the aim of taking into account all the important dynamic phenomena; the model contains some non-linear elements and is highly time-varying. The effectiveness of the model is then satisfactorily assessed by comparison with experimental data. In particular, the validity of the model assumptions was globally verified. However, there are some discrepancies, which might be due to the fact that rocker stiffnesses are not actually constant, as assumed in the model. Since the model makes it possible to relate the design and operation parameters of the mechanism to its actual dynamic behaviour, it may be a useful tool both in design optimization and diagnostics, in order to predict the magnitude of forces, impacts and bounces and to detect unacceptable dynamic phenomena.

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References


