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# Experimental methodology for the tappet characterization of timing system in I.C.E.

**G. Sequenzia · S.M. Oliveri · M. Calì**

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**Abstract** The aim of present work is the containment of the inertia forces, the stiffness components optimization and the fit tolerances of valve train in internal combustion engines (I.C.E.) 4T.

The proposed methodology allows, through the development of a test machine, the evaluation of axial stiffness of tappet depending on eccentricity of the cam tappet contact, performing a functional analysis that simulate the behaviour of the system in operational condition, even if, some adjustment of tolerances of the fit between tappet and his guide, occurred.

The dynamic study of the valve train, through modern computer codes, is performed by connecting lumped masses, springs and dampers that characterize each element. In numerical models the tappet is represented as constituted by the tappet and by the hydraulic element. Each of these elements is characterized by stiffness and mass. The structural rigidity of the tappet has, in fact, important effects on the dynamic behaviour of the entire valve train.

The test machine makes possible the choice of the dimensional and geometrical tolerances of the fit between tappet and his guide; allows furthermore the evaluation of errors occurred during construction and integration phase. In addition, the test machine is also suitable for reverse engineering applications, makes it possible to automatically draw the cam profile in polar coordinates.

**Keywords** Tappet · Timing system · Dynamics · Fit tolerance · Cam · I.C.E.

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## 1 Introduction

The dynamic analysis of the valve train, through the modern numerical codes involves a schematic lumped parameter system through the connection of masses, springs and dampers that characterize each element (Fig. 1a). The study of dynamic optimization of the system, made by multibody models, allows, through parameterization, to find the best values of the geometrical, and structural properties of stiffness and damping values to be assigned to the system. The quality of the final results in the study of optimization depend, mainly on the degree of detail of modelling and a careful evaluation of the contributions of individual components.

Multibody models usually train-set which is the valve tappet is outlined by the hydraulic tappet element (hydraulic lash adjuster) for the recovery of the backlash.

In this paper we have studied the deformation of the tappet in its real boundary conditions subjected to loads transmitted by contact between the tappet and the cam, considering the influence of this deformation

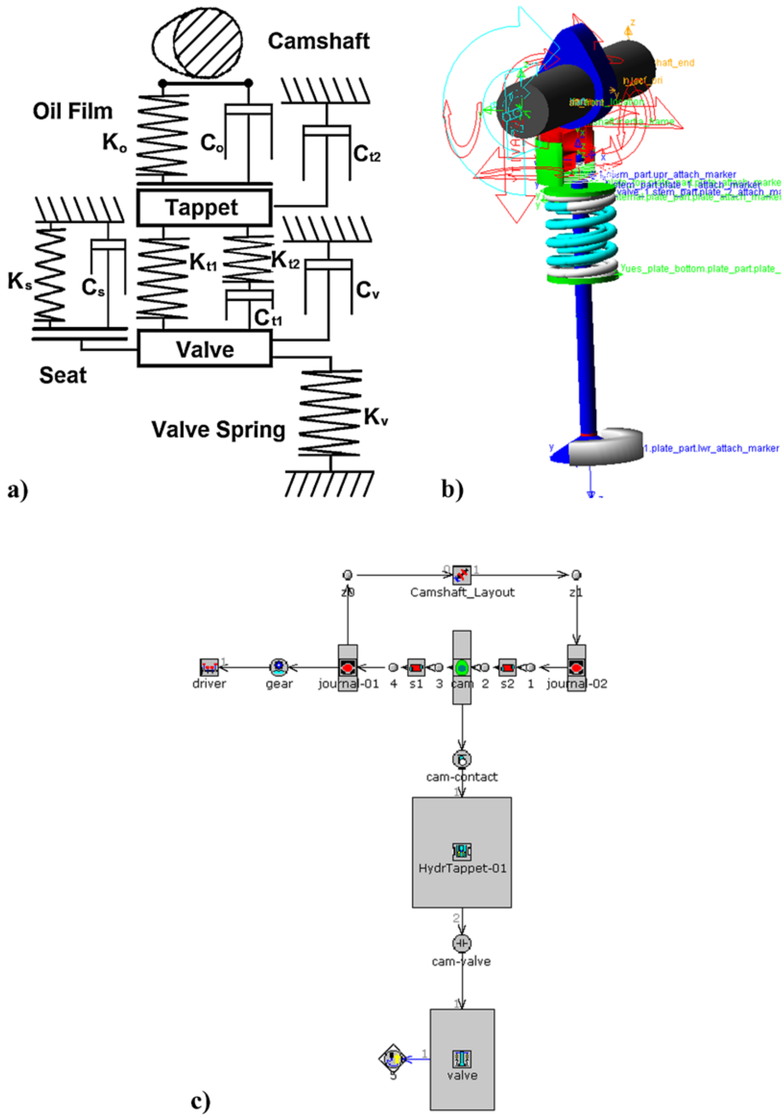


**Fig. 1** Lumped parameters model (**a**). Multibody model with MSC ADAMS (**b**). Multibody model with GT SUITE (**c**)

on the dynamics of valve train, using a test machine that allows perform with remarkable precision measurements of deformation, reproducing the actual operating conditions of mating.

By performing a functional analysis at different tolerances of the tappet in its housing is seen as the deformation value varies greatly depending on the eccentricity in the cam tappet contact, influencing both the contact pressure with the bud (CAM) that lubrication of the same contact.

