The physically-mathematically model for studying the thermal regime of the spark ignition engine concerning the filling coefficient

T Ulian¹, V Vornicu¹, E Rakosi¹, S Talif¹, I Agape¹
¹"Gheorghe Asachi" Technical University of Iasi, Faculty of Mechanical Engineering, 700050, Romania
uliantudor@yahoo.fr, vladd.alex@yahoo.com, edwardrakosi@yahoo.com, tsorinel@yahoo.com, agapeiulian@gmail.com

Abstract. The study of the development conditions of the intake process from spark plug engine, especially from thermal point of view, highlight complex influences. These influences affect in a decisive manner the energy performance of the engine. In this paper, the authors had done a study and an analysis of the aspects for the purpose to develop a physically-mathematically model which ensures the highlighting of the thermal regime influences of the spark plug engine concerning the efficiency of the filling.

1. Methodology
The study of the engine's thermal regime concerning the filling coefficient indicates that the heating of the fresh charge affects the filling by reducing the filling coefficient values. This analyzation shows that exist a complex mechanism by which the fresh charge, being at the beginning of the intake process at the ambient temperature, come into contact with the warm engine pieces and it is mixed with the residual gases from the combustion chamber [1]. These processes lead to an increase in the initial temperature of the fresh charge and to a significant reduction in the filling coefficient [3].

The modification of the state parameter changes of fresh charge from cylinder can be quantified on analytical way through the theoretical model developed by the authors. In this model was expressed as a distinct function the dependence of the filling degree on the fresh charge temperature at the end of the intake process. In the same idea, another distinct function was introduced to express the dependence of the filling degree on the increase in the temperature of the fresh charge at the contact with the warm surfaces of the engine during the intake process.

Considering that the heat flow Q is generated in the combustion chamber, it propagates to the intake channel through different areas of the cylinder head [2]. For the purpose of analyzing the phenomenon, a theoretical plan model is used to simulate the heat transmission through the walls of the cylinder, bounded by the control volume as shown in figure 1.
Figure 1. Theoretical plan model used to simulate the heat transmission

The equations for the successive transmission of the heat flux density, \( q \), from the combustion chamber through the liquid cooling jacket, lead to the basic relations:

\[
q = \alpha_g \cdot (T_g - T_{\text{chi}})  \quad (1)
\]

\[
q = \frac{\lambda_{Al}}{\delta_1} \cdot (T_{\text{chi}} - T_{p1}) \quad (2)
\]

\[
q = \alpha_l \cdot (T_{p1} - T_l) \quad (3)
\]

where:
\( \alpha_g \) – convective heat transfer coefficient from the combustion gases to the inner wall of the cylinder head;
\( T_g \) – mean cycle temperature of combustion gases;
\( T_{\text{chi}} \) – the temperature of the combustion chamber surface in the engine cylinder head;
\( T_{p1} \) – the surface temperature of the combustion chamber wall in contact with the coolant;
\( T_l \) – the liquid temperature of the cylinder head without heat release to the outside;
\( T_{lr} \) – the temperature of the cooling liquid with heat release to the outside environment;
\( \alpha_l \) – convective thermal transfer coefficient from the cylinder head walls to the coolant;
\( \delta_1 \) – the average wall thickness of the combustion chamber;
\( \delta_2 \) – the average wall thickness of the intake channel in the cylinder head;
\( \lambda_{Al} \) – the conductive heat transfer coefficient of the aluminium alloy of the cylinder head.

Relationship (3) is valid in the situation \( q_r = 0 \), as a particular case which is not found in practice in a real motor. Within this model, taking into account the value of the engine efficiencies, the Relationship (3) is valid in \( q_r = 0 \), as a particular case that is not practicable in a real motor. Within this model, taking into account the value of engine efficiency, the following situation is also interesting:

\( q_r > 0, q_r >> q_{cadm} \)

The balance of heat flux densities is:

\[
q = q_r + q_{cadm} \quad (4)
\]

which by replacement becomes:

\[
q = \alpha_l \cdot (T_{p1} - T_{lr}) = \alpha_{lpr} \cdot (T_{lr} - T_{aer}) + q_{cadm} \quad (5)
\]

where:
\( T_{aer} \) – average outdoor air temperature;
\( q_r \) – the density of the heat flow yielded by the coolant;
\( q_{\text{cadm}} \) – the density of the heat flow transmitted to the intake manifold in the cylinder head;
\( \alpha_{\text{pr}} \) – the convective heat exchange coefficient from the cooling liquid to the reduced isothermal equivalent wall, which releases heat to the outside environment.
This gives the expression of the heat flux density that reaches the intake channel in the cylinder depending on the temperature of the coolant, \( T_{\text{lr}} \):

\[
q_{\text{cadm}} = f(T_{\text{lr}}) \tag{6}
\]

Taking into account the dependence of the temperature of the combustion chamber internal surface from inside the cylinder head on the coolant temperature, experimentally deducted by [6], the authors proposed an analytical expression. This is concretized into a function of the type:

\[
T_{\text{chi}} = A \cdot T_{\text{lr}} + B \tag{7}
\]

The variation of which is illustrated in figure 2.

