Simulation model for predicting reciprocating internal combustion engine wear

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Abstract. The article presents the results of software development, using the Modelica language, designed to create and use simulation models as part of reciprocating internal combustion engines digital twins. The software was created on the basis of component, system and declarative approaches, the theory of a casual bond graphs. The software includes submodels for determining the main engine parts wear rate and allows to predict engine parts life time, to simulate normal and accelerated engine reliability and durability tests. In the course of the study, the well-known model for determining mechanical frictional losses SLM (Shayler, Leong, Murphy, 2005) was adapted to calculate the relative change in the wear rate of engine parts. Universal equations are obtained to determine the friction forces as applied to rotationally and reciprocally moving engine parts. A method is proposed for calculating the relative change in the wear rate of parts and the time of failure due to wear, taking into account the physical properties (hardness), geometric dimensions and the speed of the relative movement of parts in the conjunction.

1. Introduction
The modern economy requires the maximum reduction of time and money spent on the development and preparing of production, fast introduction of changes to the products design parameters and operation modes. The development and implementation of digital twin technology can significantly increase the effectiveness of research and development by maximizing the replacement of costly and time-consuming real product tests with virtual ones without sacrificing their quality. This fully applies to reciprocating internal combustion engines - the main sources of mechanical energy for land vehicles operating [1]. Most researchers believe that the model should implement simulation methods [2], that is, with the required accuracy to describe the processes as they would actually take place and with which it is possible to conduct computational experiments in order to obtain information about the dynamics system.

We consider the most known mathematical models of the engine from the point of view of the possibility of their use as the basis of a digital twin. The simplest modeling method is to approximate the dependence of three parameters: fuel consumption, crankshaft speed and torque (for example, the Willan's lines method [3]). Obviously, this approach cannot be used to create digital twins. The same
applies to the phenomenological MVEM models (Middle Value Engine Model), describing the processes in various engine systems [4], they cannot be combined into a single simulation model, in addition, they are not resolved regarding the angle of crankshaft position, which does not allow reliable modeling of fast-moving processes.

The complexity of CFD / FEM models currently does not allow creating a single digital twin based on them, which includes all the systems and mechanisms of engine, in addition, it will not be able to simulate time-consuming processes. Therefore, it is advisable to use the 0D and 1D component simulation models which, if necessary, are interfaced with 3D models as the basis of the digital twin of the engine. This technology is implemented in commercial software products AVL Boost, Ricardo Wave, GT-Suite, Siemens Amesim, partially Modelon, ClayTex, etc.. There are non-commercial developments of universities – OpenWAM (Polytechnic University of Valencia, Spain) [5]), Gasdyn (Polytechnic University of Milan, Italy) [6]), Allbea (Ufa State Aviation Technical University) [7].

M V Maliovanov [8] and R N Khmelev [9] (Tula State University) used a method based on modeling the engine as a single thermomechanical system. In the framework of this approach, a system of equations of the energy and mass balance of the working fluid in the combustion chamber is compiled. Gas-dynamic processes are described by a mathematical model of a one-dimensional unsteady gas flow. In the work [10] M.V. Meliomanov and R.N. Khmelev proposed the use of undirected bond graphs for engines modeling, which allows a universal method of formulating the boundary conditions of interacting engine systems. Other researchers have similar mathematical models of engines, for example [11, 12]. The component approach based on the application of the theory of bond graphs and the 0D and 1D formulation of the computational problem seems to be one of the most rational methods for creating simulation models of reciprocating internal combustion engines. It allows, with sufficient accuracy and acceptable labor intensity, to predict not only the main indicators of the engine purpose and safety (power, fuel consumption, emissions of harmful substances, etc.), but also reliability indicators.

2. Mathematical model

Mathematical model, used in this study, include description of different boundary condition, basic and complex elements, acausal connection. Mathematical descriptions of different components and connections are designed in the form of Modelica classes and combined into a library. The virtual engine model is made up of library components, for which individual parameters are specified. Each component is exchanged with related components (including components simulating the environment) with power:

\[ N(t) = f(t) \cdot e(t) \]  

(1)

here: \( f(t) \) – stream variable; \( e(t) \) – potential variable; \( t \) – time.

The mathematical model of wear is based on the well-known Archard equation [13], which assumes that volumetric wear is equal to the product of the nominal area and the convergence of the friction pair surfaces:

\[ V = k_w \cdot \frac{F_n}{H} \cdot L \]  

(2)

here: \( V \) – volumetric wear; \( k_w \)– wear factor; \( F_n \)– normal force; \( H \) – hardness of the wearable (less solid) part; \( L \) – friction path.

Equation (2) is very inaccurate, since it does not take into account the effect on the wear of parts by their temperature, sliding speed and contact geometry, as well as the contacting bodies microgeometry and the presence of critical points on the dependences of the wear rate on loading factors. In addition, (1) is usually used to simulate slip wear, but since it is based on general assumptions, it can be applied to other types of wear, including adhesive wear that occurs when the engine is started [14].

