

# MODERN COMPUTATIONAL APPROACHES TO POWERTRAIN MECHANICAL LOSS SOLUTION

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*Summary: The paper deals with modern powertrain development, mainly from the area of mechanical loss solution. The Virtual engine as a computational model is used for the solution. There are presented solutions of friction losses of a piston assembly, slide bearings or valvetrain contact couples. Mixed lubrication models are also presented. Hydrodynamic parts of solutions come from the Reynolds equation solution. A boundary lubrication model includes influences of surface roughness on contact pressure. Mechanical loss results for main powertrain parts are presented for the diesel in-line four-cylinder engine. The chosen computational results are validated by measurements.*

*Key words: CO<sub>2</sub> reduction, friction losses, powertrain, mechanical efficiency.*

## 1. INTRODUCTION

Internal Combustion Engines (ICEs) will most likely continue to be used as the powertrain in the majority of road vehicles for more than two decades to come. There are several trends which are apparent on the European markets. The market share of diesel engines is still increasing together with downsizing combined with forced induction (turbocharger, compressor or combinations). Hybrid electric/petrol or electric/diesel engines are expected to gain a market share as well [1, 2].

It is expected that similar trends will occur in the USA and in the Asian markets later. Main drivers are increasing fuel prices, emission of greenhouse gases (mainly CO<sub>2</sub>) contributing to global warming and local pollution due to harmful emissions.

For all trends low friction levels are extremely important for the overall powertrain efficiency. Components and sub-systems where friction reduction can take place are: crankshaft main bearings, conrod bearings, cam shaft bearings, balancer shafts (if present), auxiliary drives (chains, alternator etc.), valve trains and pistons / pistons ring with liners.

The primary benefit of the friction reduction with internal combustion engines is obvious: less fuel consumption, and hence, the reduction of CO<sub>2</sub> emissions.

Unlike the quality of combustion and gas exchange, the engine friction is largely independent of the engine load and particularly in part load engine operation regimes it has a significant impact on the overall efficiency of the powertrain, and therefore on fuel consumption. In normal driving, the friction loss can account for between 10 % and 40 % of the fuel consumption of a medium-sized vehicle with a 2.0 liter gasoline engine. In the case of engines with optimized combustion processes, as seen here in the example of a direct fuel

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injection applied to the same engine, the relative friction loss is even higher which makes it an especially attractive proposition to minimize the friction loss in engines featuring modern combustion processes. This is especially true for a diesel engine with its superior part load fuel consumption performance.

The mechanical losses have to be solved during the design phase of a new powertrain. There are two fundamental principles of mechanical loss solutions, namely experimental and computational solution. Experience shows that the right approach is a combination of these principles, however, the computational methods will expand and this will enable to reduce expensive experiments.

## **2. GENERAL APPROACHES TO MECHANICAL LOSSES SOLUTION**

### **2.1 Experimental approaches**

There are many experimental approaches for the mechanical loss solution of powertrains or powertrain parts. One of them is so-called “stripped method”.

The stripped method uses an electric dynamometer driving the engine or engine subsystems. The electric dynamometer is supplied by a suitable strain gauge enabling to measure torques with the accuracy of 0.5 Nm. The dynamometer - engine system is supplemented by a thermoregulation system that store oil and water temperature in an operating range. The accuracy of the thermoregulation system is about 1°C. The measurement phase starts with the measurement of the whole powertrain without combustion process. Then all the requested powertrain parts are measured separately. The friction losses of the engine subsystems are evaluated from the measurement results by the subtraction of friction loss portions of the engine subsystem [9].

The disadvantage of this approach is that it requires a real powertrain and this fact can be a problem in the first stages of the powertrain development process. Of course, there are also high financial expenses in combination with the long duration of tests.

### **2.2 Computational approaches**

In general, computational approaches to the solution of powertrain mechanical losses can be very roughly divided, for example, according to the model complexity and solution time demands into:

- Models based on analytical solutions of powertrain dynamics or using Multi-Body Systems (MBS) for powertrain dynamics solutions. The models use friction coefficients, Stribeck's curves or Sommerfeld's numbers. The solution is performed with the use of programming languages (Fortran, C++, Matlab, etc.) or spreadsheet programmes only (EXCEL). This approach is used mainly in the first stage of a powertrain design and it has to be supplemented by parameters known from the measurement on similar powertrains. This type of solution can be relatively simple and very quick, however if complex MBS models are used it is a little bit time consuming.
- Models using highly specialized single-purpose programmes. These models enable detailed solution of friction losses and are written in programming languages (Fortran,

C++, Matlab etc.). These approaches are used mainly for the analysis of power losses in existing powertrains and they are extremely time consuming.

