inertial members within the IC engine. This corresponds to the frictional (mechanical) losses resulting from the transmission, piston assembly, auxiliaries, journal and big end bearings, and valve train during operation. Therefore, only the 12 % of the total energy was used to move the vehicle and overcome the rolling and the air resistance.

Since the heat losses have the lion part, in the energy losses distribution of vehicle system, a slightly reduction of the frictional losses leads to a no-negligible increase of the organic efficiency of the engine as showed by Parker and Adams [4]. This can be elaborated by understanding that the frictional losses consumed same energy and transform another part in heat which the cooling system have to take off and it contributing in addition to the energy losses due to the thermal effects. Under the influence of cycling loaded and dynamic stresses, the wear rate of the inertial members is exacerbated, thus resulting in an overall inertial imbalance in the system with an increased effect in engine NVH (Noise, Vibration and Harshness) as stated by Eichhorn [5] and Bell [6].

In particular, as highlighted by Calabretta and Cacciatore [7], the valvetrain friction makes a significant contribution to the whole engine friction loss, especially at low engine speed. The total amount depends on the valvetrain type, engine architecture, engine speed and lubricant temperature among many other factors. As shown by Koch [8] in a graph, derived from motored strip test data, shows the valvetrain for a modern spark ignition car engine contributing about 35% to total friction at 1000 rpm and about 10 % at 6000 rpm. This type of test typically includes the timing drive as a part of the valvetrain.

One of the most recognised ways to increase the engine performance output is increase the volumetric efficiency of the engine or of its one cylinder. The volumetric efficiency, in term of the amount of air intake in the engine and the exhaust gases that goes out, depend solely on the opening and closing timing of the inlet and exhaust valve, if it keeps constant the engine breathing capacity index β. The IC engine comprises many articulating mechanical elements, but the one particular assembly fully responsible for elevating the volumetric efficiency to simply enhance the engine performance of the IC engine is the valvetrain itself as shown by Morel and Flemming [9].

Valvetrain assemblies constitute a series of inertial members, held together by some form of constrains expect for the most important component: the cam and the roller. The cam is mounted on the camshaft and form a “high pair contact” with the roller. This required that the cam should remain always in contact with the roller, unless a loss of lubricant film may ensue as highlighted by Morel and Flemming [9].

During operation, the cam profile undergoes a Hertzian deformation in the contact zone producing an elliptical pressure build up. Due to the high pressure developed between the cam profile and the roller the lubrication of this contact is elasto - hydrodynamic – EHD - . At high pressure, the viscosity of the oil increases exponentially with pressure and the oil film may be maintained between the roller and the cam. Many researchers have studied EHD lubrication and the work of Dowson [10] is classical in this field. The basic assumption of EHD is that the pressure distribution along the contact patch should satisfy both the Reynolds equations for the oil film and the Hertz’s equations for the deformation of the mating parts. Extensive studies of Dowson and Higginson [10] who devised a numerical iterative method to adapt the pressure distribution to match the contour of the oil film and the bodies deflection have shown the following basic facts: the oil film is practically parallel along the contact zone; at oil exit the flow is restricted and the oil film reaches its minimum thickness; the pressure curve is almost Hertzian and the additions are negligible.

As explained before, the most loaded contact in the valve train mechanism is the cam – tappet conjunction. The sliding nature of this lubricated conjunction, together with a highly transient loading regime, renders this contact a major cause of valve train mechanical inefficiency. Without the protective effect of a lubricant film formed in the contact, the applied load together with vibrations can cause wear and scuffing of contacting surfaces. A multi-physics model is reported by Teodorescu and Kushwaha [11], which incorporates all the aforementioned physical phenomena.

The concentrated contact conjunction between cam and follower is exposed to one of the most severe tribological conditions within the internal combustion engine. Because of high loads and small contact area, contact pressure is high and the elastic deformations are often several orders of magnitude higher than the minimum film thickness as highlighted by Beloiu [12]. In addition, at such high contact pressures, the viscosity of lubricant can be ten orders of magnitude higher than the viscosity at atmospheric pressure as shown by Hoglund [13]. The mineral oil loses its liquid character and experiences transition to glassy state – semisolid – as proof experimentally. In this case the lubricant becomes non-Newtonian. Although the Newtonian assumption of the lubricant is sufficient to predict film thickness, it cannot predict accurately friction which is usually one order of magnitude higher in measurements as shown by Johnson [14] and Dowson [15].

