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Friction dissipation in reciprocating internal combustion engines: camshaft bearings

F Sordo[1], G Scalzo[1] and M Lavella[1,2]

[1] Politecnico di Torino - DIMEAS Dipartimento di Ingegneria Meccanica e Aerospaziale, Corso Duca degli Abruzzi, 24, 10129 Torino

[2] Universita' degli Studi di Bergamo - DIGIP Dipartimento di Ingegneria Gestionale, dell'Informazione e della Produzione

E-mail: `mario.lavella@unibg.it`

Abstract. The increasing environmental awareness of the last decades has led to a huge tightening of the restrictions on vehicle pollutant and carbon dioxide emissions. Friction dissipation plays a strategic role as its reduction necessarily leads to a decrease in CO_2 emissions. Valve timing system is responsible for a significant part of the overall friction losses and provides a relatively wide margin of improvement. This system regulates intake and exhaust flows in the combustion chamber. The camshaft allows the motion transmission between the crankshaft and the valves. The main goal of this research is the friction dissipation modeling in camshaft bearings of reciprocating internal combustion engines. The proposed model is aimed to investigate possible optimization strategies and to evaluate, with an appropriate precision to the design phase, different constructive solutions of the camshaft from the point of view of friction losses. An empiric model for the evaluation of the Stribeck curve was proposed. It allows to estimate friction coefficient explicitly through a continuous function. Then, this study showed that camshaft friction losses optimization is possible, through the careful adjustment of parameters such as number of bearings, camshaft diameter and bearings dimensions.

1. Introduction

The increasing awareness and concern about environmental problems which characterize these last years have led to the necessity to set targets to contain the environmental damage. Thus, stricter limitations on pollutants and CO_2 emissions are continuously imposed on car manufacturers by competent authorities. Although carbon dioxide is not exactly a pollutant, it represents a major problem as it is a greenhouse gas. Since it is produced in every fossil fuel combustion process, the only way to decrease CO_2 emissions consists in reducing fuel consumption. Maximization of efficiency is, therefore, a fundamental point in the very next development of the automotive field. In this scenario, friction losses assume a strategic role as their minimization necessarily leads to a reduction of fuel consumption, and thus CO_2 production. Furthermore, the solutions aimed to reduce friction can be very cost-effective and many of them can be easily implemented in every type of internal combustion engine.

Most of the overall friction dissipated power is located in the piston ring assembly, in the crankshaft bearings and in the valvetrain. The valve timing system regulates intake and exhaust fluxes in the combustion chamber. It is mainly composed of valves, tappets, camshafts and the transmission drive between crankshaft and camshafts. The valves can be actuated directly, by means of actuators or through a system of pushrods and rockers.

The valvetrain alone is responsible for $10-20\%$ of the overall friction losses in the engine [1] and, considering that it is possible to design this system in many different ways, there is a relatively wide margin of improvement, in terms of friction dissipation. The camshaft allows to transmit the reciprocating motion to the valves. It has two main friction interfaces: cam/tappet and shaft/bearings; this analysis will focus on the latter. Very often, camshaft is designed taking into account economical, functional and construction criteria; energy dissipation is rarely taken into account in a quantitative way. In order to reduce friction losses, there is the need for a systematic and quantitative analysis of the different constructive solutions, to identify the most efficient ones.

This problem can be analyzed in several different ways, analysis of contact interfaces, a posteriori analysis based on models and experimental test rig and preventive models. With regard to the the contact interfaces, researches are focused on the reduction of friction coefficient by means of applications of rolling bearings [2] instead of journal bearings, special lubricants [3–6] or by application of coatings [7–12]. These are very promising because some coatings decrease the friction coefficient and increase wear resistance of the contact interface [7]. Moreover, coatings might have also important implications on fatigue behaviour of the camshaft [8–10]. However, specific studies focused on the timing apparatus [11, 12] show important effects on friction coefficient, trough microtexturing and amorphous carbon coating it is possible to achieve a friction coefficient reduction of over ten up to thirty percent [12]. These approaches, although they provide important improvements in terms of friction coefficient, give no criteria for a design optimization of timing apparatus. Friction dissipation of timing apparatus was also studied through experimental analysis [13–16], mainly in order to validate a posteriori models that require the complete definition of the engine. Consequently, this method must be applied on already existing engines, hence it cannot be used to cost-effectively compare different solutions in the early design phases. Another possible approach to the friction dissipation consists in running multibody simulations [17]; in this case it is possible to get considerably accurate results, but the complexity of the models and the computational requirements make this method expensive and time-consuming, especially if there is the need of analyzing many different solutions. Different analytical models have been proposed too, in the last decades [14, 18–20]. They often describe valvetrain frictional losses accurately, but they may have strict limitations from the design point of view. The main issue is the prediction of friction coefficients. In some cases, the assumptions made to solve it make the model limited to certain operating points (e. g. the assumption of hydrodynamic lubrication regime); on the other hand, some of them may require specific information that is often unknown in the early design stages (e. g. surfaces roughness wavelength). Very often, these models are based on assumptions that limit the optimization possibilities; for instance it is common, in camshaft friction dissipation analyses, to consider the camshaft as a statically determinate beam for the sake of simplicity. Usually, bearings reactions are solved by considering every span as a separate beam supported by just two bearings. This approach may be precise enough for a valvetrain friction dissipation first estimation. However, it does not fit the objectives of optimization that are pursued in this study.