### 3. COMPUTATIONAL MODELS BASED ON ANALYTICAL SOLUTIONS OF POWETRAIN DYNAMICS

#### 3.1 Slide bearing computational models

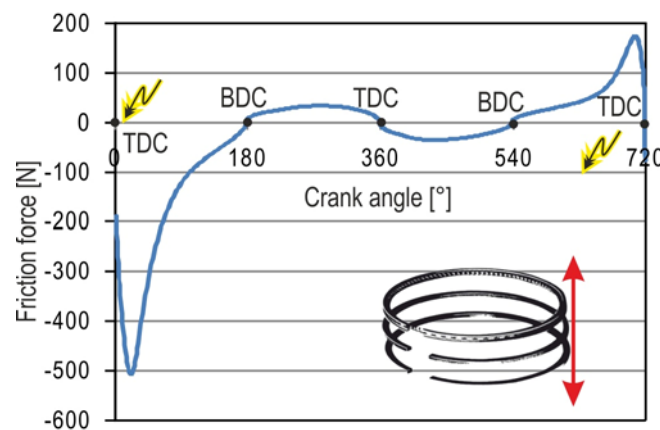
Computational models incorporating precalculated hydrodynamic databases are used for interactions between main parts. This computational model includes also pin tilting influences. Friction losses of the slide bearings are solved with the help of relatively simple analytical approaches based on Sommerfeld numbers. Details can be found in [3].

#### 3.2 Piston assembly friction losses

An empirical model [8] is used for an approximate determination of mechanical losses of the piston rings. This model was created on the basis of measurements taken on a diesel engine of similar parameters. Following [8] the total friction force of all rings can be approximately calculated as

$$F_{tk} = -sign(v) \cdot C_1 \cdot \sqrt{|v|} \cdot \left( 1 - C_2 \frac{T - T_{bez}}{T_{bez}} \right) \cdot \left( 1 + C_3 \frac{p - p_{bez}}{p_{bez}} \right) \cdot \left( \frac{D}{D_M} \right)^2, \quad (1)$$

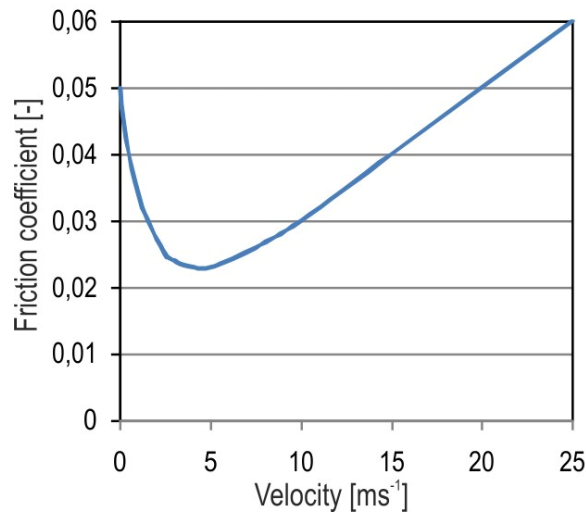
where  $v$  is piston velocity,  $T$  is oil temperature,  $T_{bez}$  is reference oil temperature,  $p$  is combustion pressure,  $p_{bez}$  is reference pressure,  $D$  is cylinder bore,  $D_M$  is reference cylinder bore and  $C_1, C_2, C_3$  are constants. The constant values can be found in [8]. Figure 1 presents the friction force of all piston rings for a first piston for an engine speed 2200 rpm.



Source: Authors

Fig. 1 - Total friction force of piston rings at 2200 rpm

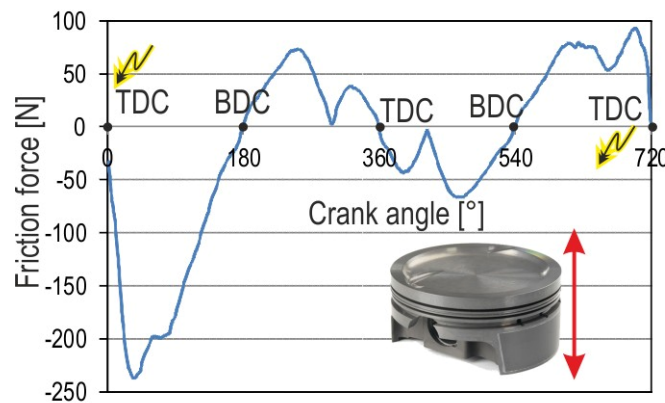
Stribeck curves obtained in measurements taken on similar engines are used for friction loss calculations of a piston-liner friction. The Stribeck curve presents the dependency of a friction coefficient on the piston velocity and it is shown in Figure 2.