The experimental analysis, carried out on the tappet through the test machine, has allowed to model, with great accuracy in terms of equivalent axial stiffness, the cam-tappet and tappet-valve contacts. In particular, by means of experimental measurements it was observed that $K_{t2}$ and $C_{t2}$ are functions of the contact eccentricity, as well as the angular position of the cam with respect to the tappet. Then, the lumped parameters modeling, shown in Fig. 1a, was developed. In this modeling, in addition to the values of lubricating oil film stiffness and damping ($K_0$ and $C_0$), to the values of valve spring stiffness and damping ($K_v$ and $C_v$) and to the values of stiffness and damping of valve impact with the abutment seat ($K_s$, $C_s$), the values of stiffness

$K_{t2}$ and damping $C_{t2}$ were introduced. The latter simulate the equivalent axial stiffness due to the play and to the geometric inaccuracies between the tappet and the cam. The stiffness $K_{t2}$ and damping $C_{t2}$ are added to the values of stiffness $K_{t1}$ and damping $C_{t1}$ due to the oil-air mixture present in the tappet. The exact values are given in Sect. 5.

In particular, the stiffness $K_{t2}$ was introduced in series with the damping $C_{t1}$, while the damping $C_{t2}$ acts in parallel to the stiffness $K_{t1}$.

The modeling described was introduced in the multibody models, developed with commercial codes. In particular, in Fig. 1b is possible to see the multibody model of the valve train developed with the code MSC-ADAMS, while in Fig. 1c is shows the functional diagram of multibody model developed with the code GT-Suite.

The deformation (or stiffness) in this coupling has important effects on the dynamic behaviour of the entire valve-motion, several studies of dynamics [1–3] demonstrate the importance of such stiffness as a function of the eccentricity in the cam tappet contact. The test machine was developed capable of accurately assessing the yielding of the coupling-tappet housing in the direction of the valve thus assessing an equivalent axial stiffness.

The paper is organized as follows: Sect. 2 describes the test machine developed in this paper; Sect. 3 introduces the problem of fit tolerance optimization; Sect. 4 describes the CAD/FEM (Computer Aided Design/Finite Element Analysis) methodology used in this paper; Sect. 5 shows the experimental results; Sect. 6 analyzes obtained results; Sect. 7 contains the conclusions.

## 2 Test machine

In previous work the authors have developed a test machine that can perform these measurements using the camshaft and the tappet of motor analyzed [4–6]. However, this system had some problems due to the fact that the readings of the measurements were performed directly on the load cell, in relation to stress, and on to goniometer relative to the camshaft angular position. This brings the inevitable mistakes, only partially offset by repeated readings. In addition, it was necessary to change, almost totally, the main components of the test machine, to vary the angular position of camshaft and tappet's guide.

The system described in this paper, developed with a design methodology MCAD/CAE (Mechanical Computer Aided Design/Computer Aided Engineering), consists of a tubular steel frame, a base, a rigid system able to apply an appropriate load, onto the tappet, a worm reduction unit, an angular potentiometer, a linear potentiometer, a load cell and a data acquisition system. Its modular structure allows to perform measurements on different types of tappets and camshafts.

The frame is the component that allows to the machine to be fastened to the structure of the hydraulic jack, that applies the load on the same chassis housing the worm gear reduction ratio of 1:100. We chose this type of gear for its irreversibility as it is necessary to maintain a fixed angular position of the camshaft during application of load. Keyed output shaft, the camshaft is located above, a multi-turn potentiometer angular (3 laps) 5 kΩ, which enables the system to

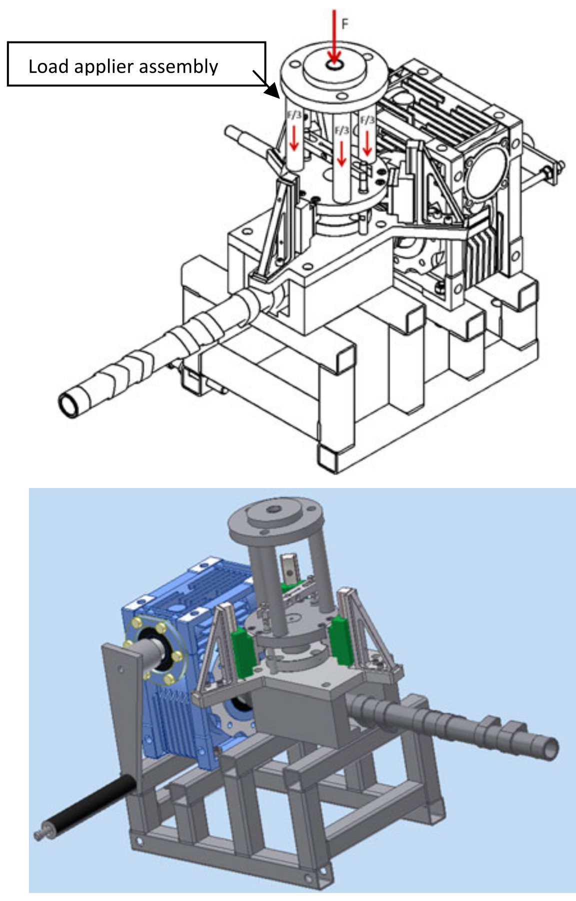


**Fig. 2** Test machine: 3D model

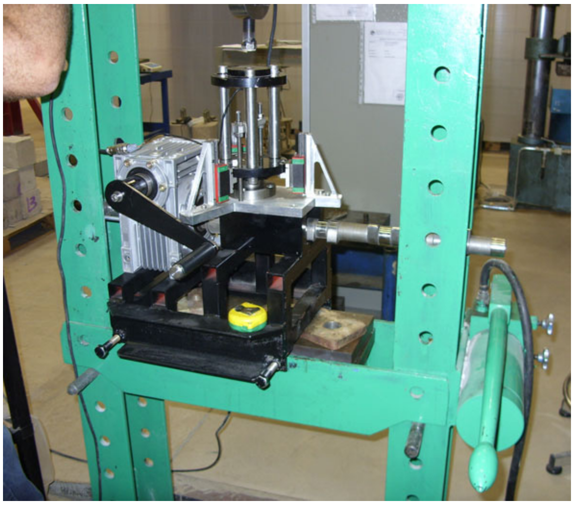

**Fig. 3** Test machine

obtain accurate positioning of the shaft angle of less than 0.1°.

