Theoretical model for determination of the spark ignition engine thermo-gasodynamic parameters on various functional conditions

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Abstract. In this paper, an analysis and a model of the factors influencing the filling and compression of the fresh load is presented, in case of a spark ignition engine. Equally, the combustion and expansion processes of the engine are taken into account. All these processes are modelled together with the engine load changing, for different engine speed values.

1. Objective
This paper starts from the observation that the technical literature does not approach the theoretically car engine behaviour in different operating modes. For this reason, it is not possible to determine the characteristic energy parameters, in particular, the power and fuel consumption, on a diversity of regimes, imposed by the automobile operation. In this context, the authors are proposing an original calculation model. This model, developed in the present paper, addresses to the specific processes for the spark ignition engine, in the context in which they are determining the evolution of the engine operating cycle assembly and subsequently its performance. Also, the related model envisages different engine operating regimes. It introduces changes of the main parameters when the engine operating mode is being changed.

2. Methodology
In this paper, an analysis and a model of the factors influencing the filling and compression of the fresh charge is presented, in case of a spark ignition engine. Equally, the combustion and expansion processes of the engine are taken into account. All these processes are modelled together with the engine load changing, for different engine speed values. The difficulty of the problem occurs especially for this engine type because the load setting is obtained by changing the throttle position [1] [2]. This adjustment significantly influences the pressure at the end of the intake process, thereby cylinder filling and engine performance. For the development of this theoretical model, it is necessary to be introduced some basic assumptions. It is mainly assumed that changing the functional mode will have an impact on the cylinder fill efficiency, in case of a spark ignition engine, being affected exclusively by the modification of the engine load. In the same time, within the developed model a certain variation of the exhaust gas coefficient in the cylinder engine, depending on its load, is adopted [3] [4].
3. Results

This paper, through the developed model, leads to the achievement of functions that emphasise pressure and temperature variations at the end of each cycle of the engine, in the same time with the engine load changing, for different engine speed values in the operating range.

The expression for the intake pressure, dependent on the engine load, can be written as the following:

\[
p_a(x) = \Phi_0 \cdot (1 - \varepsilon^{-1}) \cdot (1 - \varphi_{pu} + y_r) \cdot \begin{cases} 
-56 \cdot \chi^2 + 95 \cdot \chi + 299 \\
-28 \cdot \chi^2 + 48 \cdot \chi + 312 \\
-167 \cdot \chi^2 + 284 \cdot \chi + 263 \\
-278 \cdot \chi^2 + 473 \cdot \chi + 225 \\
0,2 \cdot \chi + 0,56, & n = 2 \cdot 10^3 \\
0,368 \cdot \chi + 0,406, & n = 3 \cdot 10^3 \\
0,475 \cdot \chi + 0,325, & n = 4 \cdot 10^3 \\
0,69 \cdot \chi + 0,17, & n = 5 \cdot 10^3 
\end{cases} 
\]

Also, this model has developed the expression for intake temperature parameter determination:

\[
T_a(x) = \begin{cases} 
-56 \cdot \chi^2 + 95 \cdot \chi + 299, & n = 2 \cdot 10^3 \\
-28 \cdot \chi^2 + 48 \cdot \chi + 311, & n = 3 \cdot 10^3 \\
-167 \cdot \chi^2 + 284 \cdot \chi + 263, & n = 4 \cdot 10^3 \\
-278 \cdot \chi^2 + 473 \cdot \chi + 225, & n = 5 \cdot 10^3 
\end{cases} \text{ [rpm]} 
\]

The expression for the compression pressure, dependent on the engine load, can be written also:

\[
p_c(x) = \Phi_0 \cdot (1 - \varepsilon^{-1}) \cdot (1 - \varphi_{pu} + y_r) \cdot \begin{cases} 
-278 \cdot \chi^2 + 473 \cdot \chi + 225 \\
-167 \cdot \chi^2 + 284 \cdot \chi + 263 \\
-28 \cdot \chi^2 + 48 \cdot \chi + 311 \\
-56 \cdot \chi^2 + 95 \cdot \chi + 299 \\
0,2 \cdot \chi + 0,56, & n = 2 \cdot 10^3 \\
0,368 \cdot \chi + 0,406, & n = 3 \cdot 10^3 \\
0,475 \cdot \chi + 0,325, & n = 4 \cdot 10^3 \\
0,69 \cdot \chi + 0,17, & n = 5 \cdot 10^3 
\end{cases} 
\]