Source: [9]

Fig. 2 - Stribeck curve defining the dependency of friction coefficient on piston velocity

Figure 3 presents the friction force between the first piston and the liner for the engine speed 2200 rpm.



Source: Authors

Fig. 3 - Friction force between the first piston a the liner for an engine speed 2200 rpm

### 3.3 Friction losses of contact couples

Accurate contact conditions between valvetrain parts are very difficult to solve. In most cases, there are elastohydrodynamic contacts in conditions of mixed-friction lubrication. Considering the demands laid on the computation of the elastohydrodynamic problems, only contact forces are included into the solution of contact couples. The contact force between two parts is defined for the penetration larger than zero. The force is zero when there is no penetration. Thus the contact force is calculated as

$$F_c = \max(k \cdot p^e - b \cdot v, 0) \quad , \quad (2)$$

where  $k$  is stiffness,  $e$  is exponent,  $p$  is penetration,  $v$  is velocity and  $b$  denotes effective damping coefficient with a value of damping,  $b_{max}$ , for a penetration larger than the penetration depth,  $d$ .

The friction force is then calculated as

$$F_{Tc} = F_c \cdot \text{step}(v_s, -v_s, -1, v_s, 1) \cdot \text{step}(|v|, v_s, \mu_s, v_{tr}, \mu_d), \quad (3)$$

where  $F_c$  is normal contact force,  $v_s$  is slip velocity (static friction transition velocity),  $\mu_s$  is static friction coefficient,  $v_{tr}$  is transition velocity (dynamic friction transition velocity) and  $\mu_d$  is dynamic friction coefficient. STEP function is special ADAMS/Solver function, for details see [7].

#### 4. HIGHLY SPECIALISED COMPUTATIONAL MODELS

##### 4.1 Solution of Slide Bearing Friction Losses

Friction loss solutions of low loaded powertrain slide bearings incorporating main and crank pin bearings, camshaft bearings or balancing shaft bearings can neglect the elastic deformations. The solution is based on the Reynolds equation

$$\frac{\partial}{\partial x} \left( \frac{\rho h^3}{12\eta} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{\rho h^3}{12\eta} \frac{\partial p}{\partial y} \right) - U \frac{\partial(\rho h)}{\partial x} - \frac{\partial(\rho h)}{\partial t} = 0, \quad (4)$$

where  $x$  and  $y$  are coordinates,  $t$  is time,  $h$  is oil film thickness,  $\rho$  is oil density,  $\eta$  is dynamic oil viscosity and  $U$  is relative velocity.

The dependency of oil viscosity on pressure can be expressed with the use of the Roeland's relation [5]

$$\eta(p) = \eta_0 \exp \left[ (\ln(\eta_0) + 9.67) \left( -1 + \left( 1 + \frac{p}{p_0} \right)^z \right) \right], \quad (5)$$

where  $z$  is pressure-viscous index, the value is typically 0.6. The value of a pressure constant  $p_0$  is  $1.96 \cdot 10^8$  Pa. The dependency of oil density on pressure can be neglected easily too because the actual oil density  $\rho$  is only 1.34 times higher ( $\rho = 1.34\rho_0$ ) for pressures approaching infinity ( $p \rightarrow \infty$ ).

The solution of a hydrodynamic problem requires reaction forces ( $F_x$  a  $F_y$ ) and angular velocity ( $\omega$ ). These values are obtained from the virtual engine. Figure 4 shows the principles of the transfer of reaction forces and angular velocities (Fig. 4b) from MBS program (Fig. 4a) to the author's written specialized programme (Fig. 4c).

Subsequently, the force equilibrium condition is solved

$$\begin{pmatrix} f_x \\ f_y \end{pmatrix} = f = F_{OUT} + F_{IN}(\varepsilon, \delta, \omega) \quad (6)$$

where  $F_{OUT}$  is a matrix including outer forces ( $F_X$  and  $F_Y$ ) obtained from the virtual engine,  $F_{IN}$  is a matrix including forces calculated from oil film pressures,  $\varepsilon$  is relative eccentricity and  $\delta$  is an angle of a minimal oil film gap compared with a coordinate system.