A data acquisition system, processes the data from the angular and linear potentiometers and load cell, returning the information related to angular and linear positioning system with reference to the load. In order to apply, the correct load value, onto tappet, we using a load applier assembly, moved by three linear sliding blocks "HIWIN" with double linear ball bearings, fixed onto the base (Fig. 2). The use of the guide is necessary because it ensures the correct position (perpendicularity) between the camshaft and the load applier assembly, while allowing the latter to slide freely along its longitudinal axis. Furthermore in the load applier assembly is housed a linear potentiometer 130 from Penny & Giles SLS 3 kΩ, to measure the deformation on the upper surface (opposite surface of the tappet cam contact) of the tappet.

In Fig. 3 is shown a picture of the machine during the tests performed. In Figs. 4 and 5 (highlight potentiometers) you can see a diagram of the test mode with the flow of forces applied onto tappet and the position of the angular and linear potentiometers.

### 2.1 Working mode

To carry out the measures the deformation of the tappets, the forces are applied through an hydraulic actuator, controlled force, which transmits the load on the top plate. Three pillars forward it to the bottom plate and then to the rod transmitter load. The piston inserted at the end of the pin receives it and transmits the thrust load on the inner surface of the tappet. The tappet is constrained on the opposite side from contact with the cam. The equipment allows you to vary the angular position of the camshaft in order to achieve different eccentricity of the cam tappet contact. So we can get diagram of force and pressure in the Hertzian contact between the cam and tappet.

The yielding is measured by a linear potentiometer with accuracy of 0.001 mm, mounted on a coaxial structure, independent of the load head. Therefore the potentiometer is not sensitive to the yielding of the base during the tests. The point of measure, of

