Multi-body numerical simulation of the distribution drive system in an internal combustion engine.

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Abstract

The purpose of the present work is the development of a methodology for the study of the dynamical behaviour of the drive distribution system, based on the techniques of multi-body numerical simulation by AVL/Tycon code. Drive systems by chain with both single cam shaft (SOHC) and double cam shaft (DOHC) are studied, investigating the dynamic behaviour varying the working rate of the engine. The analysis puts in evidence that the chain transmissions introduce high frequency due to engagement cinematic (polygonal effect), with very low amplitude regard to the harmonics of external forces. The methodology permits the evaluation of the loads on the drive shaft, on the distribution shafts, and ones acting on the engine structure trough the supports and constraint points; it represents an important forecast instrument; moreover it permits also the detailed calculation of the friction lost in the interaction among the several components of the chain.

1. Introduction

The distribution systems have a complex dynamic behaviour due to the external forces produced by the irregularity of drive shaft and to the dynamics of the valve train that excite the system priming torsion vibrations in the camshaft and longitudinal vibrations in the chain (or belt). Their complete dynamic analysis, individuating phenomena causing serious damage and critical working rate, has great importance due to the more recent restrictions regard to the effectiveness, the vibrations, the reliability and the acoustic emission [1] [2] [3].

The principal purpose of the present work is the development of a methodology for the study of the dynamical behaviour of the drive distribution system, based on the techniques of numerical simulation, evaluating the effect of the principal design parameters. A drive system by chain with a single cam shaft (SOHC) and double cam shaft (DOHC) are studied [4], investigating the dynamic behaviour varying the working rate of the engine; the conduced analysis refers to full load condition (WOT) and to pre-established working rates; the investigated working field varies from a rate 1500 rpm until the maximum rate 6000 rpm by step 500 rpm.

The developed methodology for the study of the chain dynamics, implemented by the calculation code AVL/Tycon, can be divided in three different phases: at first the analysis of the vibration behaviour of the system is executed, with reference to the torsion vibration of the cam shaft gear and of the chain load, putting in relation the difference between the rotations (angular positioning) of the transmission gears; successively the solicitations acting on the several system components are calculated to the structural verify and the vibro-acoustic behaviour (NVH) of the engine group; at last the power lost in the cam shaft drag and the friction lost in the chain are evaluated, with a detailed analysis of every contribution due to the interaction among the several components [5] [6] [7] [8].

For secrecy reasons all the quantities presented as results of the analysis are normalized with opportune values of the quantities assumed as reference.

2 – Simulation by multi-body calculation code.

The prediction of dynamic behaviour of a distribution drive system is based on a method of multi-body numerical simulation (MBS). The entire system has to be meshed in a finite number of rigid bodies [9]. The degrees of freedom are foreseen by rigid body dynamic, which motion equation is defined by the ordinary Newton-Eulero one that is written in the generic case:

$$M_{body} \cdot \ddot{q}_{body} = \vec{F}^{(e)} + \vec{F}^{*} + \vec{P}^{*} \qquad (1)$$

where M_{body} indicates the mass matrix of the system, q_{body} represents the vector of the generalized coordinates (degree of freedom) consisting in displacements and rotations. The symbols $\vec{F}^{(e)}$ e $\vec{F}^{*}$ at second side represent the external forces (loads on the cam shafts due to valve train) and the forces transmitted by the joints among the rigid bodies. In the case of members with generic motion, an additional term is joined ($\vec{P}^{*}$) to take in account eventual non linear terms of inertia. Assuming that the landmarks of the rigid bodies is their mass centres and the rotational motions are referred to the principal inertia axes, the differential equation system begins uncoupled, so that the solution for every body can be calculated for every degree of freedom [10], [11].

2.1 Motion equation of a chain element.

The formulation of the motion equation of a chain element requires the knowledge of all the forces acting on it, as the interconnection forces between an element and the other and the forces during the contact with the gear and with the guide.

a) Connection between two chain elements.

The connection between two successive elements is implemented by the joint constituted by the coupling pivot-bushing. The force acting on two adjacent chain elements, which action line coincides with the direction joining the centres of the single elements, is characterised by an elastic constant (taking in account both the stiffness in traction of the element and in bending of the pivots) and by an additional damper coefficient that is proportional to the relative speed of the two elements. The expression of this force, for the generic element n is:

$$\vec{F}_{Li,n} = k_{Li} \cdot (s_n - s_{cl}) + c_{Li} \cdot \left(\frac{ds_n}{dt} \right) \qquad (2)$$

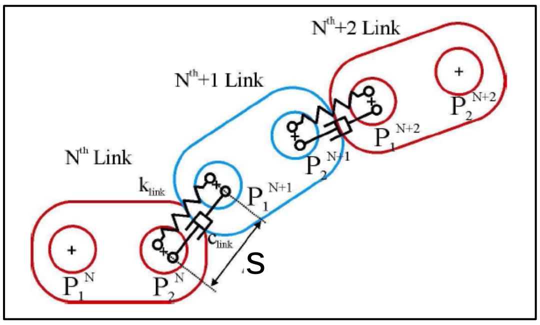


Fig 1 – Scheme of the connection between adjacent chain elements.

Where s_n indicates the relative displacement between the geometrical centre of the pivot of the n element and of the bushing of the element n+1 (fig. 2.1), while s_{cl} represents the clearance of the coupling pivot-bushing.

b) Contact between a chain element and a gear.

Chain element interacts with the gear by the roll; the contact between it (having radius R_R) and the gear can occur on the central arc of its profile (having radius R_1) or on the right or left side of the tooth (having radius R_2). The definition of the contact conditions on the two bodies requires the knowledge of the effective geometry of the chain and of the gear (fig. 2); besides the minimum distance between the roll and the gear profile has to be evaluated.

c) Contact between a chain element and the guide.

Contact between a chain and the guide is schematized in fig. 3. A guide fictitious centre is defined (C_{Gu}) and the distance with the roll centre is evaluated.

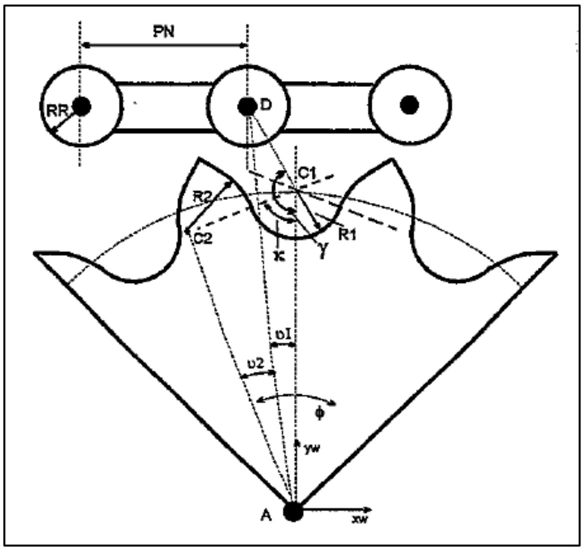


Fig. 2 – Geometric parameters of the profile of a gear for roll chain.

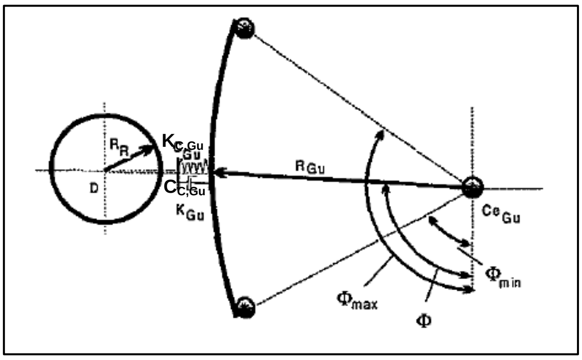


Fig. 3 – Schematization of the contact between a chain element and the guide.