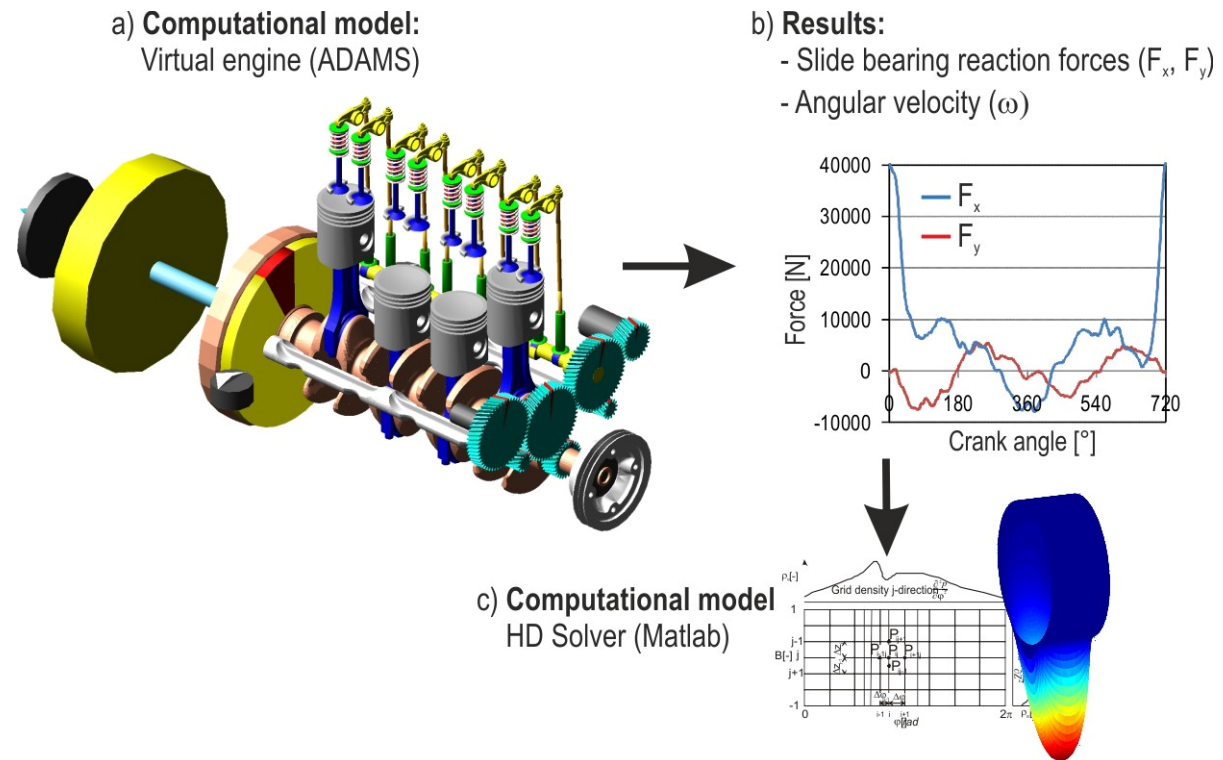
Oil film forces in axes X and Y can be calculated as

$$F_{OUT\_X} = - \iint_S p \cos(\varphi - \delta) dS = - \int_{-\frac{B}{2}}^{\frac{B}{2}} \int_0^{2\pi} p_D \cos(\varphi - \delta) R d\varphi dy, \quad (7)$$

$$F_{OUT\_Y} = -\iint_S p \sin(\varphi - \delta) dS = -\int_{-\frac{B}{2}}^{\frac{B}{2}} \int_0^{2\pi} p_D \sin(\varphi - \delta) R d\varphi dy \quad (8)$$

B denotes the bearing width and  $\varphi$  is an angular coordinate.

The slide bearing model is programmed in Matlab. Gauss-Seidell method together with SOR (Successive Over-Relaxation) techniques is used to solve the hydrodynamic problem.



Source: Authors

Fig. 4 - Friction force between the first piston a the liner for an engine speed 2200 rpm

The aim of the force equilibrium condition is to find input parameters ( $\varepsilon$  and  $\delta$ ) minimizing the f function. In the case of a numeric solution, the f function has to be smaller than the chosen error. An iterative algorithm incorporates the Newton's method and can be suggested as

$$\begin{pmatrix} \varepsilon_{k+1} \\ \delta_{k+1} \end{pmatrix} = q_{k+1} = q_k + \Delta q_k \quad (9)$$

$$\Delta q_k = J(q_k)^{-1} f(q_k) \quad (10)$$

where  $q_k$  is a matrix of generalised coordinates in step  $k$ ,  $J$  is Jacobian or tangent matrix and can be calculated as

$$J(q_k) = \begin{bmatrix} \frac{\partial f_x}{\partial \varepsilon} & \frac{\partial f_x}{\partial \delta} \\ \frac{\partial f_y}{\partial \varepsilon} & \frac{\partial f_y}{\partial \delta} \end{bmatrix} \quad (11)$$