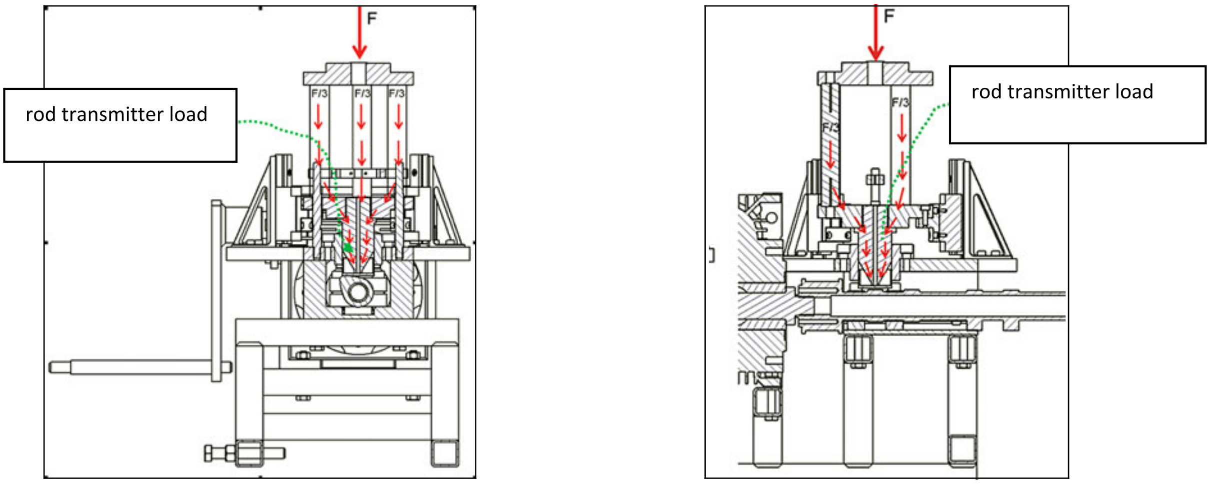


**Fig. 4** Test machine section and loads' scheme

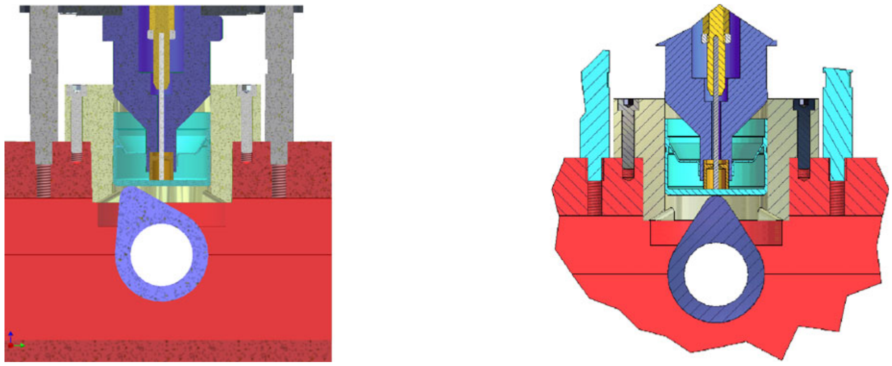

**Fig. 5** Cam tappet contact at different eccentricities

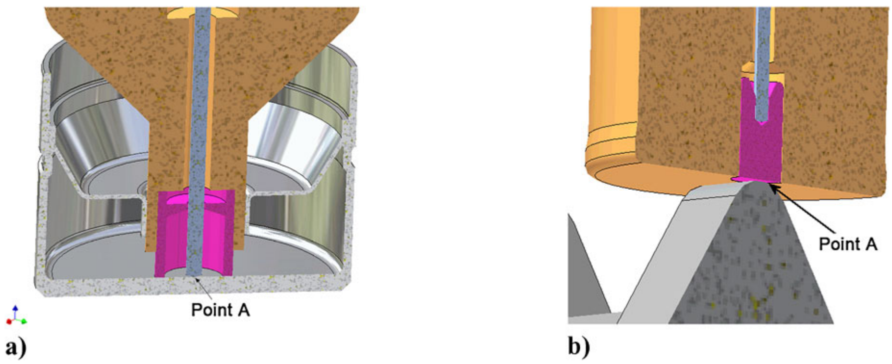


**Fig. 6** Axial compliance for tappet (**a**) and cam (**b**)

the experimental yielding, is the contact point between the potentiometer rod and the inner surface of tappet's head (Fig. 6).

### 2.2 Experimental test

The experiments were performed on two different types of tappet (Fig. 7). The first one ("tappet 1"), is a hydraulic bucket type tappet, on which was obviously removed the cylinder, the second ("tappet 2") is a mechanical bucket type tappet.

The geometrical and physical characteristics of the two types of tappet are listed in Table 1.

Each tappet was coupled to his seat with three different values of backlash and subjected to 8 cycles of loading with a maximum force of 5000 N.

To take into account the yielding of the shaft and the backlash of the system, the measures of the axial rigidity were cut of the yielding detected on the nose of the cam with no eccentricity and load more than 5000 N and the values of the yielding obtained with a full test by applying a force of 800 N in order to set the system.

## 3 Fit tolerance optimization

In literature you can find various data on the effects of manufacturing errors and assembly of the tappet on valve train dynamics [7–9]. In this study we want to define, through an experimental measure and integrated CAD/CAE models, the best allocation of tolerances based on evaluation of axial stiffness and hence on the better distribution of pressure and better lubrication conditions of the cam-tappet contact. Therefore, some critical issues in terms of pressure distribution and lubrication of the cam tappet contact can be improved with a proper set of dimensional tolerances and geometric profile tolerance.

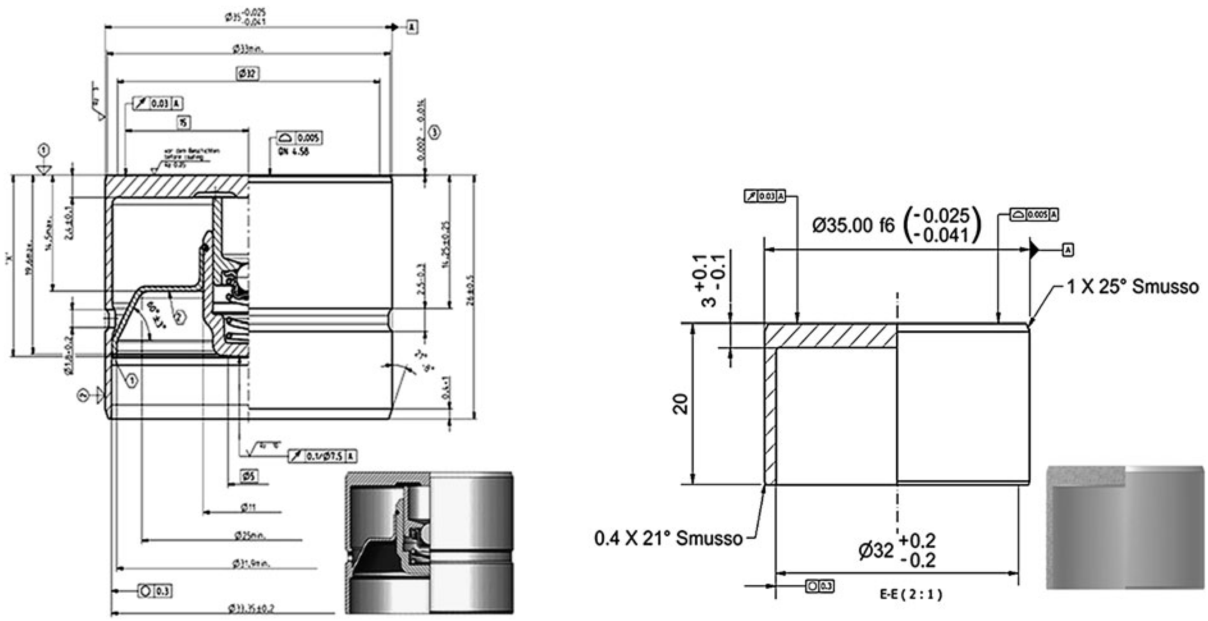


**Fig. 7** Tappet 1 and tappet 2

**Table 1** Comparison between tappet 1 and tappet 2

| | Tappet 1 | Tappet 2 |
|---|---|---|
| Weight [gr] | $55 \pm 2$ | 59 |
| Material | AISI 5120 | AISI 5120 |
| Diameter external [mm] | 35 | 35 |
| Thickness Cam/Face [mm] | 2,4 | 3 |
| Height [mm] | 26 | 20 |
| Stiffness [kN/mm] | 171 | 241 |

## 4 CAD/FEM modeling

In order to integrate and better understand the experimental results, the tappet's study was modeled in a Autodesk/Inventor and discretized with elements of CHEXA8 CPENTA6 type. In addition to the hydraulic tappets, were examined even the displacement of the cam shaft and its geometric support. This model is necessary to create a model that takes into account the deformation of the nose of the cam under the action of the test loads.

