Crankshaft Offset and it's Impact on Piston and Piston Ring Friction Behavior

ABSTRACT

Increasing fuel economy in modern passenger car engines has become one of the primary development targets, due to swiftly increasing raw oil prices. Initially, vast progress was achieved in the development of the combustion process. Computer Aided Engineering (CAE) has been one of the keys to success; however, further potential is now being investigated.

Currently, the primary focus of engine development engineers is mechanical friction. The greatest potential for reducing friction in the valvetrain lies with the roller contacts and surface treatment. The primary focus of the development direction, for the cranktrain is in reducing bearing diameters. Due to increasing specific loads on the crankshaft there are clear limits that have been identified. Meanwhile, the potential for the development of the piston group is almost untouched.

Optimization of the piston skirt contour and/or the ring pack introduces the negative risks blow-by and oil consumption. However, modifications to the crankshaft offset are relatively easy design measures that have almost no risk.

The potential for friction reduction of the pistons and piston rings is outlined here through the application of a crankshaft offset. Testing and simulation are combined for the following disciplines:

- Measurements of piston friction in a fired engine are used to validate the simulation model under some select engine working conditions that have low measurement risks. The influence of crankshaft offset is determined for these working points

- Multibody Simulation (MBS) is used to analyze a wide variety of engine working conditions and different crankshaft offset values. The resulting database, characterizes friction behavior and sensitivity regarding crankshaft offset.

The engine's rpm and load time history, as a result of the vehicle cycle simulation, is used to "drive through" the previously generated database. The information gathered from that data is used to quantify the energy saved, decrease in the average friction power and the reduction of fuel consumption for a standardized vehicle driving cycle.

Utilization of this new approach enables the evaluation of not only the benefits of operating the engine under specific working conditions, but also provides a clear recommendation concerning the magnitude of crankshaft offset under consideration for realistic vehicle driving conditions.

INTRODUCTION

The first step in reducing the mechanical friction in a combustion engine is to identify and quantify the friction sources. Although under fired conditions the friction content will look much different, an initial idea about the share of the different sources of the friction of the entire engine can be identified by measuring the friction of the motored engine, with a step-b-step disassembly of the different components. This "Strip Method" allows the possibility to evaluate the meaning of the different groups of friction sources.

A typical friction share of an inline 2.0L four cylinder gasoline engine in terms of the Friction Mean Effective Pressure (FMEP) is shown in Figure 1. At 2000 rpm and 100°C, the piston group and connecting rod bearings together have a relative friction share of more than 40% and the crankshaft (main bearings) has a share of less than 15%. Assuming, that the four connecting rod bearings may have less than or equal the friction share that the main bearings have, the friction share of the piston group without connecting rod bearings could be estimated to be more than 25% of the entire engine friction. Therefore, a significant reduction of the piston group friction will lead to considerable improvements regarding the friction of the entire powertrain.

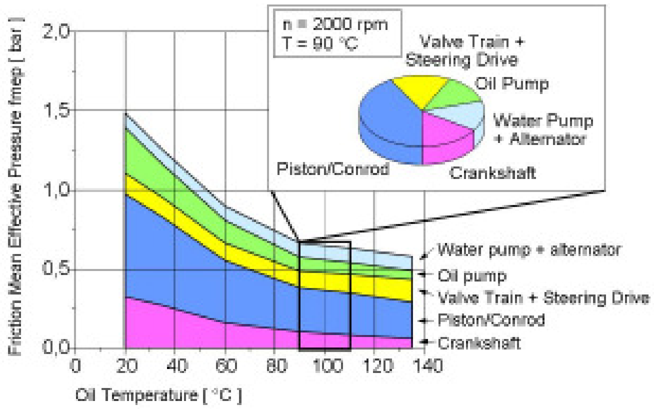

Figure 1: Friction Source Quantification for a 2.0L Gasoline Engine [7]

MEASUREMENT METHOD OF PISTON GROUP FRICTION ON FIRED ENGINE

For an absolute quantification of the piston group friction under realistic engine working conditions the strip method is definitely not sufficient, because the behavior for a fired engine is too much different from a motored one. In addition, to fully understand the physical process it is necessary to review also the friction force vs. crank angle rather than only looking at the integrated value of FMEP. Therefore a lot of testing equipment has been developed, to allow accurate measurements of the piston group friction force [1,2,3,4,7].