The friction coefficient \( f \) is not included in the Archard equation explicitly, it can be expressed in terms of the friction force (according to Amonton). Replacing the normal force in the Archard equation by the friction force and passing to the wear rate \( I \), which is the ratio of the wear value \( V \) to the friction path \( L \), on which wear occurred, we get:
\[ I = k_w \cdot \frac{F_{fric}}{H} \]  

(3)

According to [15], the friction coefficient \( f \) is transferred not to the denominator, but to the numerator of (3), and the wear factor \( k_w \) is replaced by the reduced wear factor \( k' \). This approach corresponds to the numerous experimental data, with increasing values of friction coefficient – volumetric wear increases. But, the position of the value \( f \) in (3) does not matter, since with further transformations, it is reduced (see below). The wear factor is determined experimentally. For a reciprocating engine, in most cases, it is not the wear values that are known, but the operating time of the parts before failure (life time). The task of the developers is to determine how certain changes in the design of the engine or its operating modes will affect the value of the life time. That is, with the help of the simulation model, it is necessary to estimate not the absolute change in the amount of wear, but the relative change (relative to some basic design or basic operating mode). Assuming the values of the reduced wear factor and the coefficient of friction to be constant, we obtain the ratio:

\[ \frac{I}{I_{ref}} = \frac{F_{fric \cdot H_{ref}}}{H \cdot F_{fric \cdot ref}} \]  

(4)

here: ref designation refers to the basic design or load condition of the engine.

Equation (4) allows to determine the relative change in the wear rate and, accordingly, the life time of the parts mate before failure. In this case, there is no need to experimentally determine the values of the reduced wear factor and the friction coefficient, which depend on a large number of circumstances.

To determine the friction force value in (4), one can use the well-known SLM model (Shayler, Leong, Murphy, 2005) [16], which takes into account the dependence of friction losses on oil viscosity and, accordingly, its temperature. According to the SLM model, the total pressure of piston engine mechanical losses is the sum of losses in the cranktrain, valvetrain, auxiliary mechanisms and cylinder-piston group. The SLM model has been adapted to determine the frictional force values in a mathematical engine model. In addition, the equations in the SLM model have been transformed into a form that can be used for individual engine components. Let us consider this in more detail using the example of a cranktrain. Mechanical loss pressure in the engine cranktrain mechanism, according to the original SLM model:

\[ P_{crn} = \frac{D_b}{n_c D^2 \cdot S_{p_{max}}}, \left( C_{cb} \cdot n^{0.6} \cdot D_b^2 \cdot L_b \cdot n_b \cdot \left( \frac{\mu}{\mu_{ref}} \right)^{n_1} + C_{cs} \right) \]  

(5)

here: \( n \) – crankshaft rotation frequency; \( n_c \) – number of cylinders; \( D \) – cylinder diameter; \( C_{cb} \) – coefficient of losses in main bearings; \( C_{cs} \) – coefficient of losses in oil seals; \( \mu \) – oil dynamic viscosity at a given temperature; \( \mu_{ref} \) – oil dynamic viscosity at the reference temperature; \( S_{p_{max}} \) – maximum piston stroke; \( D_b \) – bearing diameter; \( L_b \) – bearing width; \( n_b \) – number of bearings; \( n_f \) – exponent.

It is obvious that this form of equation does not correspond to the component approach in the engine modeling. An individual crankshaft bearing cannot and should not “know” the number of cylinders or their diameter, etc. We transform (5), for this we express the power of mechanical losses in a single main bearing (for a four-stroke engine):

\[ N_{crn} = \frac{V_p \cdot P_{crn} \cdot n}{30 \cdot \tau} = \frac{\pi \cdot D^2 \cdot 2 \cdot P_{crn}}{4} \cdot \frac{P_{crn} \cdot n}{30 \cdot \tau}, \]  

(6)

here: \( V_p \) – working volume of the cylinder; \( R_c \) – crank radius; \( \tau \) – number of engine strokes.

Frictional moment:

\[ T_{crn} = \frac{N_{crn}}{w} = \frac{\pi \cdot D^2 \cdot 2 \cdot P_{crn}}{4 \cdot w} \cdot \frac{P_{crn} \cdot n \cdot n}{\tau \cdot 30 \cdot 30}, \]  

(7)

we denote: \( k_m = \pi^2 / (2 \cdot \tau \cdot 30 \cdot 30) \) – dimensionless coefficient (for a four-stroke engine \( k_m = 1.371 \cdot 10^{-3} \), for a two-stroke engine \( k_m = 2.742 \cdot 10^{-3} \)), then:

\[ T_{crn} = 0.129 \cdot k_m \cdot C_{cb} \cdot m^{0.6} \cdot D_b^2 \cdot L_b \cdot \left( \frac{\mu}{\mu_{ref}} \right)^{n_1}, \]  

(8)
Equation (8) can be used to calculate the moment of mechanical losses in the crankshaft main bearings.