2.2 AVL/Tycon code.

Multi-body modelling and the relative dynamic analysis are implemented by AVL/Tycon code [12]. The entire system is divided in a series of rigid body that interact by connection elements simulating the elastic and dissipative characteristic of the contacts.

All the elements of mass and connections are initially defined as generic elements, in order to be implemented for several applications. Other typologies of elements are available, as the specific elements, to simulate the behaviour of specific organs (as the valves, the cam shaft, the hydraulic tightening), and the macro elements, constituted by a certain number of elements of the same type, to simulate the chains or the toothed belts [17]. Every chain element is schematized as a mass having three degrees of freedom (two to the transfers and one to the rotation), joined to the successive by a spring-damper element (fig. 4), to take in account the friction in the pivot-bushing coupling [13] [23].

Thanks to this multi-dimensional modelling, considering the absolute motion of every chain link and the real cinematic situation, trajectory, velocity and acceleration may be simulated, with the opportunity of calculate the effects of the polygonal action, the behaviour in torsion of the gears and longitudinal and transversal vibration of the chain. Besides, by the assignments of local coefficient of stiffness, damping and friction, the code permits the simulation of the interactions between the various components of the chain, or the contact forces between the single links of the chain, or between the links and the gears and the guides. (fig. 5).

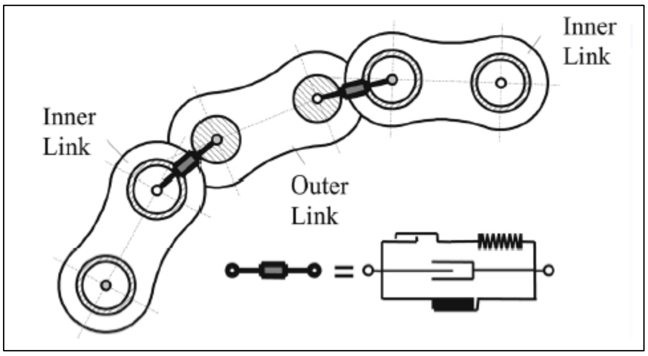


Fig. 4 – Modelling of the connection between the chain elements.

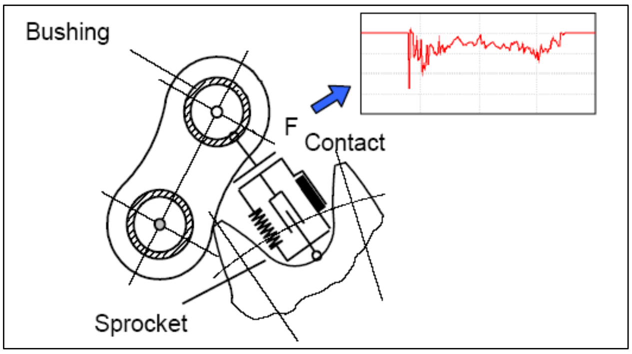


Fig. 5 – Simulation of the contact force between a chain element and a gear.

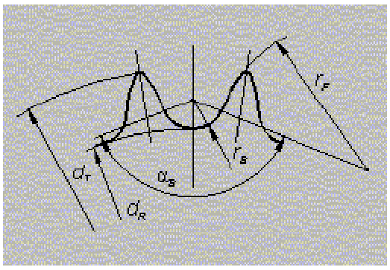
The interface between the chain and the drive shaft, permitting the simulation of the irregularity of the motion due to the gas pressure and the crank inertia, is implemented by a cinematic boundary condition on the angular speed floating versus the time, experimentally measured for every engine rotation rate, or calculated by numerical simulation [14].

The implementation of the transmission model is developed in the following phases:

- Modelling of the entire transmission in study
- Calculation of the static equilibrium and congruence equations
- Dynamic simulation
- Exportation and post processing of data by a graphic interface (Impress Chart).

2.2.1 Definition of the gears profile.

The shape of gear tooth is defined starting from the profile of a space opening delimited by two successive teeth, repeated as much times as the teeth, in circular direction around the system origin. Parameters defining the profile of the opening space (fig. 6) are automatically calculated by Tycon, according to DIN 8196 rule, assigning the pass, the teeth number and the diameter of the bushing.



d_R = primitive diameter

d_T = diameter of the head cutting off

r_B = radius of the central arc

r_F = radius of the tooth side

α_B = contact angle of the bushing (roll)

Fig. 6 – Characteristic parameters of a toothed profile following DIN 8196 rule.

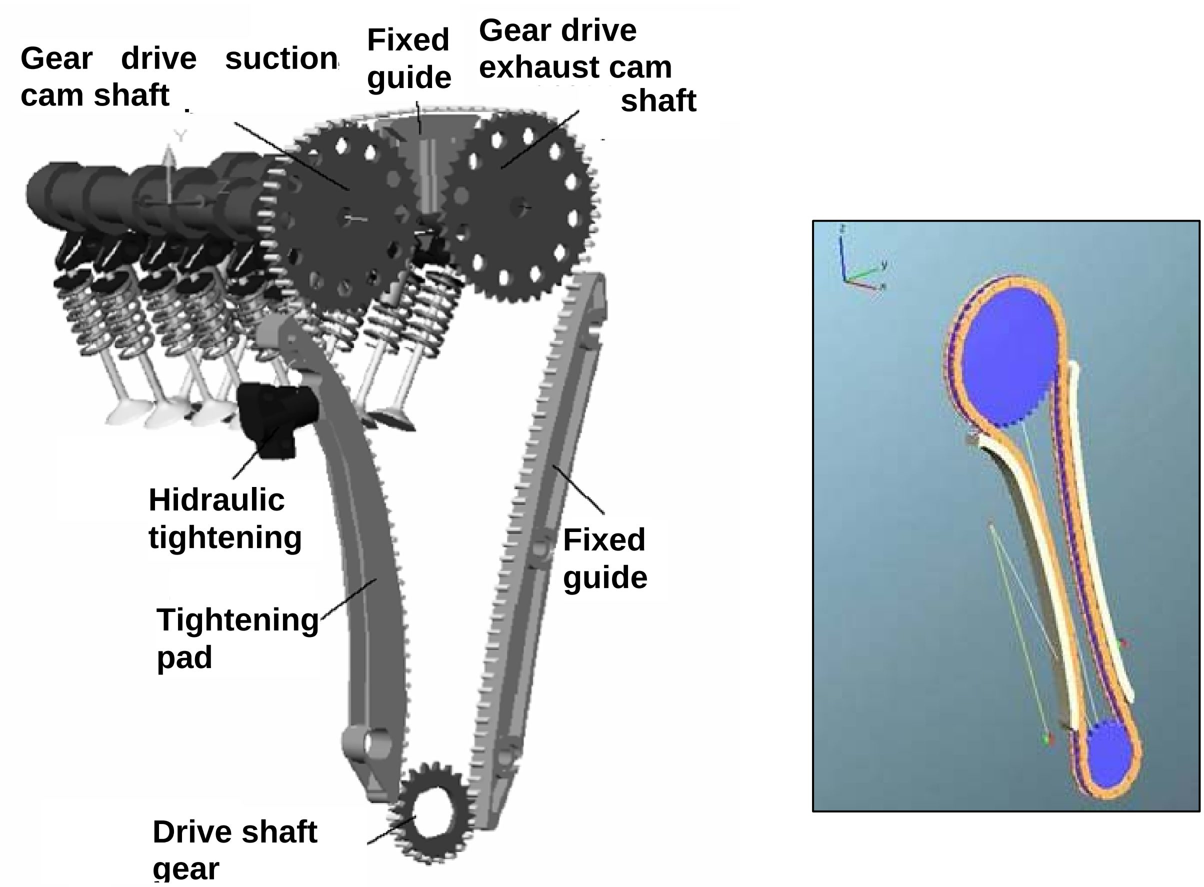


Fig. 7 – Transmission model with two (on the left) and one cam shaft (on the right).