Fig. 2 shows the FEV PIston Friction FOrce (PIFFO) measurement setup, working based on a “floating liner concept,” which allows the measurement of the dynamic piston friction force using a 3-Component Force Sensor.

Due to the dynamics of the system, caused by the inertia of the liner in combination with the flexibility of the force sensors, the technology is limited to engine Speeds up to 3000 rpm.

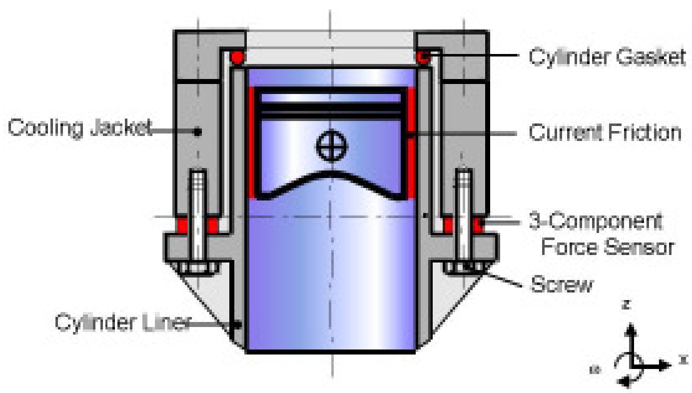

Figure 2: Measurement Setup for Piston group Friction under Fired Conditions [7]

DYNAMIC PISTON SIMULATIONS

The simulation of the piston dynamics has a long history. While the primary dynamics (i.e. piston position as function of crank angle) is given by the kinematics of the cranktrain, focus of the piston simulation is mainly the determination of the piston secondary motion. Piston secondary motion means the lateral piston movement and the tilting of the piston around the piston pin axis. Since primary and secondary movement of the piston may be fully described by three degrees of freedom in one plane, piston dynamics was considered as purely 2d phenomena for years. However, the interaction between piston and cylinder liner depends on three dimensional physics, so nowadays, with the performance of modern computer hardware, three dimensional piston dynamics simulation is state of the art [1,5,6].

In general, to support the engine development process under the strict boundary condition of short development times it is, as always, necessary to work with different model refinement levels, depending on the specific questions to be answered in the different development phases. Therefore there are two different ways, to consider the forces between piston and cylinder liner.

HERTZ CONTACT BASED APPROACH

Especially, if the exact hydrodynamics between the piston and the liner is of minor interest, a hertz contact approach may be sufficient to calculate the piston dynamics. The big advantage is that even the consideration of structural stiffness influences is possible with a limited amount of CPU Time. So, for the calculation of piston slap noise, a dry contact approach is the perfect solution to come up very quickly with results, showing sufficient accuracy.

The big disadvantage of this approach is that the model is not fully physical, but only consists of empirical data that needs to be adjusted. While global stiffness is taken from static FEA and local stiffness is calculated from the Hertz-theory, both, global and local damping effects are completely unknown from the physical point of view and need to be adjusted, when validating such type of simulation model.

Global and local stiffness representing the structural stiffness of piston skirt and cylinder liner on the one hand and the stiffness of the contact as function of the penetration on the other hand need to be modeled as series of two spring-damper systems, as shown in Fig. 3.