The purpose of this study is to provide a model to analyze the possible different options to reduce camshaft bearings friction losses, and to provide useful criteria to discern them from an energetic point of view with an accuracy that is appropriate to the early design phases. This paper presents a model to describe camshaft bearings dissipation, based on a hyper-constrained beam for the shaft and Amontons-Coulomb friction model. In particular, it proposes an empirical model to predict the Stribeck curve, showing that friction coefficient for a lubricated journal bearing can be accurately predicted through a continuous, hyperbolic function. In this way, all the complexities of the contact between camshaft and bearings are enclosed in the expression of friction coefficient. Then, the most influencing construction parameters for what concerns friction losses were identified. This study shows that, with fixed number of cylinders and engine dimensions, decreasing the number of supports usually leads to more efficient solutions, as well as maximizing support dimensions. Through the calibration of these parameters, it is possible to minimize friction losses and meet the constructive, functional and engine size requirements at the same time. Moreover, in order to achieve a proper optimization, this method should be applied specifically to the engine that have to be analyzed. In fact, by varying dimensional parameters (e. g. bore size, camshaft length, number of cylinders, etc.), the results may change. For instance, this model was used to study the recent trend of engine downsizing by car manufacturers, comparing a 3 cylinder engine with a 4 cylinder one, keeping all engine block dimensions constant. It resulted that, at least from the point of view of camshaft bearings friction losses, reducing the number of cylinders does not provide any benefit.

Nomenclature

c	bearing/camshaft nominal radial clearance
d	camshaft bearings diameter
E_i	Young's modulus of the i-th span
f_i	friction coefficient on the i-th support
$F_{i,j}$	j-th valve force on i-th span
F_0	force generated by the transmission drive
I_i	i-th span moment of inertia
l	bearings length
l_0	distance between the transmission drive
L_i	i-th span length
N	camshaft speed (rps)
p	average pressure on bearings, $p_i = \frac{R_i}{d\,l}$
r	camshaft cross-section radius
R_i	i-th support reaction
$s_{i,j}$	distance between $F_{i,j}$ and the bearing
v_i	number of valves within the i-th span
X_i	unknown moment at the i-th node
α_{M_i}	compliance of the i-th span to the moments
$\alpha^{i}_{F_{i,j}}$	left side compliance of the i-th span to the forces
$\alpha^{i+1}_{F_{i,j}}$	right side compliance of the i-th span to the forces
μ	dynamic viscosity of lubricant
ω	crankshaft speed

2. Modeling methodology

Considering that camshaft speed is half of crankshaft speed, for a four stroke cycle, it is possible to compute the friction torque and, thus, the power dissipated by the journal bearings of one camshaft:

$$P_{loss} = \frac{\omega d}{4} \sum_{i=1}^{n+1} |R_i| f_i \tag{1}$$

where ω is the crankshaft speed, d is the camshaft diameter, n is the number of spans, R_i is the reaction on the i-th support and f_i is the friction coefficient on the i-th support. The friction coefficient is determined through the Stribeck curve (figure 1). This relation was based on two dimensionless number. The first is $f(r/c)$, where f is the coefficient of friction, r is the camshaft cross section radius and c is the radial nominal clearance between shaft and bearing. The second is the Sommerfeld number $(\mu N/p)(r/c)^2$, where μ is the dynamic viscosity of lubricant, N is the camshaft speed (rps) and p is the average pressure on the bearing. The curve is characterized by two main areas, the mixed lubrication at lower Sommerfeld number values and the hydrodynamic lubrication regime, at higher ones. These regions are separated by the minimum friction coefficient. Although it is possible to estimate the coefficient of friction analytically, through already existing models, it often requires high computational power (to solve Reynolds' equation, for instance) and the knowledge of several parameters that can be undefined in the early design phases (especially for the mixed lubrication regime). Therefore, this analysis was based on an experimental Stribeck curve [3] for a common 5W-30 lubricating oil. This allow to obtain simplified models with a precision appropriate to compare the possible different solutions quickly and take conscious design decisions from the early stages. An hyperbolic function was chosen in order to approximate the Stribeck curve. It is continuous in the entire domain and it approximates accurately without any problem due to divergence (unlike polynomial functions). The model is defined as

$$f\left(\frac{r}{c}\right) = \frac{-(a_2 \cdot So + a_5) + \sqrt{(a_2 \cdot So + a_5)^2 - 4 \cdot a_3 \cdot (a_1 \cdot So^2 + a_4 \cdot So - a_6)}}{2a_3} \tag{2}$$

where So is Sommerfeld number and a_1, a_2, a_3, a_4, a_5 and a_6 are fitting parameters. In figure 1 it is illustrated the comparison between the proposed model of the Stribeck curve and the experimental data [3]. The set of fitting parameters used for this analysis is listed in table 1.