A relatively small number of mesh elements have studied the problem in a satisfactory manner with a lightweight and without making it excessively long calculation times.

Figure 8 shows the FEM models of the tappet and cam studied and the characteristics of the FEM models are shown in Table 2.

The models were constrained at the support side of the tappet and in the vicinity of the contact cam-tappet. To simulate the distribution of the load applied on the contact surface between the piston and the tappet has been used a MPC (Model Predictive Control) formulation with RBE2 element. The nodes of the element RBE2 (true rigid element). employees all belong to the interface of contact between the piston and the tappet, while the independent node has been placed on the axis of revolution of the tappet interface at a distance of 20 mm. On the independent node there was a force with an axial direction and intensity of 5000 N in agreement with the values used in the test. In this way, the nodes of the interface creates a load distribution consistent with the local stiffness of the component. The boundary conditions, placed along the line of contact on the interface tappet-cam, tie the knots along the direction of load application. The other two degrees of freedom have been constrained by blocking nodes placed on a piece of tappet where contact is made between it and the seat. This is the approximate semi-elliptical shape is the contact between two cylinders with axes at a point incident with a triangle of side 2b and height. Applying the theory of Hertz

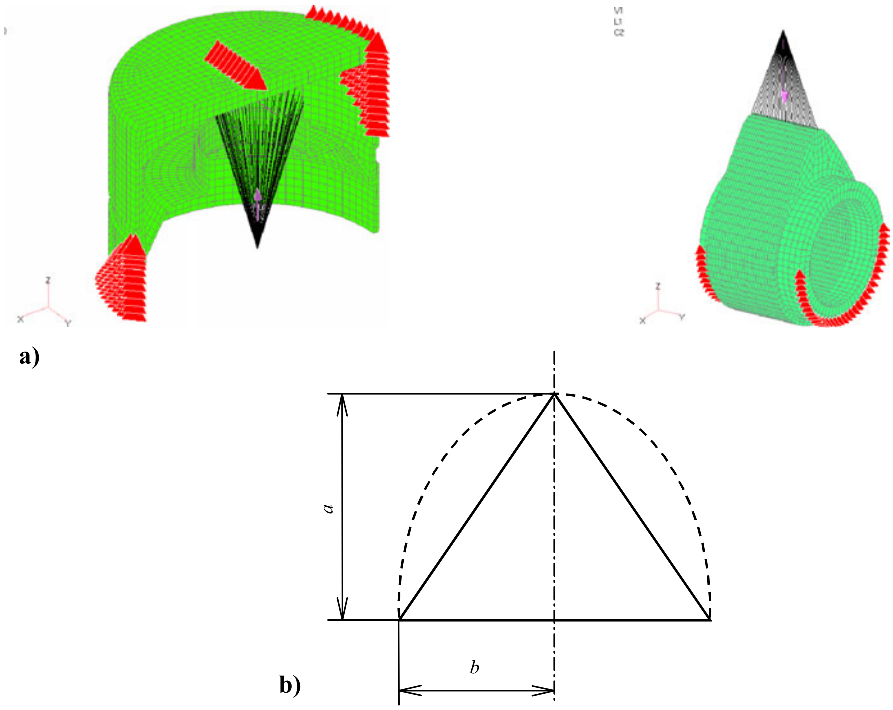


**Fig. 8** FEM model of tappet and cam (**a**); imprint in contact (**b**)

**Table 2** Node and element for tappets and cam-shaft

| | Tappet 1 | Tappet 2 | Cam-shaft |
|---|---|---|---|
| N° of node | 13804 | 13176 | 12703 |
| N° of element | 9625 | 9137 | 9962 |

were calculated extensions of $a$ and $b$ with an iterative procedure is derived and then the size of the footprints of contact between the tappet and its location on the head of the engine.

The FEM analysis of the tree returns the value of the independent movement of the node in the direction of the one degree of freedom (Fig. 9).

The stiffness of each tappet as a function of eccentricity was calculated through the work of deformation:

$$U = \int_0^{x_f} P(x) \cdot dx \tag{1}$$

In the case of linear elastic deformation can be written:

$$P(x) = k \cdot x \tag{2}$$

In summary knowing the deformation work done, it is possible to trace the value of the equivalent axial stiffness

$$k = \frac{2 \cdot U}{x_f^2} \tag{3}$$

The code provides for calculating the various results and the work of external forces $U$, so by (3) we can determine the value of $k$.

## 5 Experimental result

Figure 10 shows experimental measure of the performance characteristics of deformation and stiffness for the tappet 1. In Fig. 11 is visible the graph of deformation as a function of the eccentricity of the contact, their mean values, standard deviations and the margins of the area under the curve of distribution of $M - 3\sigma$ and $M + 3\sigma$ defining the 99.73 % of the cardinality of the results. The values "correct" take into account the displacement of the camshaft show, depending on the eccentricity of the contact, the curve of the equivalent axial stiffness.