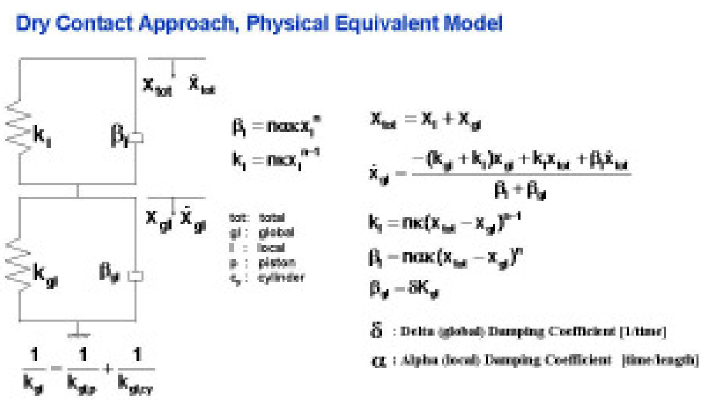

Figure 3: Global and Local Stiffness and Damping of Dry Contact Model

Once the contact pressure is known, it needs to be distributed as forces on the simulation grid of the piston and the cylinder liner, to calculate resulting force and moment for the piston secondary motion, as shown in Fig. 4.

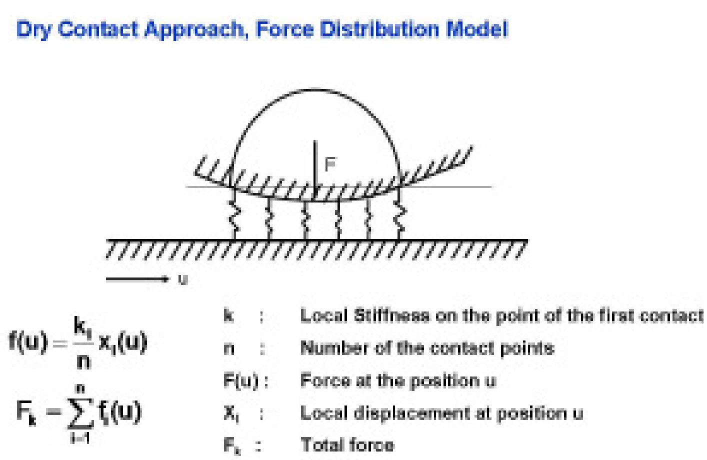

Figure 4: Force Distribution of Dry Contact Model

Beneath NVH applications, also the calculation of the friction between piston and cylinder liner may be covered by this type of simplified approach.

For this, the friction force behavior as function of normal force and sliding velocities (i.e. Stribeck curves) may be stored as lookup tables to be accessed during the dynamic analysis.

Without calculating the hydrodynamic details, the impact on the system's energy dissipation by mixed friction consisting of hydrodynamic and surface to surface contact may be matched surprisingly well as shown later in this paper.

The interesting point, using Stribeck curves is that those are transferable from cylinder to cylinder and also from engine to engine without loosing much accuracy. This means, that friction prediction is possible, using this simple and fast approach, as soon as some experience data for comparable engines has been collected.

HYDRODYNAMIC APPROACH

The usage of a dry hertz contact approach with the adjustment of damping parameters and Stribeck curves is a semi-empirical approach. To come to a more physical model, it is necessary, to understand the behavior of the hydrodynamic oil film between the piston skirt and the cylinder liner.

Assuming, that the oil pressure difference vs. the oil gap height coordinate is small compared to its other slopes and furthermore assuming, that the oil inertia terms may be neglected beneath the oil viscosity terms, the Reynold's differential equation is the mathematical root to describe the exact conditions in the oil gap between piston and cylinder liner.

The solution of the Reynolds equation is the core of any simulation tool, describing the piston secondary motion based on a hydrodynamic approach. The result is the oil pressure distribution in the oil gap. A simple mathematical integration over the gap coordinates delivers the resulting force and attachment coordinate.

Forces and moments are used to calculate the dynamic equilibrium of the piston body. The resulting accelerations are then integrated over time once, to get the velocities and once again to get the coordinates.

From angle, angular velocity, displacement and translational velocity the current gap geometry is now been re-calculated, using the piston skirt contour and the cylinder liner deformation. The input for the next step of the solution of the Reynolds equation is given then.