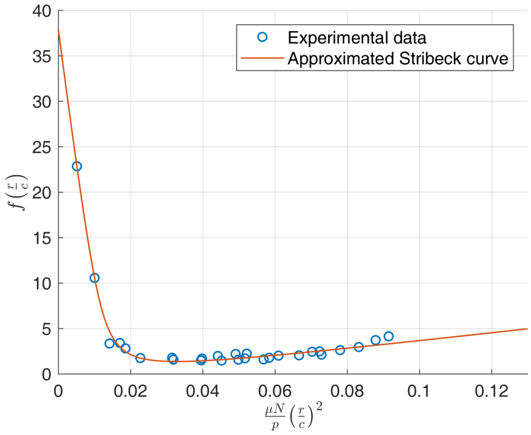


Figure 1. Stribeck curve model comparison with experimental data [3].

Table 1. Fitting parameters.

a_1	a_2	a_3	a_4	a_5	a_6
-3.847394	0.083612	0.000027	0.139919	-0.000947	0.002928

The shaft was modelized as a hyper-constrained beam, supported by $n+1$ bearings, where n is the number of spans (see figure 2a). F_0 represents the force generated by the transmission drive, while $F_{i,j}$ is the force generated by the j-th valve on the i-th span.

The general expression for the reactions can be obtained through forces and moments balancing, on the statically determined schemes in figure 2b and figure 3.

$$R_i = \frac{X_{i-1} - X_i}{L_{i-1}} + \frac{X_{i+1} - X_i}{L_i} - \sum_{j=1}^{v_{i-1}} F_{i-1,j} \frac{s_{i-1,j}}{L_{i-1}} - \sum_{j=1}^{v_i} F_{i,j} \frac{L_i - s_{i,j}}{L_i} \tag{3}$$

where X_i is the unknown moment at i-th node, L_i is the length of i-th span, v_i is the number of valves on i-th span and $s_{i,j}$ is the distance between the force $F_{i,j}$ and the bearing on its left side.

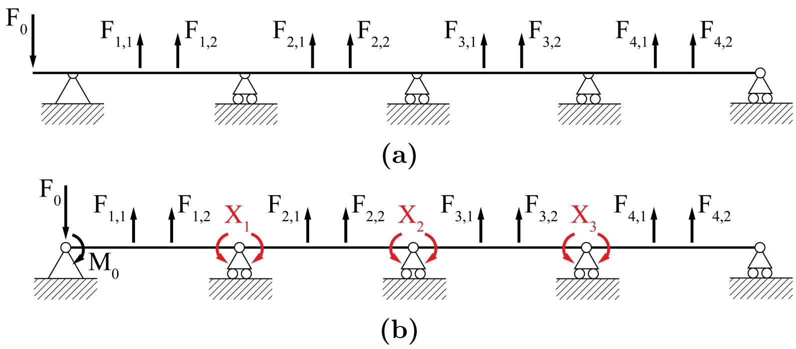


Figure 2. Static models of camshaft: (a) hyper-constrained model of camshaft; (b) statically determined reduced scheme.

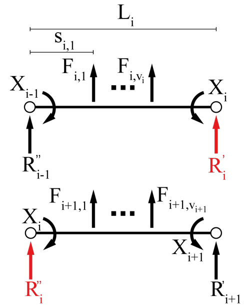


Figure 3. Static general scheme

This general expression describes the reaction of every bearing; specific cases (e.g. the last bearing on the right side) may require to delete some terms. For the closest bearing to the transmission drive, F_0 has to be added.