Regarding the temperature parameter for the compression stroke, it can be determined by using the following formula, dependent on the engine load too:

\[
T_c = \begin{cases} 
-56 \cdot \chi^2 + 95 \cdot \chi + 299 \\
-28 \cdot \chi^2 + 48 \cdot \chi + 311 \\
-167 \cdot \chi^2 + 284 \cdot \chi + 263 \\
-278 \cdot \chi^2 + 473 \cdot \chi + 225 \\
0,375, & n = 2 \cdot 10^3 \\
0,383, & n = 3 \cdot 10^3 \\
0,3937, & n = 4 \cdot 10^3 \\
0,397, & n = 5 \cdot 10^3 
\end{cases} \text{ [K]} 
\]

There was also developed an expression which can be used in order to find the evolution of the combustion process, dependent on the engine load, for four different engine speed values:
The authors developed the following expressions:

\[
p_{s} = \begin{cases} 
-114.3 \cdot \chi^3 + 225.5 \cdot \chi^2 - 150 \cdot \chi + 88.5, & n = 2 \cdot 10^3 \text{ [rpm]} \\
4.1 \cdot \chi^3 - 14.7 \cdot \chi^2 + 19.4 \cdot \chi + 47, & n = 3 \cdot 10^3 \text{ [rpm]} \\
-13 \cdot \chi^3 - 8.6 \cdot \chi^2 + 64.7 \cdot \chi + 31.3, & n = 4 \cdot 10^3 \text{ [rpm]} \\
0.75 \cdot \chi^3 - 51.9 \cdot \chi^2 + 129 \cdot \chi + 7, & n = 5 \cdot 10^3 \text{ [rpm]}
\end{cases}
\] (5)

The same consideration for the parameter of temperature:

\[
T_{s} = \begin{cases} 
-189910 \cdot \chi^6 + 671518 \cdot \chi^5 - 935209 \cdot \chi^4 + 643787 \cdot \chi^3 - 224873 \cdot \chi^2 + 36937 \cdot \chi + 529, & n = 2 \cdot 10^3 \text{ [rpm]} \\
-13246 \cdot \chi^4 + 33828 \cdot \chi^3 - 30833 \cdot \chi^2 + 11857 \cdot \chi + 1280, & n = 3 \cdot 10^3 \text{ [rpm]} \\
-18683 \cdot \chi^4 + 48565 \cdot \chi^3 - 42415 \cdot \chi^2 + 14369 \cdot \chi + 2695, & n = 4 \cdot 10^3 \text{ [rpm]} \\
-17044 \cdot \chi^4 + 44889 \cdot \chi^3 - 40680 \cdot \chi^2 + 14959 \cdot \chi + 1069, & n = 5 \cdot 10^3 \text{ [rpm]}
\end{cases}
\] (6)

For the power process, the authors developed the following expressions:

\[
p_{d} = \begin{cases} 
-5.8 \cdot \chi^3 + 11.4 \cdot \chi^2 - 7.6 \cdot \chi + 4.5, & n = 2 \cdot 10^3 \text{ [rpm]} \\
0.22 \cdot \chi^3 - 0.78 \cdot \chi^2 + \chi + 2.5, & n = 3 \cdot 10^3 \text{ [rpm]} \\
-0.7 \cdot \chi^3 - 0.47 \cdot \chi^2 + 3.53 \cdot \chi + 1.7, & n = 4 \cdot 10^3 \text{ [rpm]} \\
0.04 \cdot \chi^3 - 2.88 \cdot \chi^2 + 7.14 \cdot \chi + 0.4, & n = 5 \cdot 10^3 \text{ [rpm]}
\end{cases}
\] (7)