2.2.2 Chain modelling

Used chain characteristics [12] are typical one of a bushing chain according to DIN 8164 rule with a pitch 3/8”. The assignment of the number of the chain elements is also required and the pre-tightening force value. The link between internal and external elements is implemented according to the assigned geometrical data; the connection modelling is completed by the assignment of stiffness, damping and friction coefficients. The last, simulating the friction moment acting between pivot and bushing, permits a more realistic simulation.

The last step in the chain modelling consists in the assignment of the local stiffness and damper coefficient permitting the simulation of contact forces between the chain bushing and the gear teeth, and between the link lateral profile and the guides; these coefficients are assumed using standard value for this type applications.

3 Analysis of the chain dynamic behaviour.

The complete dynamic analysis of the system helps to understand the phenomena causing serious damage, to individuate critical working rates and to analyse the possible modification of the initial design parameters of the transmission. The critic state can regard both the system functionality and the duration. In this paper the dynamic vibration behaviour is explored varying the operative rate of a transmission by roll chain with single cam shaft in the head – SOHC and two cam shafts in the head – DOHC (fig. 7). The conduced analysis refers to full load condition (WOT) and to rotation rates prearranged; the explored working field is comprised between 1500 rpm rate until 6000 rpm maximum rate, with step 500 rpm.

3.1 Torsion vibration of the cam shaft.

The greater interest in the distribution drive system is represented by the torsion answer of the cam shaft [15]. The effective influence of the transmission and of the external forces on the torsion behaviour of the cam shaft has been evaluated; the following working conditions have been simulated:

- analysis in absence of external forces (NO LOAD)
- analysis in presence of the only force due to the valves (VALVE)
- analysis in presence of the only force due to the drive shaft irregularity (IRR)
- analysis in real condition of working (WOT).

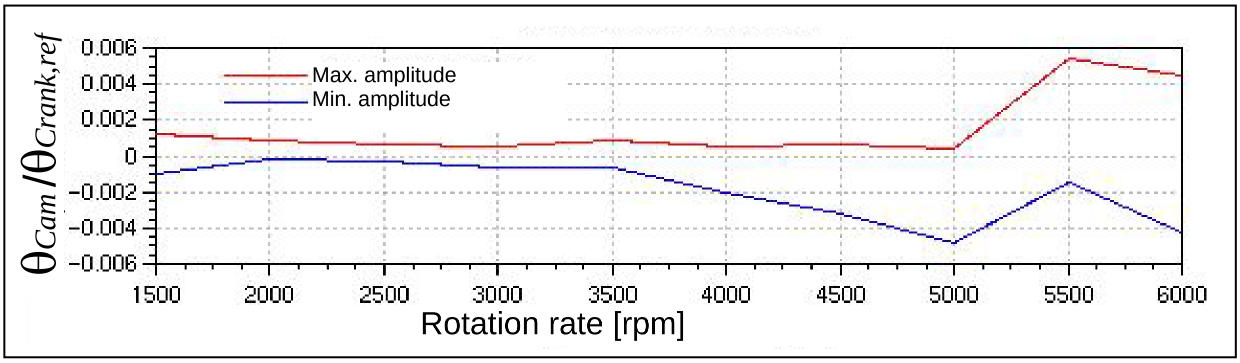


Fig. 8 – Oscillation amplitude of the cam shaft in lack of external force.

3.1.1 Analysis in absence of external forces.

Fig. 8 shows the overall of the oscillation amplitude of the cam shaft versus the rotation rate; the maximum amplitude is sensibly constant to reach a maximum in correspondence of 5500 rpm, while the minimum amplitude decreases to reach a maximum at the same rotation.

Fig. 9 shows the frequency analysis: the global effect of the 21° order due to polygonal action, introducing high frequency, but with small amplitude regard to one induced by the forces; besides a possible resonance of the system can be identified at a value of 300 Hz about, that increases the amplitude at high rates [19].

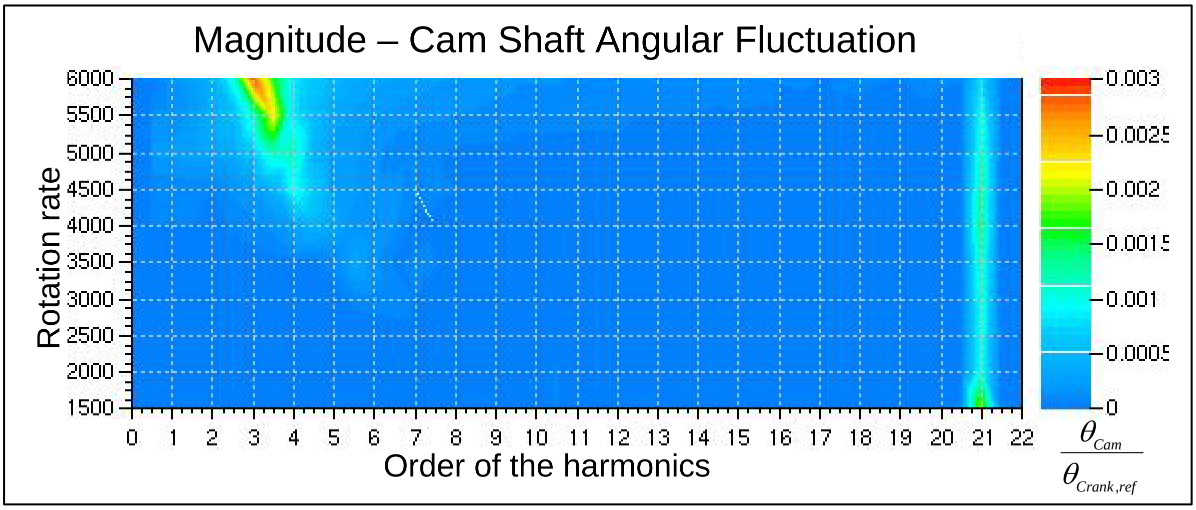


Fig. 9 – Campbell diagram of the torsion behaviour of the cam shaft.