The last step consists in computing the unknown moments X_i applied to the beam to make it isostatic. This can be made by solving a system of $n-1$ rotation compatibility equations. Every unknown moments, in this case, can be computed through the following equation:

$$X_i = \frac{\left[\sum_{k=1}^{n}\sum_{j=1}^{v_k} F_{k,j}\sum_{t=k}^{k+1}(-1)^{t+i}\alpha^t_{F_{k,j}}\left(\prod_{w=i+1}^{t}\gamma^w_{w-1}\right)\left(\prod_{w=t}^{i-1}\beta^w_{w+1}\right)\right] + (-1)^i F_0 l_0 \alpha_{M_1}\prod_{w=2}^{i-1}\beta^w_{w+1}}{2(\alpha_{M_{i-1}}+\alpha_{M_i}) - \alpha_{M_{i-1}}\beta^{i-1}_i - \alpha_{M_i}\gamma^{i+1}_i} \tag{4}$$

where index k allows to consider the effects of the forces applied on all the n spans, index j allows to sum the effects of all the v_k forces applied to k-th span and t and w are auxiliary indices. α_{M_i} is the compliance of the i-th span to the applied moments, $\alpha^i_{F_{i,j}}$ and $\alpha^{i+1}_{F_{i,j}}$ are the compliances of the left and right side of the i-th span to the applied forces:

$$\alpha_{M_i} = \frac{L_i}{6E_iI_i} \qquad \alpha^i_{F_{i,j}} = s_{i,j}\frac{(2L_i - s_{i,j})(L_i - s_{i,j})}{6E_iI_iL_i} \qquad \alpha^{i+1}_{F_{i,j}} = s_{i,j}\frac{L_i^2 - s_{i,j}^2}{6E_iI_iL_i} \tag{5}$$

β and γ are ratios between spans compliances to the moment. They can be computed recursively, putting $\beta^1_2 = \gamma^{n+1}_n = 0$:

$$\beta^w_{w+1} = \frac{\alpha_{M_w}}{2(\alpha_{M_{w-1}}+\alpha_{M_w}) - \beta^{w-1}_w\alpha_{M_{w-1}}} \qquad w = 2, 3, ..., n-1 \tag{6}$$

$$\gamma^w_{w-1} = \frac{\alpha_{M_{w-1}}}{2(\alpha_{M_{w-1}}+\alpha_{M_w}) - \gamma^{w+1}_w\alpha_{M_w}} \qquad w = 3, 4, ..., n \tag{7}$$

3. Results

Results were focused essentially on a typical small 4 cylinders spark ignition engine. Most of the design parameters (bore diameter, camshaft length, bearings length, etc.) refer to a 1000 cc spark ignition engine with a double overhead camshaft timing system (two intake and two exhaust valves per cylinder), belt driven. The average dissipated power refers to one camshaft, on one cycle. In table 2 the main engine parameters are listed. Valves positions were determined with geometrical considerations. It was assumed that valves diameter is the maximum possible in one fourth of the cylinder (with the proper clearances). With regard to the forces generated by the valves, valves inertia has been neglected. Thus, the valve force profile has been approximated as half a sinusoidal function, whose amplitude is proportional to the spring stiffness. Due to the spring preload, this profile exhibits discontinuities, as shown in figure 4. For this reason, these specific results will be more accurate for slow engines (below 2000 rpm). However, this assumption does not affect the validity of the presented method; moreover, this analysis will be integrated with the study of the cam/tappet interface, allowing to increase the accuracy of the results even at higher speeds.

Table 2. The design parameters used for the simulations.

Parameter	Value	Parameter	Value
Bore diameter	70 mm	Spring preload	4 mm
Distance between cylinders (A)	8 mm	Valve spring stiffness	35000 N/m
Camshaft length	349.5 mm	Transmission drive force	560 N
l_0	35 mm	Cylinders ignition order	1, 3, 4, 2
Camshaft diameter	30 mm	IVO	25°
Bearing length	20 mm	IVC	60°
Bearing/camshaft radial clearance	0.061 mm	EVO	60°
Lubricant dynamic viscosity	9 mPa·s	EVC	25°
Cam height	7 mm		

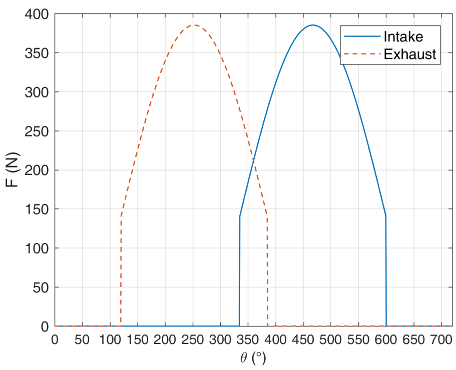


Figure 4. Valves springs forces over crankshaft angle.

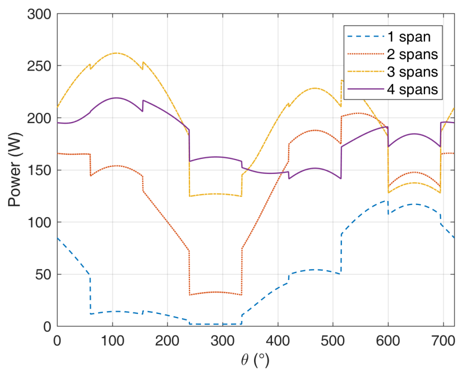


Figure 5. Instantaneous power friction dissipation of a of 4 cylinder engine.