Fig. 5 summarizes the simulation methodology simplified as it works in typical single purpose programs for the calculation of piston secondary motion.

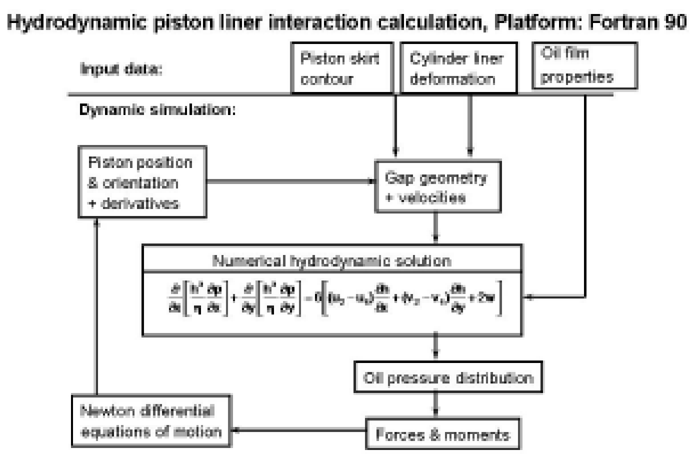

Figure 5: Single Purpose Piston Secondary Motion Simulation Program

MIGRATION TO MBS APPLICATION: ADAMS/ENGINE

The fact, that the piston primary movement is somewhat "hard coded" via the cranktrain kinematics, when working with a single purpose simulation program as described above, may lead to some modeling insufficiencies. Typical examples are:

- Consideration of crankshaft rotational speed fluctuation for the piston primary movement
- Possibility to consider inertia effects of a cable linkage system (“grasshopper leg”) of verification measurements
- Consideration of modified cranktrain kinematics by usage of an additional linkage of a Variable Compression Ratio (VCR) concept.

Due to these limitations FEV decided not to continue the development and enhancement of the single purpose code. Instead the integration into the commercial MBS system ADAMS was focussed.

In a further step, this type of piston dynamics simulation became part of the commercial powertrain application software family: ADAMS/Engine powered by FEV, as shown in Fig. 6. The big advantage is, that the program is now driven by the existing ADAMS/Engine data model and a user friendly GUI.

Topology modifications as linkages and speed fluctuations may now be just modelled into the dynamic system instead of being “programmed” into it by modifying text based source code.

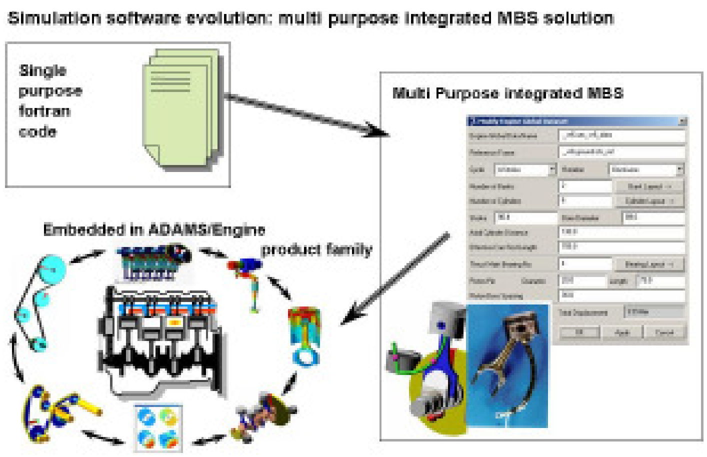

Figure 6: Integration of Dynamic Piston Simulation into ADAMS/Engine

Looking at the software architecture, this integration step looks like described in Fig. 7. From the original calculation code only the hydrodynamic analysis is left. The solution of the Newton equations of motion is now the job of the core MBS integrator. The handling of the input data is being embedded into the relational data structure of ADAMS/Engine. Also the existing post processing features of this application are now used, without any problem.