\[
T_{d} = \begin{cases} 
-97971 \cdot \chi^6 + 346423 \cdot \chi^5 - 482456 \cdot \chi^4 + 332117 \cdot \chi^3 - 116007 \cdot \chi^2 + 19055 \cdot \chi + 273, & n = 2 \cdot 10^3 \text{ [rpm]} \\
-7186 \cdot \chi^4 + 18352 \cdot \chi^3 - 16727 \cdot \chi^2 + 6432 \cdot \chi + 695, & n = 3 \cdot 10^3 \text{ [rpm]} \\
-10394 \cdot \chi^4 + 2702 \cdot \chi^3 - 2360 \cdot \chi^2 + 800 \cdot \chi + 1500, & n = 4 \cdot 10^3 \text{ [rpm]} \\
14439 \cdot \chi^5 - 32479 \cdot \chi^4 + 9555 \cdot \chi^3 + 28990 \cdot \chi^2 - 29619 \cdot \chi^2 + 10247 \cdot \chi + 477, & n = 5 \cdot 10^3 \text{ [rpm]}
\end{cases}
\] (8)

The main results obtained derive from the simulation performed on the modification of the parameters. These parameters modification is expressed and analyzed through graphs which are illustrating the variation of them.

Simulation of the engine cycle at 2000 [rpm] and various loads is illustrated in figure 1 (p-V):
Figure 1: Engine cycle evolutions (p-V) at 2000 [rpm]

The figures 2, 3 and 4 (p-V) illustrates a simulation of the engine cycle evolutions at 3000, 4000 and 5000 [rpm] at various loads.
Figure 2: Engine cycle evolutions (p-V) at 3000 [rpm]

Figure 3: Engine cycle evolutions (p-V) at 4000 [rpm]
**Figure 4:** Engine cycle evolutions (p-V) at 5000 [rpm]

**Figure 5:** Dependence between filling coefficient and engine load
Regarding the dependence between the inlet pressure and engine load calculation, a linear function is introduced in the model which shows, for spark-ignition engines, the filling coefficient variation in case of engine load modification, for different engine speeds. Establishment of this linear variation was made by analysing a data series from the literature [4] [5] [6]. The representation of this variation, for an engine speed interval which ensures a steady engine running is showed in figure 5, as a family of lines with different tendencies. Analysis of this different tendencies highlights that for low engine loads, the filling coefficient has high values in low engine speeds domain. By this way, the lower rate of airflow contributes to lowering the gaso-dynamic losses. Increased efficiency of the filling in the low engine speeds domain is also favoured by increased of the filling time. In case of high engine loads, the gaso-dynamic resistance is decreasing, irrespective of engine speed value, which leads to increased filling coefficient for the high engine speeds domain, when the inertia of the intake air flow is higher, encouraging in this way the filling. This somewhat particular variation of the filling coefficient in the low-speed engine domain, which takes into account the experimental data available for low- and medium-spark ignition engines, influences the performance of the engine. As can be seen from the graphical evolutions presented in figure 1, the developed model leads to values of the mean indicated pressure in accordance with the particular variation of the filling coefficient at the 2000 [rpm] engine speed. On the other hand, for determining the mean indicated pressure of the engine cycle, \( p_i \), this theoretical model does not take into account the change of the excess air coefficient values when changing the engine load. Thus, in real cases, the influence of the variation of the filling coefficient on the behavior of the engine is diminished. This limitation of the theoretical model is diminished in the medium-speed and high-speed engine domains, the graphical evolutions in figures 2, 3 and 4 being consistent with the actual behavior of the propulsion engine in various regimes. The proposed theoretical model establishes also a variation of the exhaust gases coefficient, depending on the engine load. Together with this, a new value of the constants that intervene in the calculation expression of the compression and expansion average polytropic exponent is proposed, dependent on the engine speed.

4. Conclusions

Due to the complex modification of the parameters involved in the calculation of the operating cycle, as well as the lower influence of the engine speed on certain indices, the model mainly takes into account the variation of some factors of influence, variation determined mainly by the engine load and not by the engine speed.

It is considered that the theoretical study and the presented model from this paper are useful in the interaction between the engine and the car, which can be improved through a better estimation of propulsion system performance.

Compared to other papers, developed by authors on this topic, this paper fully addresses the processes from the engine, integrating the burning and expansion processes.

These results lead to the idea of developing and perfecting the model by introducing several factors of influence, thus creating the premises of an experimental validation.
5. References

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