The viscosity chosen for these results ($\mu = 9$ mPa·s) refers to a common 5W-30 lubricating oil at 100°C. Figure 6 shows average dissipated power on one cycle and one camshaft with 4 spans assuming different values of dynamic viscosity of the lubricant. Results show that by increasing the value of lubricant viscosity it is possible to shift the operating points of the bearings towards the hydrodynamic regime, thus decreasing dissipated power. For lower values of viscosity (μ) friction losses increase monotonically with crankshaft speed, as most of the operating points of the bearings are shifted in the mixed friction regime. For higher values of dynamic viscosity, on the other hand, operating points become located around the minimum of the Stribeck curve, as dissipated power firstly increase and then decrease by increasing crankshaft speed. As dynamic viscosity is strongly sensitive to temperature variations, friction losses may change depending on environmental and operating conditions.

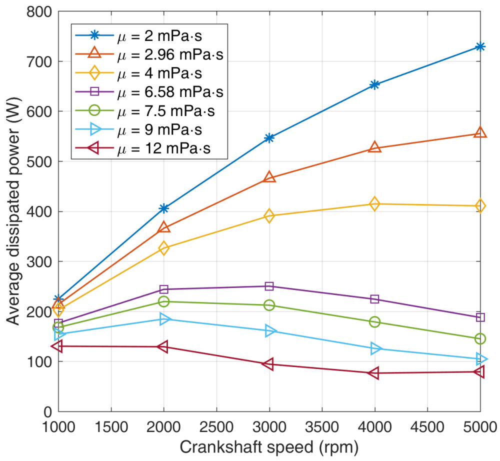


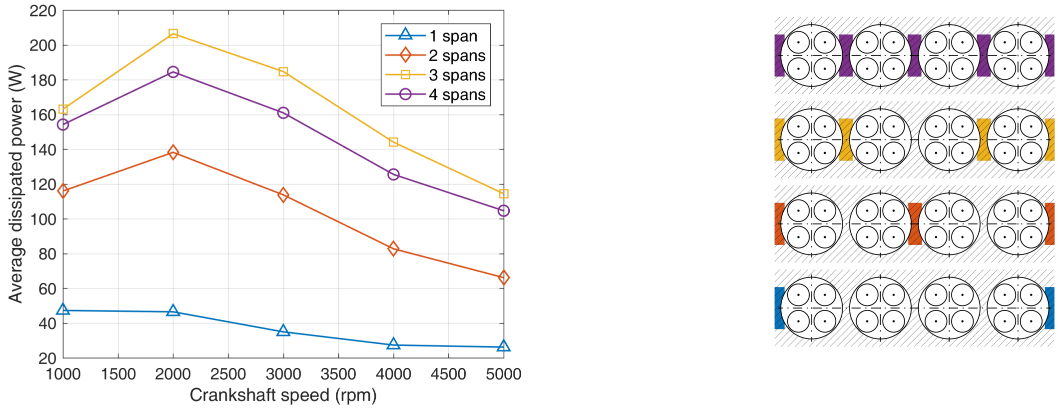
Figure 6. Average dissipated power on one cycle for a 4 spans camshaft at different speeds, assuming different values of lubricant viscosity.

3.1. Case 1: number of spans

The first case study focuses on the number of spans. Instantaneous friction dissipated power of a of 4 cylinder engine at 1500 rpm, in the cases from 1 to 4 spans is shown in figure 5. Instantaneous dissipated power is clearly unsteady, therefore average power was computed in order to compare different constructive solutions. Figure 7a illustrates the average dissipated power on one cycle for one camshaft for the same engine, assuming a number of spans from one to four (one per cylinder, see figure 7b). Friction dissipation increases with the number of spans up to 3, four spans exhibit a slight lower dissipation than 3. Usually, camshaft bearings are located between the cylinders; thus, the three spans camshaft was modelized with one span for the first cylinder, one for the second and third and one for the fourth (in order to have a symmetric configuration). Losses are higher at slower speeds, as for these operating points the bearings are more likely to work in mixed friction regime (figure 7a).