As shown by Calabretta and Cacciatore [7], at the cam - tappet contact the power loss is a function of contact load, sliding velocity and friction coefficient. The contact load is a function of spring force, inertia force and vibration. The sliding velocity is determined by the cam profile, the geometry of the mechanism and the engine speed. The coefficient of friction is a complex variable dependent on contact force, sliding speed, lubricant film thickness, lubricant viscosity, lubricant temperature, surface texture, materials combination, etc. The regime of lubrication at the cam - tappet contact varies rapidly during the cycle from hydrodynamic on the cam flanks at high engine speed to boundary lubrication on the nose at low speed.

To enable optimization, as stated by Calabretta and Cacciatore [7], of the valvetrain and component design it is necessary to able to predict the various contributions to valve train friction at the design stage so that friction level at low speed – and effect on fuel consumption – can be weighed against dynamic performance – and affect on engine power – at high speed.

The regime of lubrication is function of load: in case of low loaded contact, the regime of lubrication is mixed and, in particular, there is a hydrodynamic regime plus boundary friction condition. In the other hand, the regime of lubrication of a high loaded contact is elastohydrodynamic or at last, mixed, where over the elastohydrodynamic regime, there is the boundary friction condition as well as described by Guangteng [16].

Many workers have attempted to model the elastohydrodynamic lubrication behaviour of rough surface, first analytically, e.g. by Tallian [17], and Patir [18], and more recently using numerical methods as exemplified by Kweh [19], Chang [20], Venner [21], and Elcoate [22].

This has progressed from a consideration of stationary, one-dimensional, idealized roughness features to recent work on moving surfaces having two-dimensionally varying roughness based on measured surface topographies. Almost all work has, however, focused on relatively thick film conditions, and only very recently have workers started to address the question of what happens when there is true mixed lubrication and some load is borne at asperity contact and some by fluid pressure as reported by Chang [23].

A number of techniques have been used to study mixed lubrication experimentally, e.g. that one developed by Spikes [24], but the most informative has probably been optical interferometry which has been applied by Jackson [25], Cusano [26], Kaneta [27], De Silva [28], Liang [29], and Tondor [30] among others. This has generally been applied to investigate the behaviour of artificially-created features such as bumps, dents and scratches. Such features tend to be larger than the asperities present on real rough surfaces and thus easier to examine optically. Their study also provides more insight into the influence on film thickness of the various component geometrical features of a rough surface than is provided by employing a more complex system.

Kaneta and Cameron [27] used optical interferometry to study the behaviour of an artificially-roughened ball surface against a glass disk in both nominally pure rolling and in sliding. Kaneta [31, 32] also investigated the behaviour of a sputtered ridge at a range of slide roll ratios, from pure rolling to full sliding, and, recently, in reciprocating motion.

The valvetrain system needs several tools to be fully characterized because this system involves several phenomena at different time-scale and length-scale. The first and easier approach is the kinematic one. The kinematic analysis of mechanism helps in answering many questions related to motion of the follower. The next step is the pure dynamic and the then, the multi-body methodology.

The basic design of the conventional IC engine valvetrain is such that engineers can gain significant and useful insights about a valvetrain by applying relatively simple computational tools aimed at cam profile construction, kinematic and quasi-static analysis of the valvetrain mechanism, or rigid dynamic analysis with one or few degrees of freedom as reported by Hundal [33], Johnson [34], Sakai [35], and Kanesaka [36].

Along with such methods evolved ways of interpreting their “inexact” results in order to address valvetrain performance, durability and noise issues. Desai [37] is carried out the complete kinematic and dynamic analysis of cam and follower mechanism using analytical method. The equations for governing motion of the follower have been taken from the Shigley’s book [38]. The dynamic analysis includes the static and inertia force analysis of the follower. For normal working of mechanism, the resultant vertical force has to be in downward direction. If the instant force changes its direction, lifting of follower will take place and design will fail. In the Desai’s paper [37], kinematic parameters and forces are calculated analytically and critical angular speed of rotation is found for the design specification.