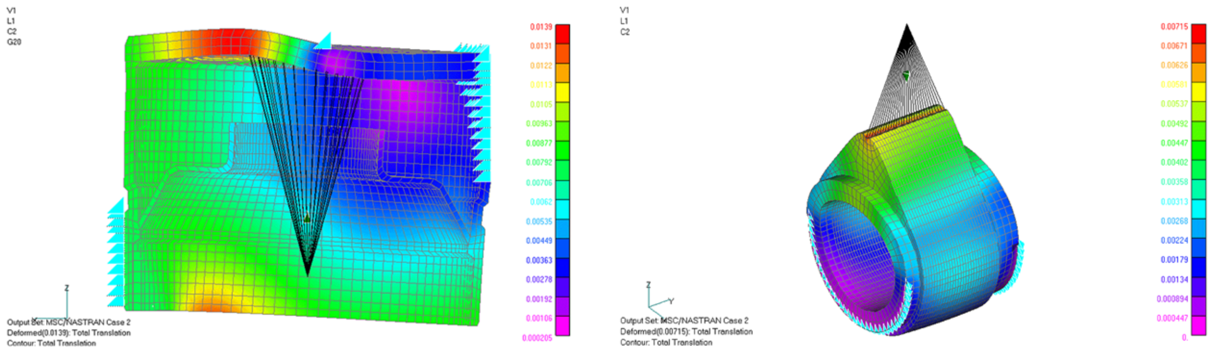

**Fig. 9** Cam and tappet 1 deformation with eccentricity of 6 mm

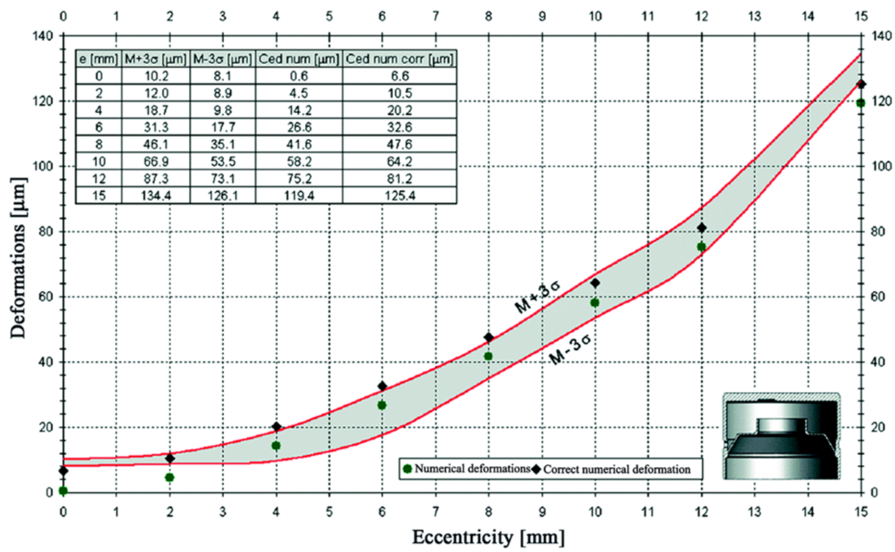


**Fig. 10** Deformation vs. eccentricity for tappet 1

Comparison of the two tappets are highlighted, with the same dimensional and geometric tolerances, higher values of axial stiffness equivalent are present in tappet 2. This result is probably due to the greater thickness of the cam-tappet face for tappet 2 (Fig. 12).

### 5.1 Experimental numerical correlation

The deformations obtained with the FEM analysis for tappets 1 and 2 were compared with experimental curves that define the margins of the distribution curve $M+3\sigma$ $M-3\sigma$ as a function of eccentricity (Fig. 13). Summing the numerical value of the compliance of the tappet to the tree yields a result comparable to the experimental one.

It is also noted that when the eccentricity is maximum (15 mm) the difference is due to an "edge effect" due to the fact that the constraint in the head (bud-contact tappet) is influenced by the rim of the tappet.

## 6 Result's analysis

Through measures performed with test machine was possible to evaluate the equivalent axial stiffness and the dynamic behavior of the valve motion with the two types of tappet and perform a functional analysis that simulates the behavior of the system in operation at different tolerance dimensional and at the variation of geometric tolerances on the surfaces.