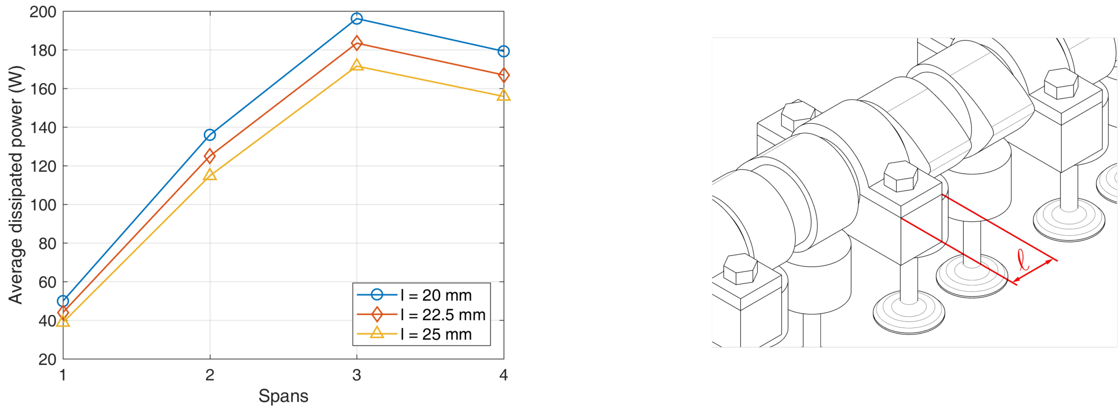
3.2. Case 2: bearing length

Bearing length is another interesting parameter, as its increase in length decreases the average pressure on the support. Thus, operating condition shifts over the Stribeck curve. As a consequence, friction coefficients decrease in all the bearings operating in mixed friction. Since in the mixed friction part of the curve friction coefficient decreases rapidly, mean friction coefficient is likely to decrease. Figure 8a shows average dissipated power on one cycle for one camshaft at 1500 rpm, for different values of bearings length, 1 to 4 spans solutions. In this case, friction losses decrease when bearing axial dimensions increase. Figure 8b shows the definition of bearing length.



(a) (b)

Figure 7. Influence of the number of bearings on average dissipated power: (a) average dissipated power on one cycle for one camshaft at different speeds, 1 to 4 spans cases; (b) position of bearings in the 4 cases analyzed in case 1.



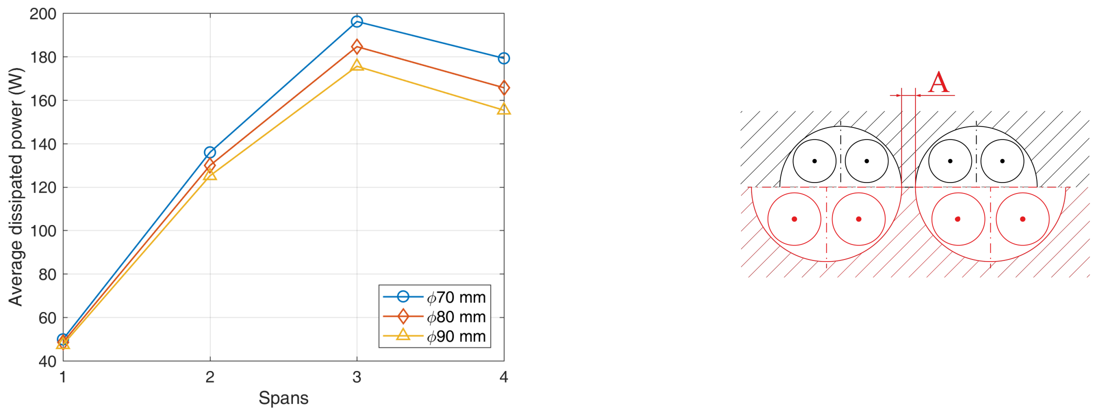
(a) (b)

Figure 8. Effect of bearings length on friction losses: (a) average dissipated power on one cycle for one camshaft for different bearings lengths at 1500 rpm, 1 to 4 spans cases; (b) Definition of bearing length.

3.3. Case 3: bore sizes

This analysis was aimed to investigate the effect of bore diameter on friction losses, to see if it is more convenient to design an engine with bigger or smaller cylinders. While the minimum distance between the lateral surfaces of the cylinders (A in figure 9b) was kept constant, valves dimensions and positions have been changed together with bore sizes, following the geometrical considerations previously exposed. These variations in engine parameters are shown in figure 9b. Figure 9a shows average dissipated power at 1500 rpm for different bore sizes, for 1 to 4 spans camshafts.

The results show that a solution with larger cylinders appears to be more convenient, regardless of the number of spans. Furthermore, the differences in friction losses between engines with different bore sizes increases when the number of bearings increases.