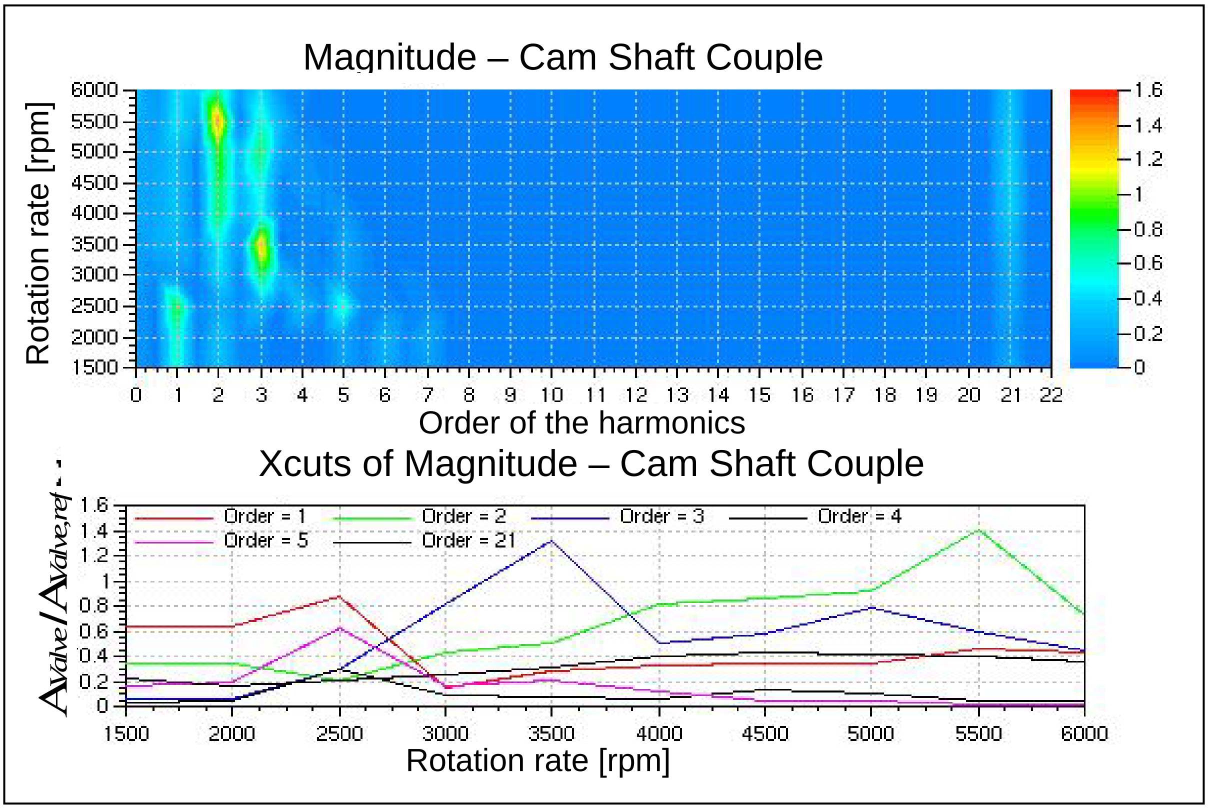


Fig. 10 – Harmonic Analysis of the couple acting on the system.

3.1.2 Forced Analysis 1 (valves).

Successive step is the simulation of the system working in answer to the valve train dynamics, imposing a constant speed. The forcing couple of the valves acts on the cam shaft directly, provoking angular acceleration/deceleration in function of the engine working cycle.

Fig. 10 shows the harmonic analysis of the forcing couple, acting effectively on the system. The calculation is executed subtracting the inertia contribute by the external couple, multiplying the cam shaft inertia by the angular acceleration value.

$$C_{eff.} = C_{est.} - C_{in.} \Rightarrow C_{est.} - I_C \dot{\omega}_C \quad (3)$$

In this case the inertia contribute is very small (however comparable to the effect of external couple of the valves); in the other simulate conditions (IRR and WOT) the irregularity effect of the shaft motion leads high values of the cam shaft angular acceleration, the contribute being predominant regard to the valves couple. Fig. 11 shows the torsion trend of the cam shaft in answer to the effective forcing couple.

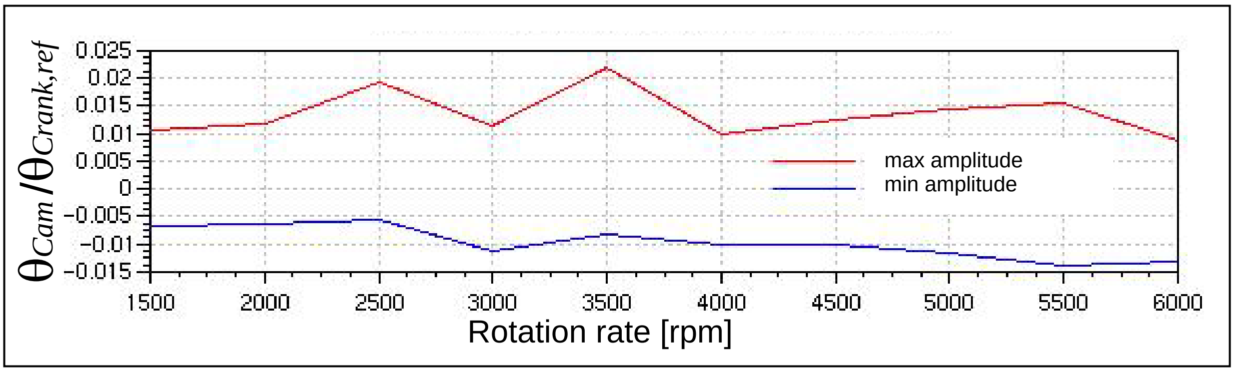


Fig. 11 – Torsion answer (rotation) of the cam shaft in absence of irregularity.

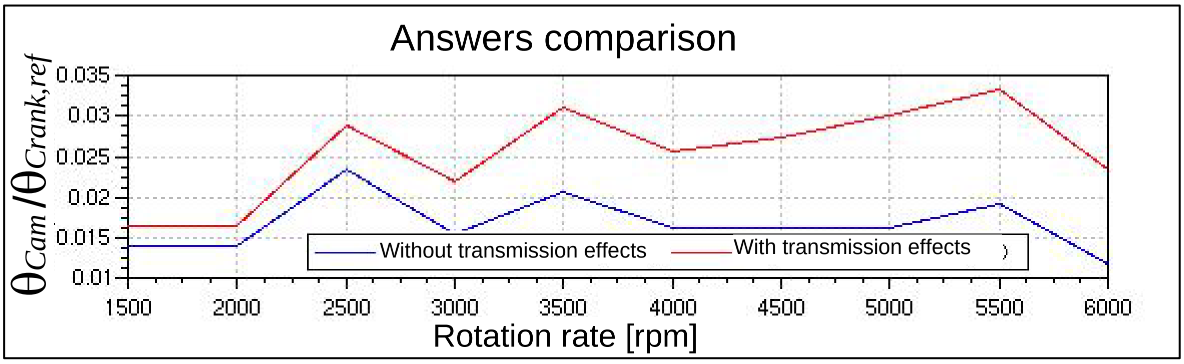


Fig. 12 – Torsion answer of the cam shaft with and without transmission effects.

With the purpose to evaluate the transmission effects on the answer, a comparison between the cam shaft rotation calculated by the code ($\theta_{C,eff.}$), and one that the shaft endures by the external couple without the connection to the system ($\theta_{C,est.}$), is executed. The following calculation procedure is applied: the external couple value is divided by the cam shaft inertia, obtaining the angular acceleration value:

$$\ddot{\theta}_{C,est.}(t) = \frac{C_{est.}(t)}{I_C} \quad (4)$$

This value is integrated assuming one engine cycle as integration interval, obtaining the angular speed fluctuation and, successively, the shaft rotation due to external couple exclusively:

$$\dot{\theta}_{C,est.}(t) = \int_T \ddot{\theta}_{C,est.}(t)dt \qquad \theta_{C,est.}(t) = \int_T \dot{\theta}_{C,est.}(t)dt \tag{5}$$

Fig. 12 shows the comparison between the two curves; the transmission has the effect of increase of the oscillations in the entire working field explored, most to the high rates.