This point in time was also the right opportunity to include surface contact and mixed friction effects into the calculation procedure. The surface roughness of both sliding surfaces is taken into consideration to model the transition from Coloumb friction over mixed friction up to purely hydrodynamic sliding.

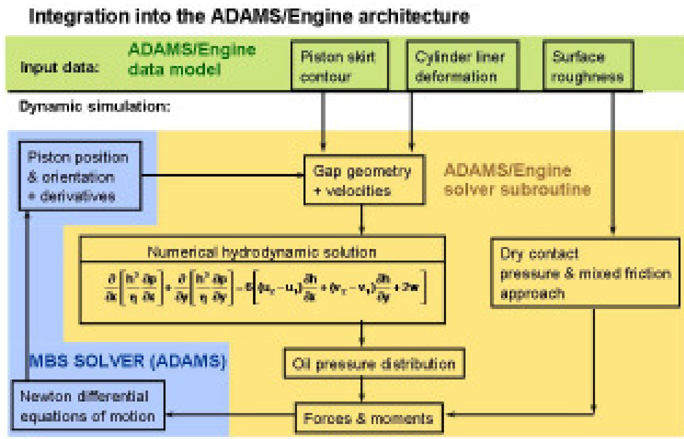

Figure 7: Calculation Procedure for Hydrodynamic Piston Liner Contact Embedded in ADAMS/Engine

The resulting embedded simulation code allows, understanding the piston dynamics at the highest level, including a large number of different effects. Fig. 8 shows a comparison of measured and calculated piston secondary motion for a gasoline passenger car engine. The measurement has been done using a grasshopper leg linkage system which was included for the simulation model as well.

The correlation between measurement and simulation is very good, such that with this simulation model additional engine working conditions, that may be difficult to measure, could be predicted very accurately.

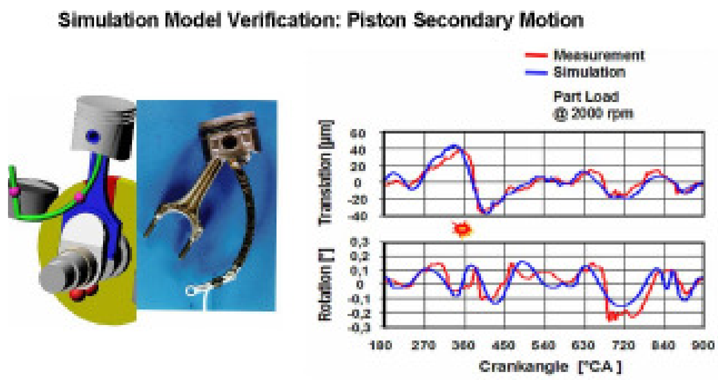

Figure 8: Comparison of Measured and Calculated Piston Secondary Motion

INVESTIGATION OF CRANKSHAFT OFFSET INFLUENCE ON PISTON FRICTION

A Stribeck curve approach is used to predict friction between piston group and cylinder liner and the influence of crankshaft offset on this portion of friction. The simulation model was fed by Stribeck curves from an experience database. All friction values in this chapter contain of piston friction plus piston ring friction. No parameter adjustment was done, such that the simulation has the quality of a real prediction.

The engine on which measurement and simulation are applied is specified as follows:

Rated Power:	75 [kw]
@rpm:	5600 [1/min]
Max Torque:	148 [Nm]
@rpm:	3800 [1/min]
Displacement:	1,8 [l]
Bore Diameter:	81.0 [mm]

With this setup a matrix of cases were measured calculated and compared as follows: 5 rpm's X 3 loads X 3 crank offsets = 45 cases.