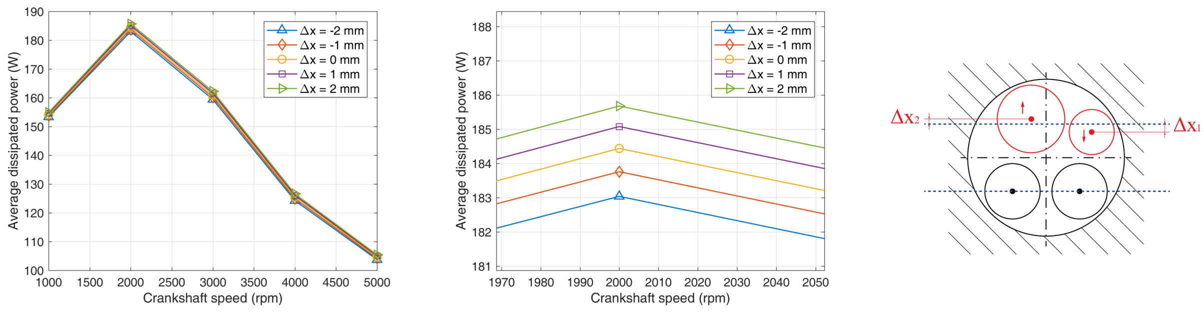


(a) (b)

Figure 9. Effect of bore size on friction losses: (a) average dissipated power on one cycle for one camshaft for different bore sizes at 1500 rpm, 1 to 4 spans cases; (b) example of how the engine block dimensions changed in the compared cases.

3.4. Case 4: valves position

In this case study the effects of valves position was investigated. As mentioned, valves diameters have been determined assuming the maximum diameter in one quarter of the cylinder. With this assumption, the position of the force generated by the valves is necessarily fixed. However, in order to maximize intake flow, exhaust valves are usually smaller than intake ones. Consequently, their position can be slightly moved inwards or outwards. Figure 10c shows the shifting of an exhaust valve, to maximize intake valve dimensions. Therefore a simulation has been run, shifting the valve stem position outwards or inwards ($\pm$ 2 mm). Figure 10a shows a slightly

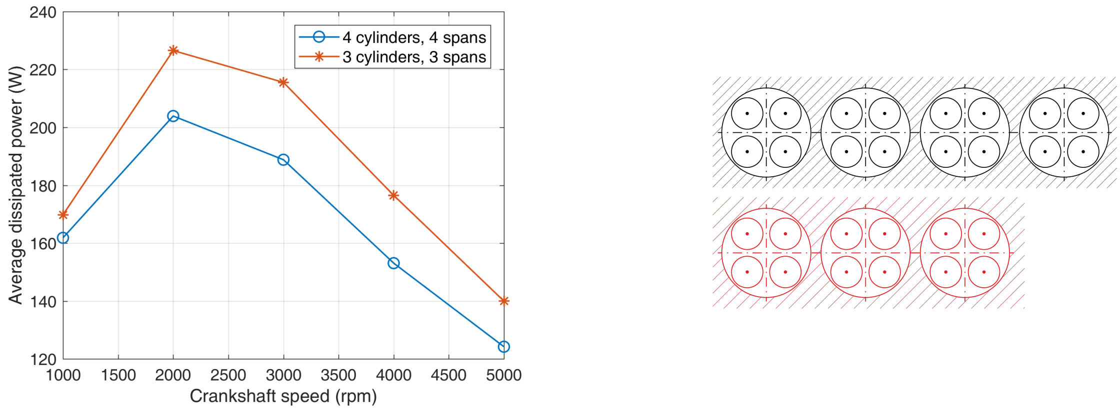


(a) (b) (c)

Figure 10. Effect of valve shifting on friction losses: (a) average dissipated power on one cycle for one camshaft at different speeds, ± 2 mm valves position shift, 4 spans camshaft; (b) zoom on figure 10b; (c) example of valve shifting.

beneficial effect as a consequence of a shift of valves position. Positive values of Δx represent inward shifts, negative ones represent outward shifts. Substantially there is no important difference in terms of average dissipated power if the valves are shifted inwards of 1 - 2 mm.

3.5. Case 5: 4 cylinders vs 3 cylinders
This analysis is focused on the comparison between a 4 cylinder and a 3 cylinder engine, assuming one span per cylinder. All the engine dimensions (bore diameter, distance between cylinders, etc.) were assumed constant. Indeed, car manufacturers has, recently, started to replace 4 cylinders engines with smaller 3 cylinders ones, as they are assumed to be more fuel efficient. The schemes of the the two engines compared in this analysis are illustrated in figure 11b. Figure 11a



(a) **(b)**

Figure 11. Comparison between a 4 cylinders engine and the corresponding 3 cylinders version: (a) average dissipated power on one cycle for one camshaft at different speeds, comparison between 4 cylinders, 4 spans and 3 cylinders, 3 spans; (b) scheme of the engine blocks.

illustrates the average power dissipated during one cycle. With regards to the friction dissipation of the camshaft, between 3 or 4 cylinder with a span per cylinder the 4 cylinder engine appears to be less energy consuming, at every crankshaft speed.