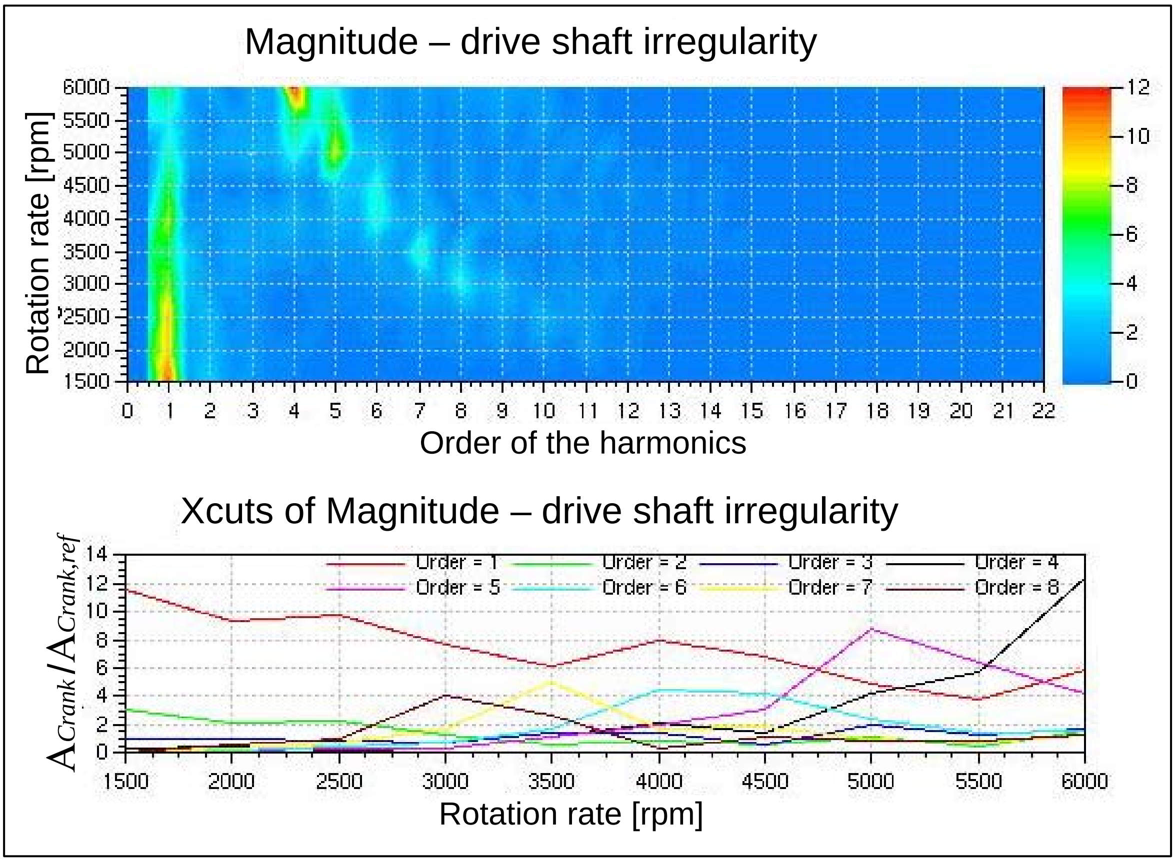


Fig. 13 – Harmonics of drive shaft irregularity (angular speed).

3.1.5 Force analysis 2 (drive shaft irregularity).

Force acting on the drive shaft, resulting by the irregularity of its motion due mainly to the gas pressure in combustion chamber and to the inertia of the crank, is manifested by oscillations of the drive gear that are transmitted to the cam shaft gear [16]. The analysis of the diagrams relative to the angular speed (fig. 13) [18], put in evidence a resonance phenomenon of the force (probably a frequency of own vibration of the shaft in torsion), exciting the system with a frequency constant and equal to 400 Hz.

3.1.5 Forced Analysis in real working condition.

The simulation of the effective working condition is conduced considering both the previous forces acting on the system. In fig. 14 the answers in torsion of the cam shaft for the several working conditions are put in comparison; the green curve represent the force of the valves, the blue one the irregularity of the drive shaft and both the force are represented by the red curve. The effect of the drive shaft irregularity is predominant respect to the couple of the valves, so that the last may be considered negligible.

3.2 Analysis of the behaviour of the tightening pad

Tightening pad of the chain reacts to the traction variation of the slack branch due to the dynamic motion condition. The dynamic behaviour of the tightening system is studied starting from the analysis of the rotation of the guide which the tightening is connected.

The tightening answer does not introduce resonance to the frequency of own oscillation of the pad, resulting by the following expression:

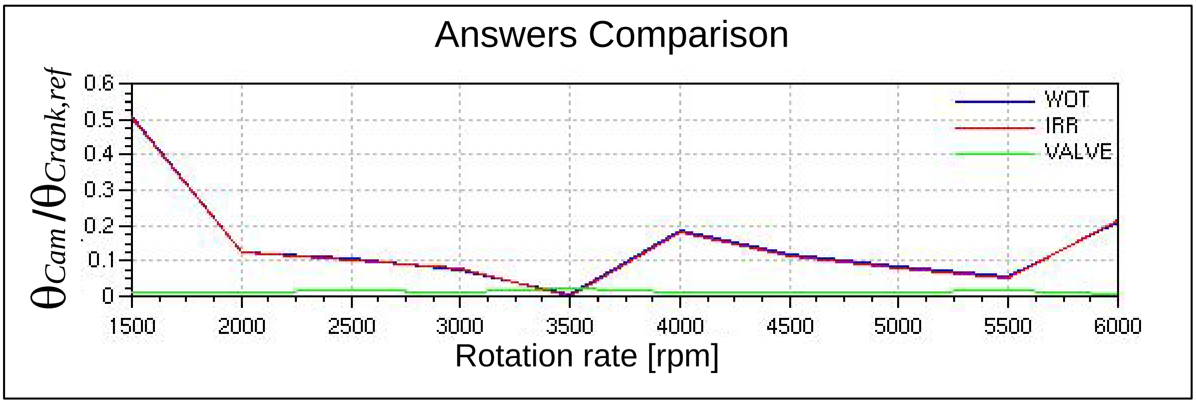


Fig. 14 – Torsion answer of the cam shaft for the several simulate condition.

$$f_{n,Gu} = \frac{\omega_{n,Gu}}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{K_{Tors.}}{I_{Gu}}} \tag{6}$$

where I_{Gu} is the polar inertia moment of the pad respect to its centre of instantaneous rotation and $K_{Tors.}$ the stiffness coefficient in torsion of the tightening, calculated dividing the value of the moment of the tightening force respect to the constraint hinge (fig. 15) by the corresponding rotation:

$$K_{tors.} = \frac{M_T}{\theta_T} \tag{7}$$

Substituting the values of these quantities, the value of the equivalent torsion stiffness of the pad can be obtained in function of the tightening stiffness:

$$K_{tors.} = \frac{F_T \cdot l_T}{x/l_T} = \frac{(K_T x) l_T}{x/l_T} = K_T l_T^2 \tag{8}$$

3.3 Evaluation of the solicitations acting on the transmission components.

The evaluation of the solicitations is very important, not only to verify the structure, but also to evaluate their influence on the vibro acoustic behaviour (NVH) of the propulsion group. The solicitation calculation includes the loads on the drive shaft and on the distribution shaft due to the chain traction, the force altogether acting on the fixed guide, the force on the tightening and the overall force acting on the pad. Table 3.2 reports the values of the barycentre coordinates and the direction of some elements of the studied transmission. The chain direction in correspondence of the points of engagement and disengagement of the pad are not constants, but depends instant by instant on its orientation angle [15]; the small rotations subjected by the guide permit to consider constant the above mentioned quantity.

3.4 Loads Evaluation.

By the temporal history of the loads acting on the shafts, maximum and mean values of the loads on the transmission shafts are calculated. The increases of the chain load in correspondence of the branch in traction are balanced by diminishing of equal entities in the slack branch; in consequence the oscillation of the load on the shafts are joined prevalently to the high frequency of 21° order due to the cinematic of the gear, with maximum amplitude in correspondence of the high rates. The force acting on the tightening is proportional (on the basis of own stiffness) to the rotation that the pad endures due to the tension variation of the slack branch of the chain [19] [20] [21] [22]. This force is expressed by the following analytical formulation:

Table 3.2 – Barycentre Coordinates and chains direction

[	**Coordinates** [m]				**Direzione** [deg]	
	impegno		**disimpegno**		**impegno**	**disimpegno**
asse a camme	x_1=-0.0867	y_1=0.2728	x_2=-0.0138	y_2=0.2918	α_1=31.865	α_2=6.152
guida fissa	x_5=-0.0155	y_5=0.2758	x_6=0.0111	y_6=0.0493	α_5=5.995	α_6=19.104
albero motore	x_7=0.0203	y_7=0.0227	x_8=-0.0263	y_8=0.2087	α_7=19.104	α_8=1.365
pattino tenditore	x_9=-0.0268	y_9=0.0435	x_{10}=-0.0773	y_{10}=0.2374	α_9=2.591	α_{10}=31.865

$$F_T = F_0 + \left[K_T \left(\theta_{Gu} - \theta^0{}_{Gu} \right) + C_T \dot{\theta}_{Gu} \right] \cdot l_T \qquad (9)$$

where F_0 is the value of the initial pre load, K_T e C_T the coefficients of stiffness and camping of the tightening respectively, $\theta^0{}_{Gu}$ the initial orientation of the pad and l_T the distance between the application point of the force and the centre of instantaneous rotation of the guide (fig. 15).