Relative p.. error calculation vs. measurement [%]

Engine Speed [1/min]	Full load 0	Full load 6	Full load 12	Part load 3 bar 0	Part load 3 bar 6	Part load 3 bar 12	Motored 0	Motored 6	Motored 12
	Crank Offset [mm]								
1000	-5,2	-5	-1,5	0	2,8	5,9	4,3	3,6	1,3
1500	-7,95	-6,7	-4,7	1,1	5,2	-1,6	1,5	-1,9	-8,4
2000	-5,53	-3,55	-0,8	0,01	5	4,5	-0,6	-3	-3,45
2500	-7,96	-0,35	-0,9	2,24	2,7	1,3	-3,38	-3,7	-4,2
3000	-9,69	-8,9	-9,3	1,9	-3,1	-0,1	-9,67	-8,8	7,7

Figure 9: Table of Variants and Relative FMEP Errors

The Friction of all variants were reduced to Friction Mean Effective Pressures (FMEP), to express the difference between measured and calculated FMEP as relative error percentage which fills up the matrix in Fig. 9. Although no adjustments have been applied to the simulation model parameters the maximum relative error magnitude is below 10%.

The resulting errors do not show a clear significant dependency on the engine load or on the crank offset. Regarding the engine speed, there is a tendency, that the correlation is better for the lower rpm's.

The most probable reason for the differences at higher engine speed is, that the measurement setup obviously shows resonance vibrations that cannot be covered by the simulation and are of course not realistic, when looking at the engine without this measurement equipment. These vibrations are, as explained above, the reason, why fired friction measurement is limited regarding engine speed.

CONCLUSION AND OUTLOOK

Crankshaft offset has a great deal of potential to reduce friction. The key is to quantify the reduction. The amount of friction that can be reduced at full load or at a specific part load condition and if this will over-compensate possible disadvantages at no load are not as crucial as some of the other more relevant questions.

The primary question, once the research is complete is to answer during the development process how much potential will exist in a standard driving cycle and what the optimum crankshaft offset should be to reach this potential.

The process flow for evaluating the potential to reduce friction in a realistic standardized vehicle driving cycle is illustrated in Figure 10. The simulations need to be extended to different engine loads and to higher engine speeds. The result is a completely interlaced surface data structure, describing FMEP as function of the engine's load and speed. The model depth for the piston dynamics is moderate; therefore, the calculation of one engine working condition only takes a couple of minutes.

The engine specific results of a driving cycle simulation, which are basically engine speed and load vs. time, the database is accessed so that for each time step a friction power value becomes available.

Integrating those values across the entire cycle time provides the energy loss by the piston group. The comparison of the different crankshaft offset setups then lead to a relative energy difference for the quantification of the crankshaft offset benefit.

Generally, regarding the benefit of crankshaft offset for engines that work mainly at lower part load points, the benefit will be paid off through their lifetime.

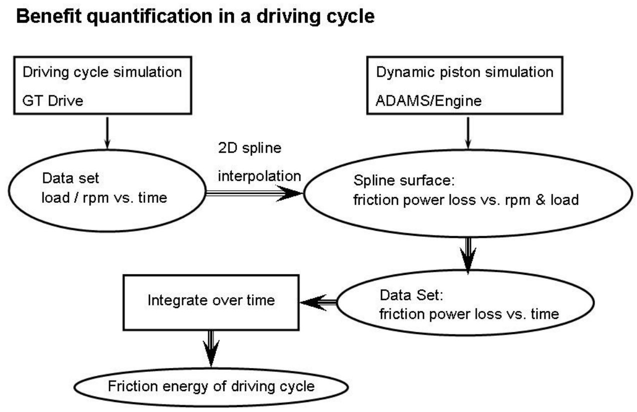

Figure 10: FMEP vs. Crankshaft Offset for Different Engine Speeds for Motored Engine

Engines that will work at or close to full load, crankshaft offset is a good design measure to save fuel. A beneficial future application may be production hybrid driveline concepts.

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ABBREVIATIONS

CAE	Computer Aided Engineering
FEA	Finite Element Analysis
VCR	Variable Compression Ratio
GUI	Graphical User Interface
EHD	Elastohydrodynamic
MBS	Multibody system
FMEP	Friction Mean Effective Pressure
TDC	Top Dead Center
ITDC	Ignition Top Dead Center