The kinematic tool could be expanded to include other kind of analysis up to involve the multi-body analysis as the multi-purpose valvetrain analysis tool developes by Keribar [39], which is aimed at addressing all design issues arising in various stages of valvetrain development. Its capabilities include polynomial cam design, valvetrain mechanism kinematics, quasi-dynamic analysis, spring design / selection, multi-body elastic analysis of a single valvetrain with cam-follower and bearing tribology, and multi-valvetrain dynamics with camshaft torsional vibrations.

The same methodology cold be applied to develop complex and up-to-date VVA – Variable Valve Actuator - valvetrain system but pay attention to that software includes hydromechanics and hydraulics tools to solve the equations derive from cam phaser and lash adjuster. The design process of VVA mechanism can be greatly accelerated through the use of sophisticated simulation tools. Predictive numerical analysis of systems to address design issues and evaluate design changes can assure the required performance and durability. One notable requirement for the analysis and design of novel mechanically-actuated VVA system is a general-purpose fast and easy-to-use planar mechanism kinematics analyzer with cam solution/design features, which can be applied to general mechanisms.

The work of Okarmus et al. [40] introduces a general simulation and design tool, which features general planar kinematics and multi-body dynamics analysis capabilities, as well as integrated hydromechanics and hydraulics to model devices such as lash adjusters and cam phasers. Application of the methodology to various mechanically-driven variable valve actuation systems is discussed in the Okarmus et al. work [40], with focus on a specific system. The Authors brake down the modelling process into multiple stages. First, they analyze the kinematic motion of valvetrain components along with the procedure to calculate the cam shape profile required to produce the desired valve lift. Second, a constrained-dynamics simulation of a rigid system is carried out in search of nominal – quasi-dynamic -, inter-component forces, valve spring margin and cam-follower separation speed. Third, a complete multi-body dynamics analysis, which considers the elasticity of valvetrain components and inter-component contacts, is employed to produce a wide array of detailed dynamic predictions. In particular, the swing-cam type, variable valve actuation mechanisms presented by Flierl [41] and other authors [42] are discussed in the Okarmus' paper [40].

As stated by Du and Chen [43], a high–fidelity flexible multi-body system model is a powerful tool for development of valve train performance. Better representation of the critical parts of valve train is achieved with the simulation of the forced vibrations of the valvetrain due to cam, gas and inertia forces. Analysis of the valve train using calculation methods are best carried out at earliest stage of engine design. A lumped/distributed–parameter dynamic model developed by Hsu and Pisano [44] was introduced into the Du et al. [43] work to investigate the dynamic response of a finger – follower cam system. While Lin and Pisano [45-46], and Lin and Hodges [47], respectively, have worked on helical springs in the areas of variable pitch angle and radius, spring performance under high – speed dynamic loading conditions, and resonance suppression.

Du and Chen [43] have developed a DADS model where the valve springs are treated as flexible bodies to account for mass effects and coil contact as the springs deform, in according with the work done by Zeischka [48] and Schlachter [49]. The cam and follower are treated as a force contact relation, instead of a kinematic constraint, to allow separation of the cam from the follower. Many important effects are predicted with this simulation which includes torsional wind-up of the cam, valve float, spring coil clash, and contact forces throughout the system.

A typical valve train comprises a large number of contacting elements whose interactions are governed by a wide range of coupled phenomena. Consequently, to accurately predict their mechanical behaviour, a detailed transient dynamic model of the mechanism is an important prerequisite as highlighted by Teodorescu and Kushwaha [11].

Teodorescu and Kushwaha [50] outline an initial multiphysics approach to carry on an elastodynamic transient analysis of a four – cylinder valvetrain system, consisting of constrained multi-body dynamics, component flexibility, kinematics and frictional behaviour of contacts. These analyses were also verified by an experimental test rig and by closed-form analytical models.

Therefore, tribological study of the cam - tappet pair cannot be divorced from the dynamic of the valve train system as a whole as highlighted by Teodorescu and Votsios [51].