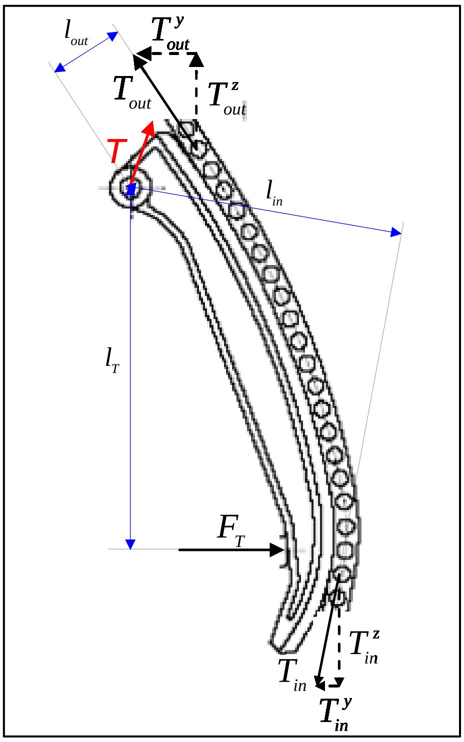


Fig. 15 – Schematic representation of the loads acting on the pad..

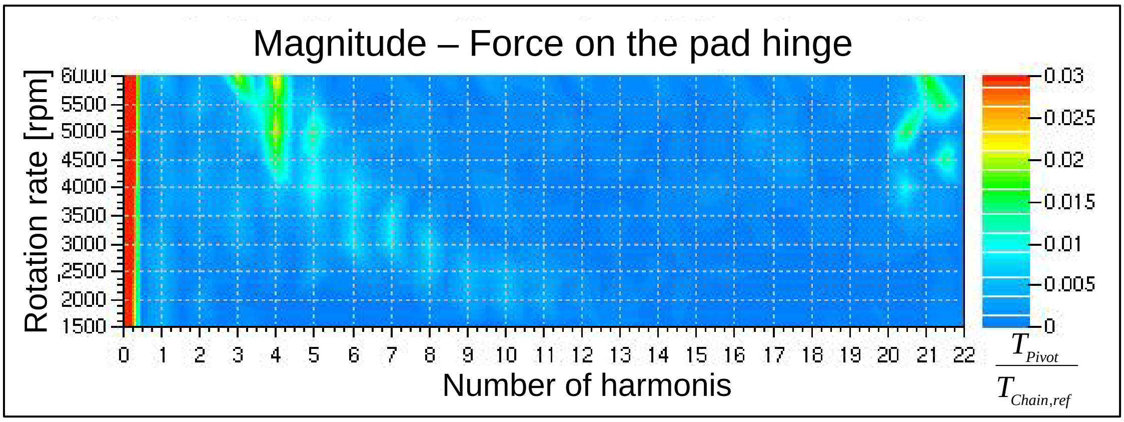


Fig. 16 – Campbell diagram of the force on the pad.

Tightening is modelled constraining the pad in the application point of the force and assigning the characteristic values of stiffness and damping to the relative connection element; the value of this force, that acts on the structure points trough the constraints, is calculated by the code as constraint reaction. The harmonic analysis of the force on the tightening reflects the dynamic behaviour of the chain load and does not highlight resonance phenomena.

The calculation procedure to evaluate the loads acting on the tightening guide is the same than one of the loads acting on the shafts. The difference is that the tightening force acts also on the guide (fig. 15). The components of the above mentioned force have to be summed to the algebraic sum of the force along the directions of the coordinate axes:

$$\begin{cases} T_y = T_{in}^y \pm T_{out}^y \pm \left(F_T \cdot \cos\alpha_T\right) \\ T_z = T_{in}^z \pm T_{out}^z \pm \left(F_T \cdot \sin\alpha_T\right) \end{cases} \tag{10}$$

where α_T represents the angle individuating the work direction of the tightening (in fig. 15 $\alpha_T = 0$).

Modulus and direction of the resulting force acting on the pad individuate the constraint reaction performed by the hinge that is transmitted to the engine structure. The dynamic equilibrium of the pad is guaranteed by its rotational inertia and by the friction moment of the hinge, following the rotation equilibrium equation:

$$I_{Gu} \cdot \ddot{\theta}_{Gu} = M_{Gu} - M_T - M_{Fr,Gu} \tag{11}$$

Where the left side represent the inertia term, the right side the moments of the loads acting on the pad; in particular $M_T = F_T \cdot l_T$ is the load moment on the tightening, $M_{Gu} = (F_{in} \cdot l_{in}) \pm (F_{out} \cdot l_{out})$ the resultant moment of the loads in correspondence of the entrance and exit points and $M_{Fr,Gu}$ the friction moment performed by the hinge (it is neglected). Fig. 16 shows Campbell diagram relative to the resulting force on the pad, that is solicited by the harmonics of the chain load due to polygonal effect. It is not found in the harmonic analysis relative to the guide rotation because this quantity, being calculated integrating the motion equation, is subjected to filtering of the higher harmonics [18]. The solicitation acting on the fixed guide, due to lateral shake-up of the chain, is evaluated adopting analogous calculation methodology. Also in this case the harmonics of the above force follow the dynamic behaviour of the chain load of the strained parts.

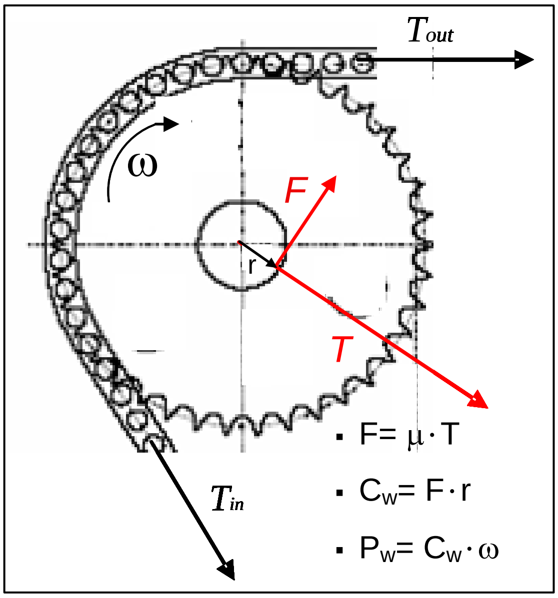


Fig. 17 – Friction lost in the support bearing of the cam shaft due to the chain draught.