4. Discussion
Results shows that the number of camshaft supports can be optimized in order to reduce friction dissipation. With regards to a four cylinder engine, the three spans constructive solution should be avoided, since its losses are similar to the 4 spans ones (see figure 7a), generating larger and more asymmetrical displacements along the camshaft. Excluding this constructive solution, the relationship reported in figure 7a illustrates that friction dissipation increases when the number of camshaft supports increases. This result might not have a practical application because effects of shaft deflection and relative vibration were neglected. However this problem decreases with the length of spans and with the increase of the shaft diameter within the span. Moreover, the friction dissipation shows two type of dependencies on bearing diameter. On one hand, it increases if the diameter of bearing increases as a consequence of equation 1. On the other hand, friction dissipation as a function of the bearing diameter firstly decrease and after increase as a consequence of Sommerfeld number. The same dependency exists between friction dissipation and bearing length. In the analyzed example, increasing the bearings dimensions has positive consequences on losses (figure 8a): it reduces the average pressure on the bearing, thus shifting the operating points on the Stribeck curve towards the hydrodynamic regime. However, this effect depends on other aspects, such as the engine speed and transmission drive loads, hence the same solution could lead to a different trend for another engine. As a consequence, constructive solutions could be optimized using the number of spans, the diameter of the camshaft within the spans and the diameter and length of bearings as parameters.

The advantages linked to valve shifting (figure 10a) are far too little to justify any specific design choice. For this reason, valves position should be determined in order to minimize other types of dissipation and optimize the combustion chamber. For instance, on one hand, moving the exhaust valves inwards could result into a minimal increase in bearings friction losses. On the other hand, this could allow to maximize the intake valves diameter, hence more significantly reducing fluid dynamic losses. During the optimization of the combustion chamber, it should also be taken into account that camshaft bearings friction losses decrease when bore size increases (figure 9a). This solution could result into more space between the cams, allowing to use larger and less consuming bearings too. However, this choice clearly have significant consequences on the engine. For these reasons, valves position and bore size were not considered in the optimization criterion mentioned above.

In the last few years, car manufacturers have more and more often been downsizing their engines, passing from 4 cylinders to 3 cylinders ones. This may be convenient in terms of costs and production, and may have led to the general idea that a decreasing in the number of cylinders necessarily implies a reduction of friction losses. In order to analyze this trend, engine dimensions are kept constant for the comparison between 4 cylinders and 3 cylinders engines. Indeed, it has been assumed that the reasonably easiest and cheapest way to design a 3 cylinder engine block consists in "removing" a cylinder from an already designed and tested 4 cylinder engine. The results showed in figure 11a demonstrate that, at least from the point of view of friction losses in camshaft bearings, the 3 cylinders engine is not better than the 4 cylinders version. There are no advantages in terms of dissipated energy (it is actually slightly higher with just 3 cylinders, according to this model) and, conversely, there may be several drawbacks linked to a higher phase shift between ignitions.

5. Conclusions

An analytical model that allow to optimize camshaft friction losses was developed. It is appropriate to the design process since the early stages, as it can provide results quickly and can be easily integrated with other more or less complex models. It is aimed to support the design of the valve timing system, by providing criteria that allow to evaluate the possible options from an energetic point of view. By knowing some points of the Stribeck curve for a journal bearing (they can be determined experimentally or taken from the existing literature) and the dimensional parameters of the engine it is possible to immediately compare different design choices. Through the integration with more accurate models describing other dissipative phenomena, such as cam/tappet friction and transmission drive losses, it would be possible to adapt this method to every case study with the proper precision.

Indeed, the introduction of other parameters could change the results, as it is not easy to tell in advance which solution is more efficient. For example, the comparison between the 4 cylinders engine and the corresponding 3 cylinder version showed that, against the recent trend of engines downsizing, the 3 cylinders engine does not necessarily represent a more efficient solution from every point of view.

Valves shifting and the variation of bore diameter are constrained by the optimization of the combustion chamber. Therefore, they may not be used as optimization parameters.

Anyway, the examples analyzed in this study shows that camshaft bearings friction losses optimization is possible, and that careful design choices allow to significantly reduce energy dissipation and thus carbon dioxide emissions.

Every optimization process should start from an accurate Stribeck curve. As the curve is characterized by a minimum, between mixed friction and hydrodynamic lubrication regimes, it is the key point for a proper reduction of friction losses. An empirical model for the Stribeck curve was proposed; it resulted that the friction coefficient can be accurately predicted through an hyperbolic function, that can be computed through experimental data interpolation and provide an explicit expression for friction coefficient.

Subsequently, friction losses minimization can be achieved through the calibration of those parameters which positively affect energy efficiency without excessively influencing other systems. A reduction of the number of spans can lead to a significant decrease in friction losses. The consequent increase of camshaft deflections and vibrations can be compensated adopting a camshaft with a bigger moment of inertia. Moreover, by increasing journal bearings dimensions, the operating points of the supports can be shifted along the Stribeck curve, towards the hydrodynamic regime. By properly adjusting bearings length and diameter, it is possible to move most of the operating points around the minimum of the curve and comply with the engine size limits at the same time.

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