The ideal model should account for the transient interactions between several physical phenomena ranging from system level interactions to those at microscale. This approach by Teodorescu and Kushwaha [11] is called multi-physics multi-scale. The approach requires integrated solutions for all the interacting phenomena, such as Lagrangian dynamics for rigid body motions, and Reynolds equation for lubricated conjunctions. For a complex system such as a valve train, this requires differential equations of motion for several parts, constraint functions for their assembly, Reynolds and elasticity equations for all the contacts. The result as shown by Teodorescu and Kushwaha [11] is a system of differential-algebraic equations, which must be solved simultaneously both in time and space domains.

After the multi-body analysis of the valvetrain system, the lubrication analysis is run out. The buffers from this two kind of analysis are the results of the multi-body simulations, in terms of a matrix of loads and speeds, which become the input of the lubrication analysis. Computational methods have also been applied to model surface contact and lubrication between components, in order to help address the key valvetrain tribological issues, i.e. friction, wear and durability as highlighted by Zou [52], Dyson [53], Staron [54], and Colgan [55]. Usually in the cam – tappet interface or in the cam – roller contact the elasto-hydrodynamic lubrication regime is established and, historically, the Grubin's equation solves it out.

Grubin [56] in his pioneering work introduced the elastic deformation of the solids and the piezo - viscosity of the lubricant in the hydrodynamic calculations. Grubin used the Hertzian contact shape to describe the inlet geometry of the gap and in combination with the Reynolds equation he was able to calculate the film thickness at the beginning of the high pressure zone – Hertzian zone -, then he assumed that down stream the film is parallel – incompressible lubricant -. By solving numerically this simplified problem he was able to derive the first EHL regression formula for central film thickness calculations, he also speculated about the existence of the pressure spike at the outlet of the contact. Block [57] and Cameron [58] argue that Ertel is the true originator of the work, therefore the work shall be referred as Ertel – Grubin. The work of Morales-Espejel and Wemekamp [59] takes the task of reviewing the main aspects of Ertel – Grubin solutions for film, pressure, friction, temperatures, transient effects, roughness, and shear thinning.

In the elasto-hydrodynamic lubrication – EHL – of line contacts, many formulae to calculate the minimum film thickness have been presented. However, there are very few formulae to calculate the central film thickness. Because the research on partial EHL is often based on the central film thickness of full EHL as the work of Zhang and Wang [60], the accurate film thickness formula for full EHL calculations is needed. In 1978, Dowson and Toyoda [61] presented a central film thickness formula which was more accurate than Grubin's one [56]. The coefficients and indexes in the film thickness formula should be changed for different conditions. However, those of either Grubin's formula or Dowson – Toyoda's formula are all constants. Therefore they are only suitable for certain ranges.

Zhang and Gou [62] and Zhang [63] have worked at many complete numerical calculations over a wide range of operating conditions. Using these numerical results, they have presented two universal formulae, to calculate the minimum film thickness and the central film thickness in elastohydrodynamic lubrication – suitable for all regimes -.

In addition, they have proposed the limitations and the definite applicable ranges of Grubin's and Dowson – Toyoda's formula.

In the elasto-hydrodynamic lubrication regime, the infinite line contact case has been the first case solved by analytical solution of the Grubin's equation. The solution of this case has achieved step by step and not includes at this point the presence of the film squeeze effects. The first step was made by Crook [64], which simplified the inlet gap geometry by introducing the remarkable good approximation to the Hertzian geometry $z \propto x^{\frac{3}{2}}$. Whit this approximation, Archard and co-workers [65] derived an analytical solution for the central film thickness. Then, they studied the inlet pressures and rolling friction [66] and also starvation effects [67]. Greenwood [68] extended the work of Ertel – Grubin to include the outlet constriction in the gap and the pressure spike, but did not include the outlet pressures. Morales-Espejel [69] and Greenwood and Morales-Espejel [70] using a semi-analytical approach based on linear fracture mechanics, introduced the outlet pressures in the contact and derived a simplified log function to describe it. In this way, a full analytical approach for the line-contact EHL problem with smooth surfaces was possible. Greenwood and Morales-Espejel [71] tried to modify the Ertel – Grubin scheme to include surface waviness at the inlet. Chow and Cheng [72] used an Ertel – Grubin scheme to investigate the effects of roughness and used a stochastic approach.