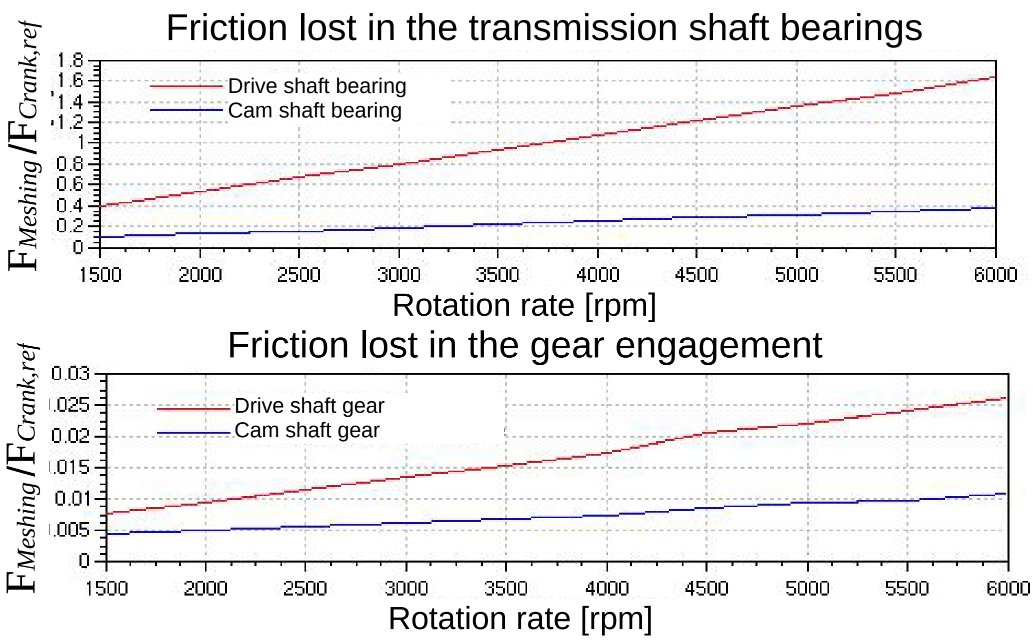


Fig. 18 – Friction mean values in an engine cycle.

3.5 Friction Evaluation.

A notable importance aspect in the motor engineering is constituted by the friction lost to drive the distribution. In the chain systems, the contributions of the friction are constituted by the internal friction of the chain in the articulation joint among the links, by the friction lost to the rubbing of the lateral profile, on the fixed guide and on the mobile pad, in the engagement of the gears and by one lost in the bearings of the drive shaft and of the cam shaft to load the chain.

The friction lost on the guides and in the gear engagement is calculated integrating the contribution of all the chain elements in contact with the considered component. The friction lost in the support bearings of the drive shaft and of the cam shaft are evaluated calculating the loads acting on they, multiplying by the friction coefficient in the couplings (estimated value: μ =0,05) and by the respective bearings radii in correspondence of the contact, multiplying the value of the tough couple(C_w) so obtained by the mean angular speed of the shaft, the contribution of the friction lost in the bearings due to the chain is so obtained (fig. 17).

By the obtained results for every contribution, the mean values are extrapolated in an engine round. Fig. 18 shows the friction lost in the bearings and in the gear engagement. The values increase in linear way versus the working rate, most the friction on the drive shaft bearing, both for the different radii value and of the different rotation speed.

Tab. 3.3 – Mean (normalised) **values of the several contribution of the friction in the chain**

Rotation rate [rpm]	**Chain Internal Friction**	**Friction on the fixed guide**	**Friction on the pad**	**Friction gear – drive shaft**	**Friction gear – cam shaft**	**Friction Bearing – drive shaft**	**Friction Bearing – cam shaft**
1500	0,316	0,049	0,084	0,007	0,004	0,401	0,102
2000	0,322	0,064	0,110	0,009	0,004	0,534	0,124
2500	0,331	0,081	0,139	0,011	0,005	0,668	0,144
3000	0,343	0,095	0,165	0,013	0,006	0,802	0,168
3500	0,356	0,112	0,190	0,015	0,006	0,936	0,191
4000	0,367	0,126	0,210	0,017	0,007	1,073	0,215
4500	0,390	0,145	0,245	0,020	0,008	1,211	0,233
5000	0,406	0,153	0,260	0,021	0,009	1,350	0,264
5500	0,427	0,173	0,288	0,024	0,009	1,487	0,281
6000	0,464	0,186	0,305	0,026	0,011	1,634	0,333

Tab. 3.4 – Maximum (normalised) **values of the solicitations on the chain components.**

Rotation Rate [rpm]	**Chain load**	**Drive shaft load**	**Cam shaft load**	**Load on the tightening**	**Load on the pad**	**Load on the fixed guide**
1500	0,534	0,983	0,875	0,387	0,081	0,232
2000	0,601	0,976	0,878	0,435	0,097	0,261
2500	0,668	0,975	0,884	0,465	0,104	0,289
3000	0,714	0,976	0,897	0,492	0,117	0,308
3500	0,759	0,976	0,900	0,506	0,131	0,330
4000	0,778	0,981	0,901	0,497	0,144	0,340
4500	0,824	0,993	0,922	0,514	0,152	0,355
5000	0,909	1,022	0,940	0,560	0,163	0,394
5500	0,857	1,000	0,947	0,566	0,182	0,374
6000	1	1,079	1,022	0,556	0,185	0,433

Tab 3.3 shows the mean normalised values of the friction; tab. 3.4 shows the maximum values of the loads for the several transmission components, calculated for every working rate.

Moreover the entity of each of these quantities on the friction calculation is evaluated for two working rate (fig. 19 and 20). The friction lost in the engagement of the links on the gears represents a very trifling contribution. The friction lost for the rubbing of the lateral profile of the links on the pads has an effect between 6% and 10% of the total friction (with slightly greater values for the mobile pad); this contributions are smaller than one of the bearings and of the chain, but are not negligible. In the belt command this contribution results concentrated in the friction at the contact between the belt back and the tightening roll.

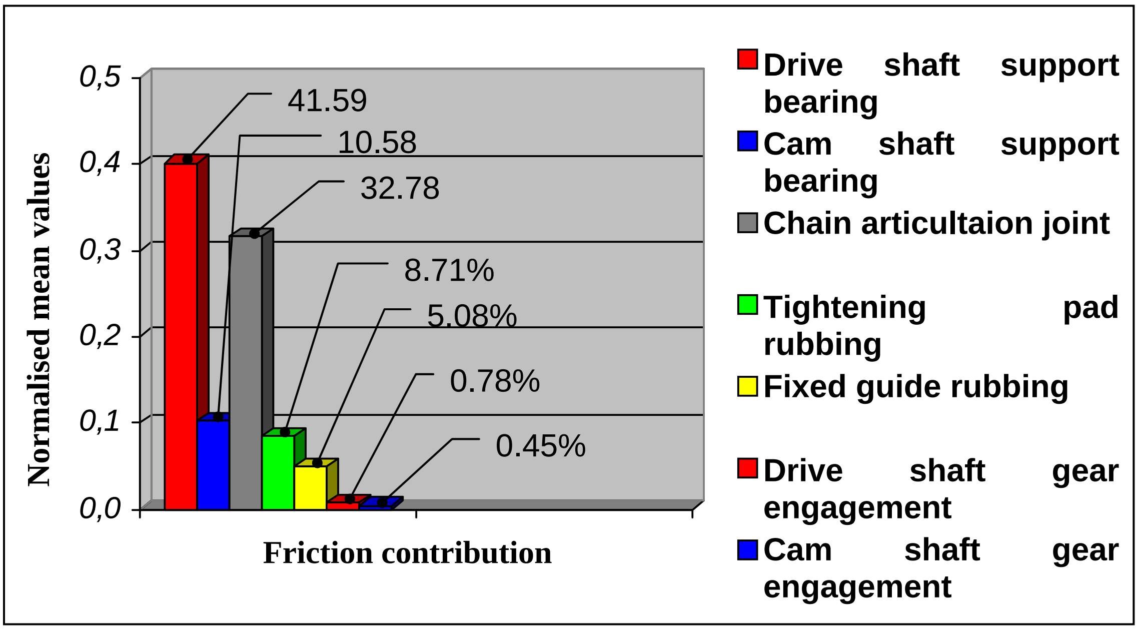


Fig. 19 – Per cent influence of the several contribution on the total friction **(1500 rpm).**