Friction power:

\[ F_{crn} = 2 \cdot \frac{T_{crn}}{D_b} = 0.258 \cdot k_m \cdot C_{cb} \cdot m^{0.6} \cdot D_b^2 \cdot L_b \cdot \left( \frac{\mu}{\mu_{ref}} \right)^{n_1}. \]  

(9)

Equation (9) no longer contains “unnecessary” variables from the point of view of the component approach. Substituting (9) with the values of the variables for the simulated and base engines into (4), it is possible to determine the relative change in the wear rate and operating time to failure due to wear. For other rotating couplings, for example, for the valvetrain mechanism bearings, equations similar to (9) can be obtained. Thus, these equations are universal and can be applied to various rotary components of the engine simulation model.

Mechanical loss pressure at the mate between the reciprocating piston and the cylinder liner according to the original SLM model:

\[ P_{pist} = \left( C_{ps} \cdot \left( \frac{v_{p.mid}^{0.5}}{D} \right) + C_{pr} \cdot \left( \frac{v_{p.mid}^{0.5}}{D^2} \right) \right) \cdot \left( \frac{\mu}{\mu_{ref}} \right)^{n_2}. \]  

(10)

here: \( C_{ps}, C_{pr} \) – coefficients of losses in conjugation of the piston skirt and cylinder liner, conjugation of piston rings and cylinder liner, respectively; \( v_{p.mid} \) – average piston speed; \( n_2 \) – exponent.

Performing transformations similar to (6–9) and passing from the average piston speed to the instantaneous \( v_p \), we get:

\[ F_{pist} = k_m \cdot v_p^{0.5} \cdot \left( C_{ps} \cdot D + C_{pr} \right) \cdot \left( \frac{\mu}{\mu_{ref}} \right)^{n_2}. \]  

(11)

Equation (11) is also universal and can be applied to the components of the simulation model with reciprocating movement of parts, for example, for valve guides (with \( C_{pr} = 0 \)). Loss coefficients \( C \) and exponents \( C_{1,2} \) in (10) and (11) can be determined experimentally, or taken from well-known publications [16].

3. Verification

The developed mathematical model was implemented in the software for simulation of reciprocating internal combustion engines [2], created using the Modelica language and the OpenModelica program [17]. To verify the mathematical model, previously obtained experimental data were used [18], including data on the wear rate of parts of four-stroke, 12-cylinder liquid-cooled diesel engines of 12ChN15/18 type with a power of 618 and 246 kW at a nominal speed of 2000 and 1400 min\(^{-1}\), respectively. The wear rate was determined by micrometry of the engine mechanisms main parts every 100 hours of operation and by the rate of wear products accumulation in the engine oil filter (centrifuge). The tests were carried out according to the standard method for determining engines reliability for 800 hours. To tune the simulation model, the parameters of the tested engines, the oil characteristics, data from publication [16] were used. The relative wear rate was determined by calculation for typical loading modes and then reduced to an integral value taking into account the operating time in each mode.

The table below shows the results of comparison of experimental and calculated data on the wear rate of some engine parts.

The rate of wear products accumulation in the engine oil filter of a 618 kW engine (23 g per 100 test hours) is 45% higher than that of a 245 kW engine. The calculation, taking into account the wear rate and the geometric dimensions of individual parts, gave an integral estimate of the increase in the wear rate – 51% (error – 13.3%).
Table 1. Relative wear rate of the engines 12ChN15/18 type main parts.

| Engine parts                  | Relative wear rate (618 kW / 246 kW engines) | Error, % |
|------------------------------|---------------------------------------------|----------|
|                              | Experiment | Simulation |          |
| connecting rod bearings      | 1.26       | 1.35       | 7.1      |
| intake valve guides          | 0.42       | 0.51       | 21.4     |
| exhaust valves guides        | 0.58       | 0.52       | 10.3     |

The obtained calculation errors can be considered satisfactory. They can be reduced by more accurate selection of the empirical coefficients included in the mathematical model in the course of further research.

4. Conclusion

Using the Modelica language, the software for simulation of reciprocating internal combustion engines has been developed. The mathematical model underlying the software is based on the component approach and the theory of bond graphs. In the course of the study, the well-known model for determining mechanical frictional losses SLM (Shayler, Leong, Murphy, 2005) was adapted to calculate the relative change in the wear rate of engine parts. Universal equations are obtained to determine the friction forces as applied to rotationally and reciprocally moving engine parts. A method is proposed for calculating the relative change in the wear rate of parts and the time of failure due to wear, taking into account the physical properties (hardness), geometric dimensions and the speed of the relative movement of parts in the conjunction. The verification of the mathematical model and software was carried out using the results of previous experiments. The research results can find application in the creation of digital twin technology for reciprocating internal combustion engines.

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