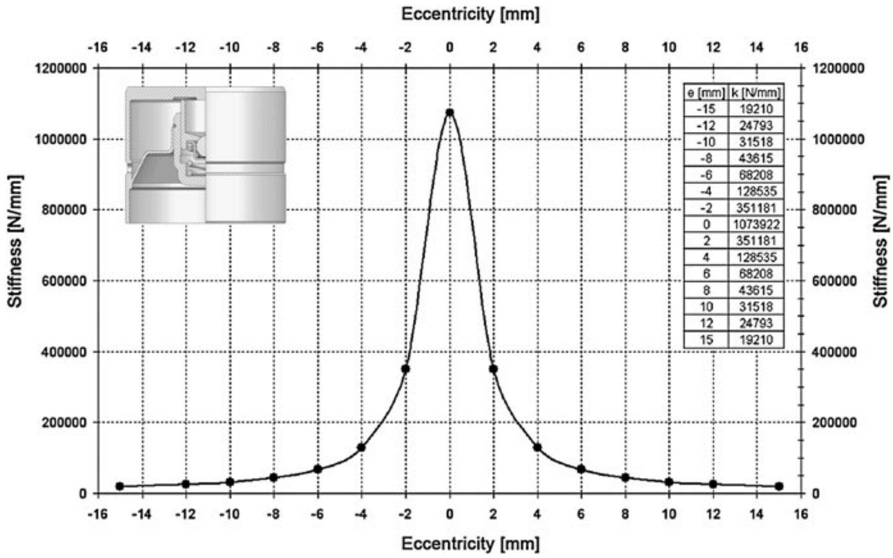


**Fig. 11** Stiffness vs. eccentricity for tappet 1

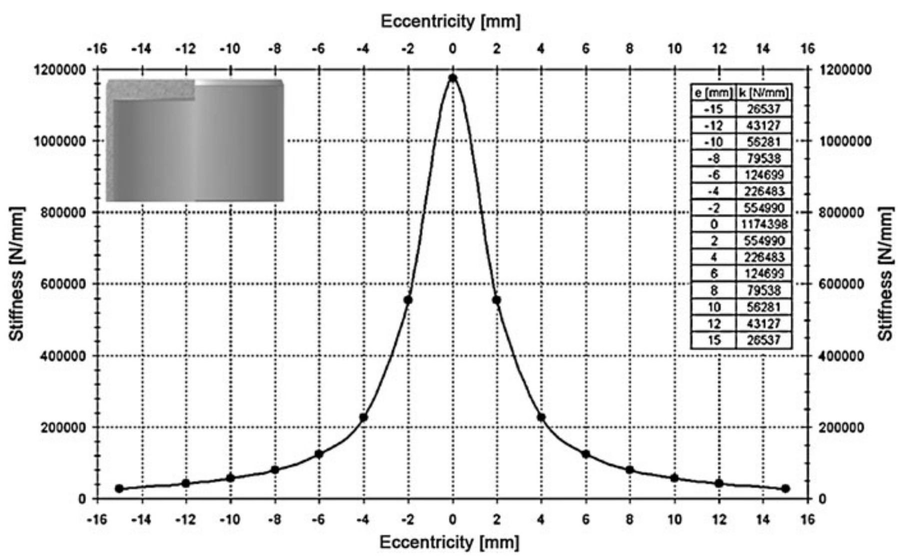


**Fig. 12** Stiffness vs. eccentricity for tappet 2

The tappets are inserted in their place with a base hole (H7) tolerance, usually using for the tappet a tolerance f6. At the upper surface of the tappet is usually assigned a tolerance value of the surface shape of 0.01 mm.

In particular it was estimated the value of the forces exchanged between the cam and tappet and the extent of pressures in the same Hertzian contact (Fig. 14) variating the above dimensional and geometric tolerances. The latter value (Hertzian pressure in the presence of friction) is particularly important in the 1st and 2nd node lubrication, that is, those points where the relative velocity between the cam and the tappet is zero and the lubrication conditions are critical (Fig. 15).

The estimation of these quantities is done at idle speed for the pressure and the resulting lubrica-

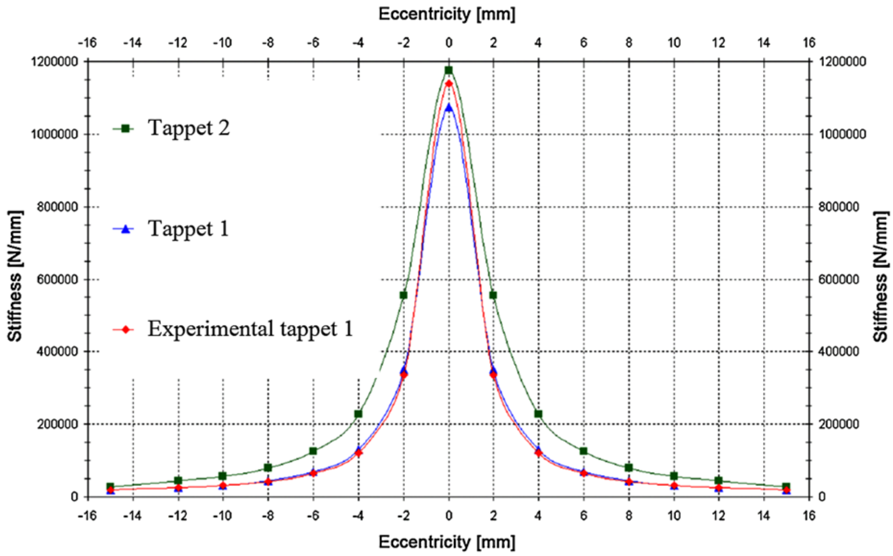


**Fig. 13** Numerical and experimental stiffness

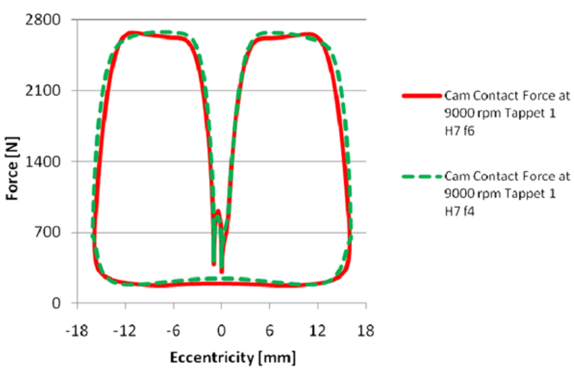


**Fig. 14** Force for H7/f6 and H7/f4 tolerances

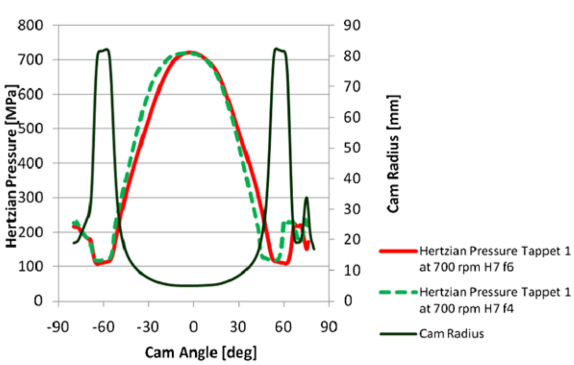


**Fig. 15** Hertz pressure for H7/f6 and H7/f4 tolerances

tion number and at the maximum speed for the forces exchanged with the cam at different tolerances.