After the line contact case in the elasto-hydrodynamic lubrication regime, the finite line contact case has approached. This case has solved by numerical approach and includes the squeeze film effects. As described in the Rahnejat's book [73] this case could be studies both in steady state condition and in transient condition. Under steady state conditions, theoretical analysis point to the loss of lubricant film in the vicinity or prior to the cam nose – follower contact as highlighted by Dowson [74] and Xiaolan [75] under pure entrain motion. In fact, it is claimed, through theoretical investigations, that in the aforementioned regions the predominant regime of lubrication is due to the boundary films. Experimental evidence reported by Hamilton [76] does not concur with these theoretically based suppositions. Until recently, the presence of lubricant film measured in these regions could not be explained using the theory of elasto-hydrodynamic lubrication – EHL -. Solutions obtained for transient EHL conditions point to a combined entraining and squeeze film action as shown by Dowson [74] and Xiaolan [75]. The critical role of squeeze film action in lubricant film retention is now established, particularly when lubricant film formation due to entraining action becomes insignificant as stated by Rahnejat [77] and Mostofi [78]. This can occur either at low speeds of entraining motion or as a result of inlet boundary reversal in the vicinity of the cam nose – flat follower contact.

Glovena and Spikes [79] and also Nélias and Trujillo [80] have used modified Ertel – Grubin schemes to include the squeeze – film effect in transient conditions. Elliptical contact extensions of the Ertel – Grubin scheme have been proposed by Archard and Cowking [81] and Snidle and Archard [82].

Up to now, two kind of analysis are encountered: the quasi-static analysis and the transient analysis. In the quasi-static analysis, the loads come from the dynamic calculation, while the speeds come from the kinematic calculation, and both of them become the input for the tribology model. In fact, the concentrated counterformal contact between the cam and the follower is one of the most severe working tribological components in the internal combustion engine. This is due to the fact that the contact conjunction between the cam and the follower is subjected to high loads, which bring about excessive friction within the mating region.

The estimation of the contact load is very important in the design process. The maintenance of a high contact force is beneficial for a number of reasons. First, the conditions that yield low loads can result in separation phenomena such as jump or bounce in the mechanism and contribute to noise and vibration. Second, low contact loads render reduced Hertzian pressures that can result in poor lubrication owing to diminution of elasto-hydrodynamic conditions.

In one of their earlier studies of cam to follower lubricated contacts, Dowson et al [61] carried out a quasi-static analysis to predict the variation of the lubricant film thickness during a cam cycle. They obtain contact loads, Hertzian stresses, film thickness and torques – including frictional torques – for the action part of the cam cycle. The prediction of the oil film thickness between the cam and the follower were based on the elasto-hydrodynamic lubrication theory, using established oil film extrapolation formulae for line contact conjunctions. Their analysis also accounted for the arbitrary movement of the point of contact between the two mating surfaces as it affected the entrain velocity and hence, the generated elasto-hydrodynamic pressures. Taylor [2] also utilized this idea to study the lubrication patterns in valvetrains. Similar quasi-static studies were carried out by Dowson et al [74] under steady state operating conditions. The results obtained from the above analyses indicated that over the base circle and the flank regions, where the radius of curvature and the entraining velocity were high, the cam to follower contact enjoyed thicker lubricant films, whereas in the nose region and around it, the films were very small - $< 1\mu m$ -. Kushwaha et al [50] have carried out finite line EHD solutions under combined rolling and squeeze film action under quasi-static conditions for kinematic cam and follower contact conditions. They studied a modified cycloidal cam profile and predicted the oil film thickness for a complete revolution of the cam by employing the already established solutions for finite line EHD contacts obtained by Rahnejat [77] and Mostofi and Gohar [78]. In the solution provided by Kushwaha et al [50, 86], a lubricant film of the order of about 52 nm was predicted in the zero entraining velocity regions. Similar values, for the minimum film thickness, were also obtained by Fessler and Ham [87] in their study of lubrication and stress analysis of a cycloidal cam.

Most valvetrain analysis are confined to either the study of dynamics of the system or its tribological performance for a given contact zone. This approach is not holistic and ignores their interplay, which leads to tribo-elasto-multi-body dynamics as highlighted by Teodorescu and Votsios [51]. They [51] suggest that a multi-physics analysis approach is required, encompassing large rigid-body displacements of rigid elements, small amplitude vibration of elastic members, and elasto-hydrodynamics of cam – follower