Jacobian (11) is solved numerically, when for every step  $k$ , the function  $f$  is evaluated in axes  $X$  and  $Y$ :

$$f_x(\varepsilon_k), f_y(\delta_k), f_x(\varepsilon_k + \Delta\varepsilon_k) \text{ and } f_y(\delta_k + \Delta\delta_k). \quad (12)$$

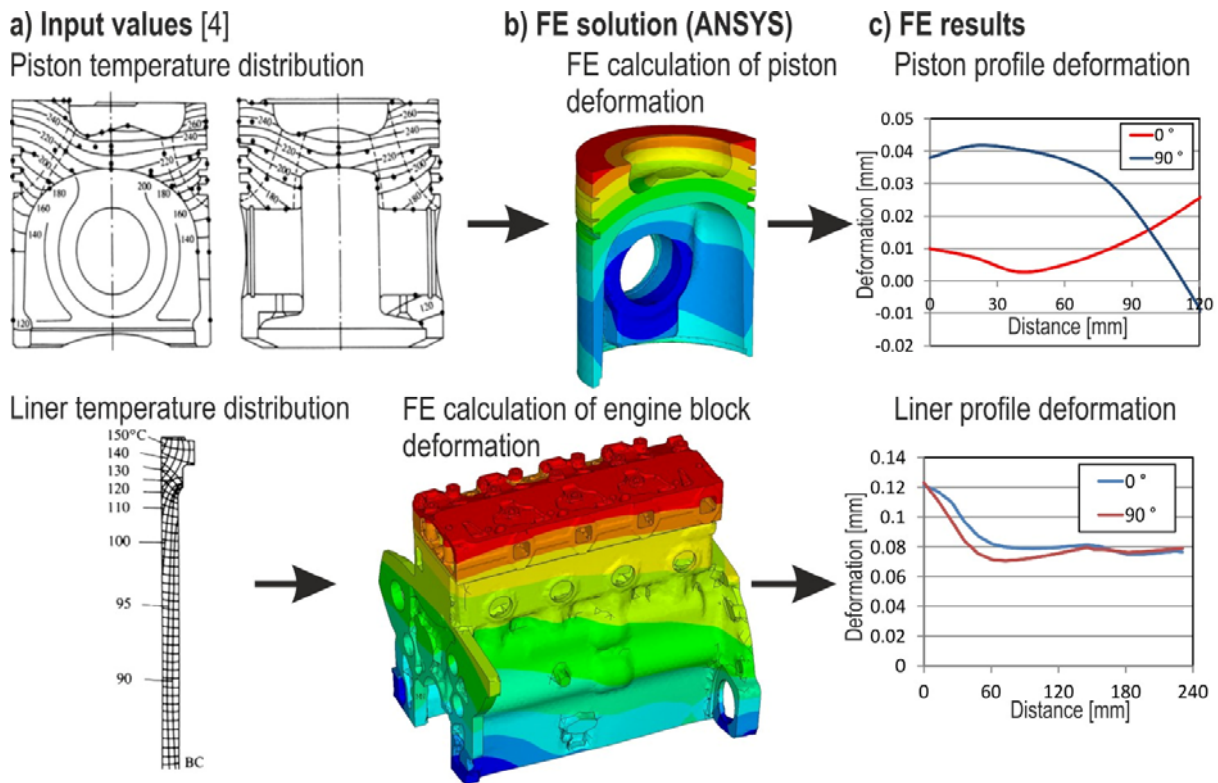
If the pressure distribution in oil is known, the friction force can be calculated as

$$M_t = R \iint_S \left( \frac{\eta\omega R}{h} - \frac{h}{2} \frac{\partial p}{\partial x} \right) dS, \quad (13)$$

where  $R$  denotes the bearing radius.

## 5. SOLUTION OF PISTON MECHANICAL LOSSES

A piston-liner interaction is a complex thermo-mechanical problem. The MBS approach is used for the solution of piston dynamics including piston-liner mechanical losses. The MBS solution requires several inputs, one of them is an exact shape of piston and liner side profiles at room temperature. Values of the piston or liner profiles can be found in the supplier's documentation.