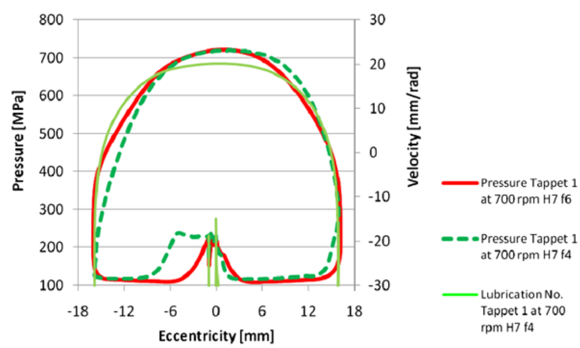


**Fig. 16** Hertz pressure for H7/f6 and H7/f4 tolerances

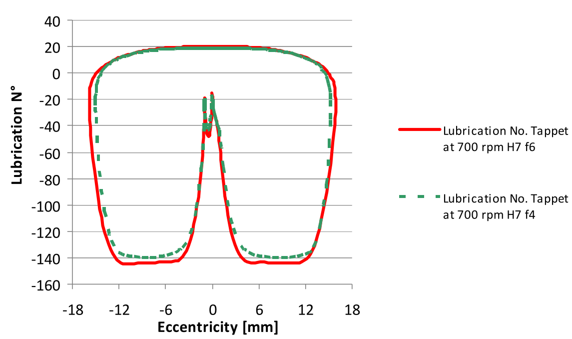


**Fig. 17** Lubrication N° for H7/f6 and H7/f4 tolerances

Figures 16–20 will show the results obtained for the tappet by using the values of tolerance indicated by the manufacturer of the tappet (H7/f6) made in compari-

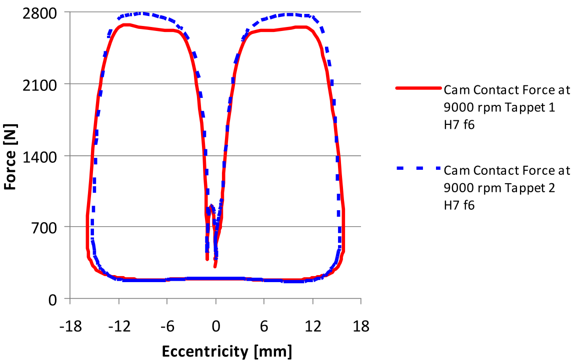


**Fig. 18** Contact force for tappet 1 and tappet 2

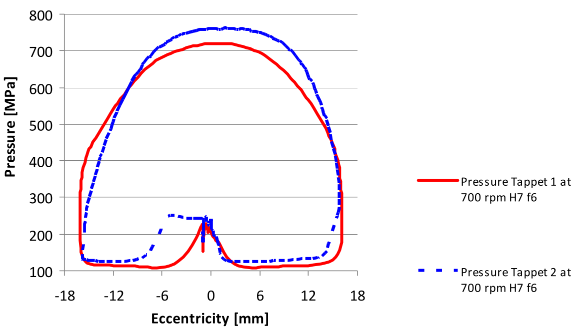


**Fig. 19** Contact pressure in tappet 1 and tappet 2

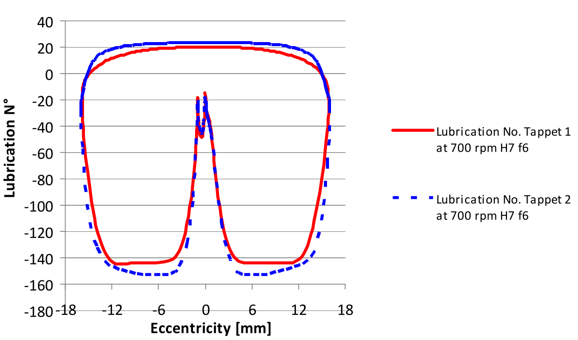


**Fig. 20** Lubrication N° for tappet 1 and tappet 2

son with the tolerance values proposed by the authors (H7/f4) to improve the system dynamics.

In fact, the simulations showed that the tappet in a mating with conditions close to those of maximum material, or with a dimensional tolerance value equal to f4, has better performance in terms of Hertzian pressure and lubrication number [10–12].

The improvement comes from a lower mass, lower inertia forces and a different pattern of equivalent axial stiffness.

## 7 Conclusions

The test machine developed provides a valuable tool for the designer to make the optimum choice of geometric and dimensional tolerances in tappet-cam and tappet-guide contact. The simple and precise determination of the curves of stiffness was the aim of this study. The proper characterization of the dynamic of valve train requires, in fact, a careful evaluation of the equivalent axial stiffness and its variation depending on tolerances of tappet with its headquarters and with the cam.

The proposed method provides not only a tool for a rational study of the possible coupling between tappet and his guide, but also a method for building a database to study more precisely the dynamic behavior of valve train. The types of tappets can be compared using a critical appraisal of the constructive schemes based on the knowledge of the equivalent axial stiffness.

The models integrated and validated by the results of experimental tests are a valuable source of data. The results can shed light on various aspects of the dynamics of valve train and provide objective parameters useful for an informed choice on the type of tolerance to be used.

The test machine proposal lends itself to reverse engineering applications, makes it possible to automatically build, in polar coordinates, the profile analyzed.

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