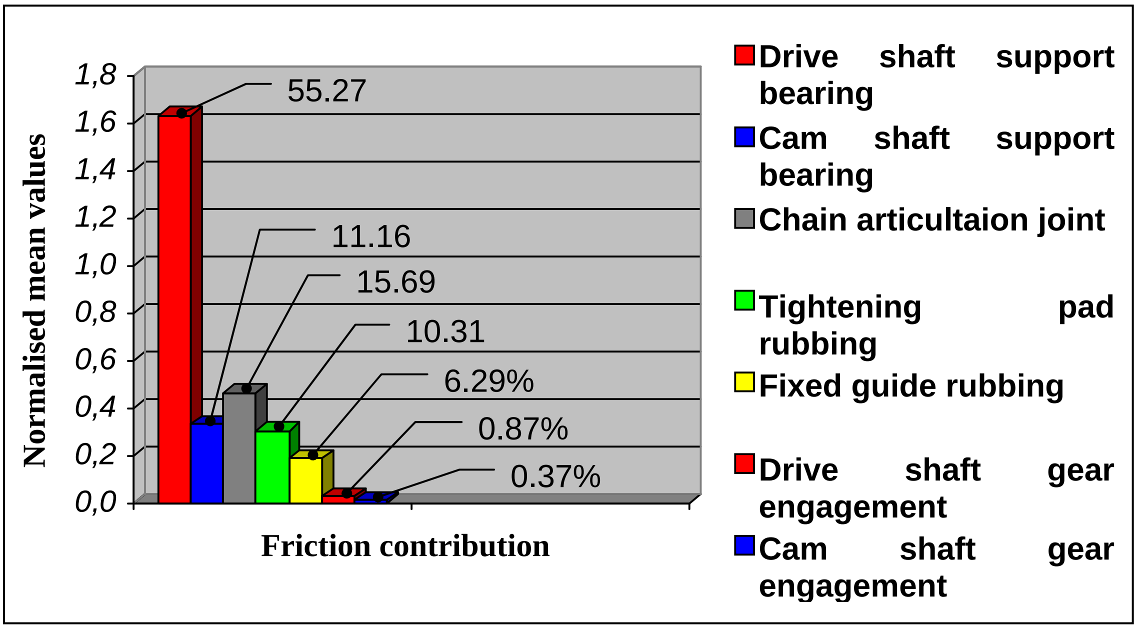


Fig. 20 – Per cent influence of the several contribution on the total friction **(6000 rpm).**

An other very important quantity in terms of engine effectiveness (and of consumption) is represented by the power lost to the cam shaft rotation, which evaluation is executed multiplying the value of the couple absorbed by the drive shaft, calculated multiplying, for every inquired rate, the equivalent peripheral stress (stress in the stressed part minus stress in the slack part by the effective engagement radius R_C) (fig. 21), by the mean value of its angular speed:

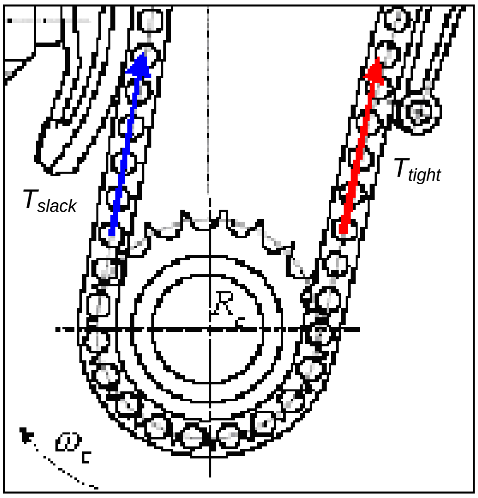


Fig. 21 – Calculation of the couple absorbed by the drive shaft for the cam shaft rotation.

$$C_{rot} = R_C\left(T_{tight} - T_{slack}\right) \qquad\qquad P_{rot} = C_{rot}\,\omega_C \qquad\qquad \textbf{(12)}$$

Fig. 22 shows the trend of the mean values of the power absorbed by the drive shaft in an engine cycle versus the rotation rate; two picks are evidenced in correspondence of 4000 and 6000 rpm respectively.

4 Conclusions.

The analysis of the system dynamic puts in evidence that the chain transmissions introduce high frequency due to engagement cinematic (polygonal effect), with very low amplitude regard to the harmonics of external forces. The conduced study shows that the dynamic behaviour of the distribution drive systems is strongly influenced by the motion irregularity of the drive shaft due to combustion, the effect of the external couple due to valves is negligible.

The developed calculation methodology permits the evaluation of quantity, as the loads on the drive shaft and on the distribution shafts, as the loads acting on the engine structure trough the supports and constraint points of tightening and of the guides, and the friction lost in the chain. It represents an important forecast instrument, in both of a complete analysis of the vibro acoustic behaviour (NVH) and of the entire propellant group, and in terms of engine effectiveness.

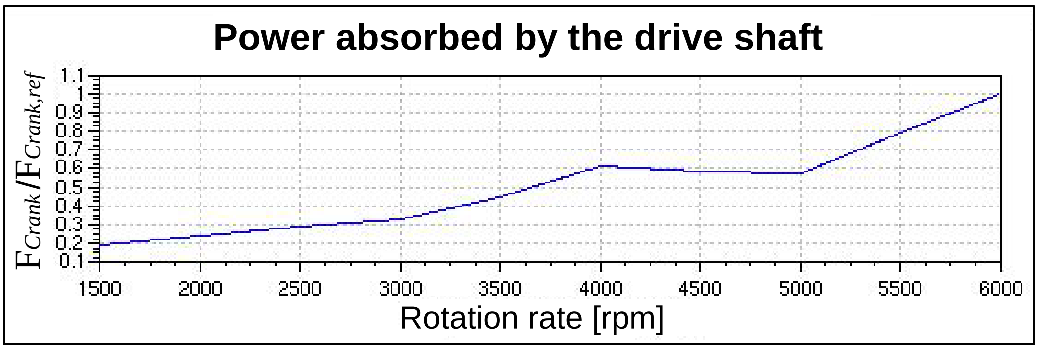


Fig. 22 – Mean power absorbed by the drive shaft varying the working rate.

A resonance phenomenon of the force, due to drive shaft irregularity, is found by the harmonic analysis of the forces, exciting the system by a frequency that is maintained constant in the entire working field, with a value equal to 400 Hz. Besides a resonance in correspondence of the maximum rate 6000 rpm is found by the harmonic analysis of the torsion answer of the cam shaft, that can be trace back to a possible own frequency of the system, with a value equal to 300 Hz. The first harmonic of the forces exciting the own mode of the system is the third; with a very low energetic content. This resonance does not constitute a critical event for the studied system.

Answer of the tightening system follows the dynamic behaviour of the chain load in the slack part and does not influence the dynamic characteristic of the system because the value of the own oscillation frequency, evaluated calculating the equivalent stiffness in torsion of the pad, is maintained superior to the frequency of the system working.

Detailed calculation of the friction lost in the interaction among the several components of the chain evidences that the friction lost in the engagement of the links on the gears represents a very small contribute, while the friction lost for the rubbing of the lateral profile on the pads has an incidence between 6 and 10% of the total friction, with a few greater values for the mobile pad; these contributions are smaller than ones evidenced by the bearings and the chain links, but are not negligible, moreover in the optics of a comparison with the belt drives, where this contribution is concentrated on the contact between the belt back and the tightening roll.

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