Source: Authors

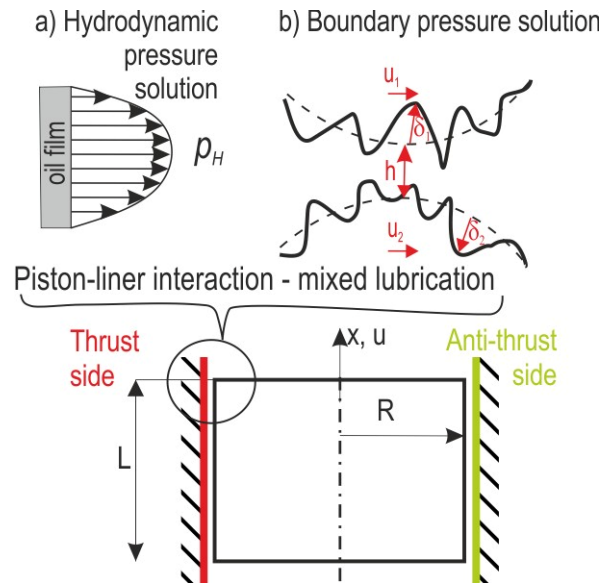
Fig. 5 - Principles of piston and liner deformation solutions by thermal loads during engine cycle

The shapes of a piston and an engine block including liners are highly influenced by thermal loading during the combustion process. Temperatures during an engine cycle are in general time variable, but with relatively small errors temperature distributions can be considered as steady-state for the whole engine cycle. The thermodynamic analysis of the target engine is not available, thus the temperature distribution of similar powertrain is used

[4]. The principles of piston and liner deformations caused by thermal loading are presented in Figure 5.

Elastic deformations caused by hydrodynamic or contact pressures also have to be considered for the solution. Stiffness analyses of the piston and engine block with liners are calculated using Finite Element Methods (FEM) and the results represented by stiffness in discrete points are used for the solution of piston dynamics.

A mixed lubrication model is used for interactions between the piston and the liner. The model consists of hydrodynamic and boundary lubrications, the principles are shown in Figure 6.



Source: Authors

Fig. 6 - Friction force between the first piston a the liner for an engine speed 2200 rpm

The hydrodynamic solution is based on the solution of Reynolds equation (4), but the Reynolds equation is simplified and only two-dimensional form of this equation is considered, then the two dimensional form is

$$\frac{d}{dx} \left( h^3 \frac{dp_H}{dx} \right) = 6\eta u \frac{dh}{dx} + 12\eta \frac{dh}{dt} \quad (14)$$

Simplified conditions like constant oil density and viscosity together with the splitting of the solution domain into two separate domains (thrust and anti-thrust side) are also used. There are not presumed any side oil flows. The solution for each time step runs iteratively using Finite Difference Method (FDM).

When the solution of boundary lubrication is proposed to be based on Greenwood and Tripp [6], then the contact pressure is

$$p_B = KE' F(h / \sigma) \quad (15)$$

where K and E' can be calculated using [4] as

$$K = \left( 8\pi \frac{\sqrt{2}}{15} \right) (N\beta\sigma)^2 \left( \frac{\sigma}{\sqrt{\beta}} \right) \quad \text{and} \quad (16)$$



$$E' = \frac{2E_1E_2}{E_1(1-\nu_1^2) + E_2(1-\nu_2^2)} \quad (17)$$

where  $N$  is the number of asperities per unit area,  $B$  mean radius of curvature of asperities,  $\sigma$  is standard deviation of combined roughness,  $E_i$  is Young's modulus of the contact parts,  $h$  is gap width,  $F(h/\sigma)$  is roughness contact pressure function,  $w$  is function low-limit parameter and  $N$  function exponent.

The roughness contact pressure function is

$$\begin{aligned} F(h/\sigma) &= (w - h/\sigma)^n && \text{for } h/\sigma < w && \text{or} \\ F(h/\sigma) &= 0 && \text{for } h/\sigma \geq w && . \end{aligned} \quad (18)$$

Then the total pressure is

$$P_{MX} = P_H + P_B \quad (19)$$

The integration of total pressure over the piston domains gives the total force and moment. The friction force is calculated using the equation (14) for hydrodynamic part and a coefficient of the friction for boundary part of lubrication.

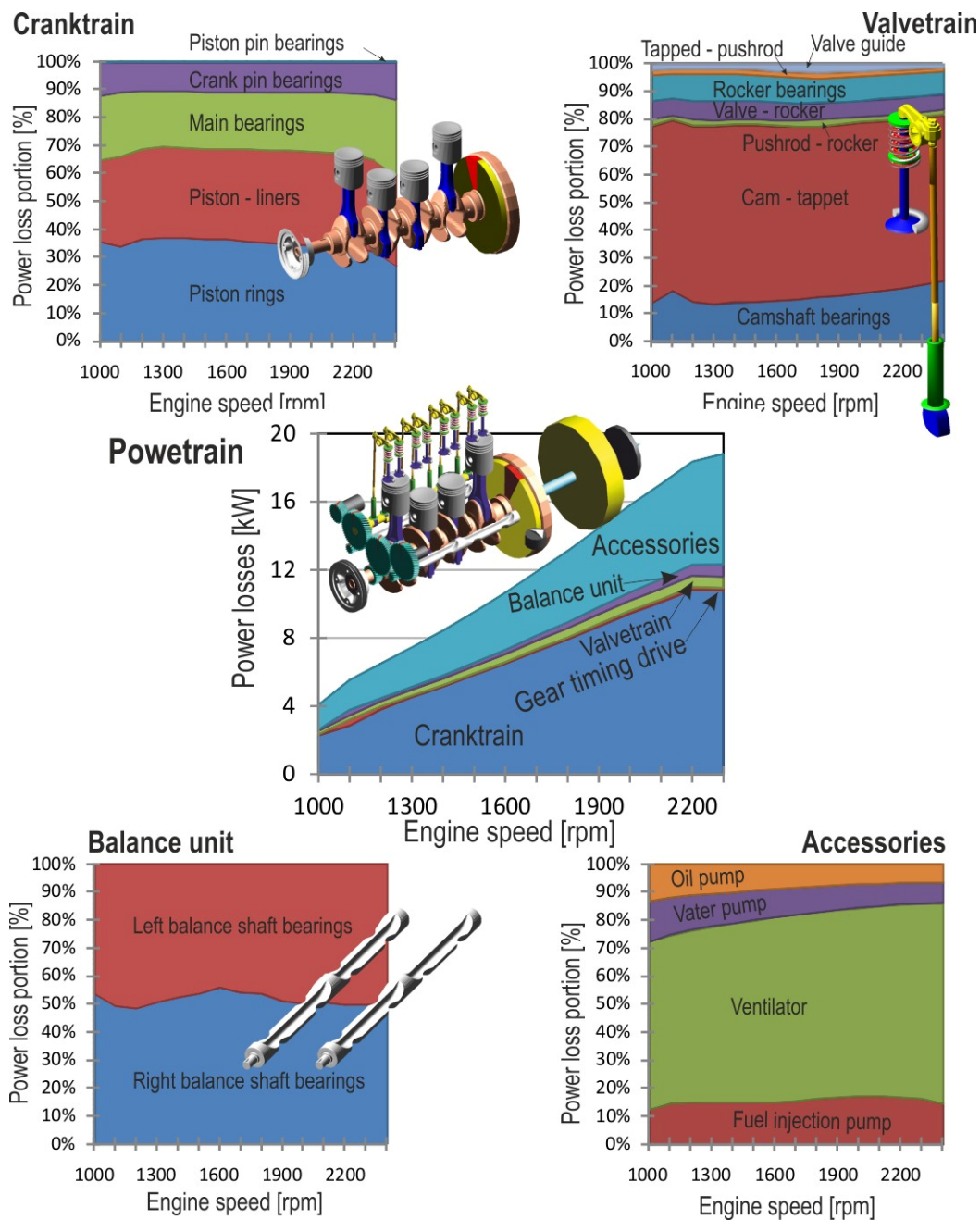
## 6. RESULT EXAMPLES

In general, the choice of a target powertrain is not too important for the presentation of computational method potentials but powertrains with a relatively bigger displacement can be sometimes better choice for validations by experimental methods. The turbocharged diesel inline four-cylinder engine has been selected for the application of computational and measurement methods of mechanical loss solution. Some of the engine parameters are as follows: engine volume 4.12 litres, peak power output 96 kW at engine speed 2200 rpm and compression ratio 17.8. The engine includes an OHV valvetrain with two valves per cylinder and a camshaft located in a crankcase. The slide bearings are used for the bearings of the crankshaft as well as the camshaft. The valvetrain is driven by the front end of the crankshaft using helical gears. A balance unit with two shafts is used for the balancing of the second order inertia forces. The engine includes a mechanically controlled fuel injection pump and other accessories such as a piston compressor, oil and water pumps or a cooling ventilator. The crankshaft does not include a torsional damper.

In general, there are many ways of mechanical loss presentations. Power loss [kW] units or Friction Mean Effective Pressure (FMEP) [bar] can be used to quantify the losses but for this paper the power losses are used. Mechanical losses represented by power losses are determined on the basis of virtual engine results and presume engine full load conditions. The results are presented for engine speeds from 1000 rpm to 2400 rpm.

Figure 7 shows the summary of mechanical losses of target powertrain including powertrain parts. It is evident that the highest portion of mechanical losses is taken by the cranktrain. A relatively high portion of the power losses occurs in powertrain accessories (such as the ventilator, oil and water pumps, the compressor or the fuel injection pump). An accurate determination of accessory power losses vs. engine speeds without any measurement is very difficult, therefore, it is necessary to use the data provided by accessory producers, often for engine nominal speeds only.

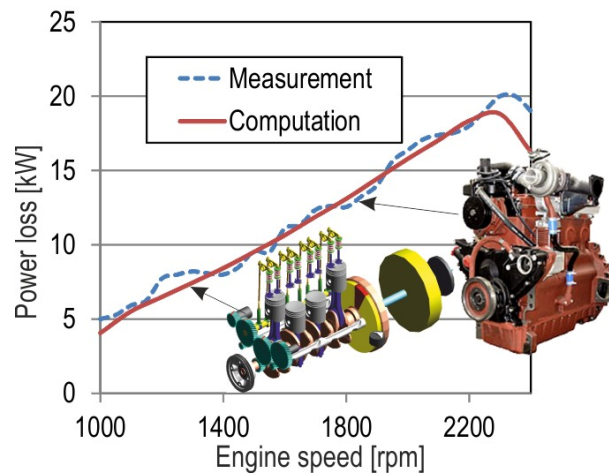
The biggest portion of valvetrain mechanical losses occurs in the cam-tappet contact and partially in camshaft bearings.



Source: Authors

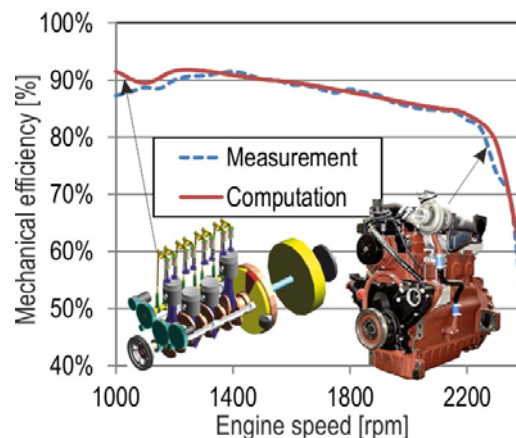
Fig. 7 - Summary of mechanical losses of target powertrain including powertrain parts

The presented computational methods based on a virtual engine are validated by measurements of the target powertrain. The comparisons of the result examples are shown in Figures 8 and 9, which compare the computed and measured powertrain mechanical losses and mechanical efficiency.



Source: Authors

Fig. 8 - Powertrain mechanical losses – computation and measurement result comparisons



Source: Authors

Fig. 9 - Powertrain mechanical efficiency – computation and measurement result comparisons

## 7. CONCLUSION

As the development phase of products in automotive industry is being shorter and shorter, the computational methods applied for the solution of different physical phenomena are more important and they represent powerful means also for the development. In new powertrains the massive increase of turbocharged SI (Spark Ignition) or diesel engines with the emphasis even more on downsizing will even more intensify the utilization of high detailed computational methods.

The presented methods enable to solve mechanical losses in high details, but there are still many input parameters which have to be supplied by experiments on similar powertrains. Further development in the area of these methods is therefore vital and it will be in a close cooperation with the experimental methods because the computational method results have to be validated.

## 8. ACKNOWLEDGEMENTS

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## REFERENCES

- [1] SCHWADERLAPP, M., DOHMEN, J., HAUBNER, F. Friction Reduction – Design Contribution to Fuel Economy. In *11. Aachen Colloquium*. Aachen: VKA RWTH Aachen, 2002.
- [2] FRIEDRICH, H. E. Market Share of Powertrain Scenarios till 2030. *ATZ Autotechnology*, 2010, Vol. 10.
- [3] NOVOTNÝ P. *Virtual Engine – A Tool for Powertrain Development*. Inaugural Dissertation, Brno: Brno University of Technology, 2009.
- [4] Heywood J. B. *Internal Combustion Engine Fundamentals*. New York: McGraw-Hill, 1988.
- [5] ROELANDS, C. J. A. *Correlational Aspects of the Viscosity-Temperature-Pressure Relationship of Lubricating Oils*. Ph.D. Thesis. Delf: Technical University Delft, The Netherlands, 1966.
- [6] GREENWOOD, J. A., TRIPP, J. H. The Contact of Two Nominally Flat Rough Surfaces. *Proc. Instn. Mech. Engrs*, 2006, vol. 185.
- [7] MSC Software. MSC. ADAMS/Solver C++ 2005r2. [User's manual], 2006.
- [8] CHOCHULOWSKI, C. *Simulationsrechnung der Kolbensekundarbewegung*. Ph.D. Thesis. München: Technische Universität München, 1985.
- [9] NOVOTNÝ, P., AMBRÓZ, A., PÍŠTĚK, V. Friction Loss Analysis of Powertrain Parts. *MECCA Journal of Middle European Construction and Design of Cars*. Praha: ČVUT v Praze, 2009, Vol. 2.