![Figure 2. Temperature of the combustion chamber internal surface variation](image)

This function provides a more general character for the proposed mathematical model. Determining the values of coefficients \( A \) and \( B \) of this function obtains: \( A = 3.2333 \) and \( B = -606.0333 \). Through development and grouping we obtain successively for \( q_{\text{cadm}} \), the expressions:

\[
q_{\text{cadm}} = \alpha_l \cdot A \cdot T_{\text{lr}} + \alpha_l \cdot B - \alpha_l \cdot \frac{\delta_1}{\lambda_{\text{Al}}} \cdot \alpha_g \cdot T_g + \alpha_l \cdot \alpha_g \cdot \frac{\delta_1}{\lambda_{\text{Al}}} \cdot A \cdot T_{\text{lr}} + \alpha_l \cdot \alpha_g \cdot \frac{\delta_1}{\lambda_{\text{Al}}} \cdot B + \alpha_{\text{ipr}} \cdot T_{\text{aer}} - \alpha_l \cdot T_{\text{lr}} - \alpha_{\text{ipr}} \cdot T_{\text{lr}} \tag{8}
\]

respectively

\[
q_{\text{cadm}} = \left[ \alpha_l \cdot A \cdot \left( 1 + \alpha_g \cdot \frac{\delta_1}{\lambda_{\text{Al}}} \right) - \left( \alpha_l + \alpha_{\text{ipr}} \right) \right] \cdot T_{\text{lr}} - \alpha_l \cdot \alpha_g \cdot \frac{\delta_1}{\lambda_{\text{Al}}} \cdot (T_g - B) + \alpha_{\text{ipr}} \cdot T_{\text{aer}} + \alpha_l \cdot B \tag{9}
\]

Assimilating the inlet channel in the engine head with a tubular flow path having a linear and a curved portion, a convective heat transfer between the wall channel with the \( T_{\text{pcadm}} \) temperature and the fresh charge is considered [4]. The heat transfer from the intake channel wall leads to an average increase in
the temperature of the fresh charge with the $\Delta T$ value. This temperature rise, $\Delta T$, leads, as has been shown above, to a thermal loss at intake due to contact with the hot surfaces of the engine, especially with the inner surface of the intake channel. Thus, the mixture will heat from the temperature $T_0$ to the temperature $T_0' = T_0 + \Delta T$.

As it is known, the particularities of heat transfer by convection influence in this case also the coefficient of convection $\alpha$ which, essentially, quantifies the set of factors defining the process. These factors are mainly identified with the type of motion of the mixture as well as by the shape and orientation of the heat exchange surface.

For the further development of the proposed model, which is in this case the forced convection and the turbulent flow of the mixture through the tubular flow through the cylinder head, a reference relationship, recommended by [5], is used. This relationship is valid in the following areas:

$$7 \cdot 10^3 \leq Re \leq 2 \cdot 10^5$$

and

$$10^{-1} \leq Pr \leq 5 \cdot 10^2$$

and materializes as follows:

$$St = \frac{Nu}{Re} = \frac{Nu}{RePr} = \frac{\frac{\xi}{8}}{1 + \frac{8}{1-Pr} (Pr-1)}$$

In this relation $\Gamma$ represents a constant dependent on the value of the $Pr$ criterion. Equally, the factor $\zeta$ which represents the coefficient of flow resistance of the fresh charge in to the channel occurs in this relation. The heat exchange inside the cylinder head intake path is characterized by the equation:

$$q_{cadm} = \alpha_{pcadm} \cdot (T_{pcadm} - T_0')$$

In this way, for the fresh charge, we get the expression of the temperature increase due to contact with the warm surfaces of the intake channel from inside the cylinder head, $\Delta T$, depending on the coolant head temperature, $T_{lr}$.

Due to the reduced wall thickness of the intake channel we have: $T_{pcadm} = T_p2$ and $\Delta T(T_{lr})$ becomes:

$$\Delta T(T_{lr}) = T_p2 - \left[ \frac{a_l}{a_m} \cdot A \left( 1 + \alpha_g \frac{\delta_1}{\lambda_{Al}} \right) - \alpha_{lpr} \frac{a_l}{a_m} \cdot T_{lr} - \frac{\alpha_{lpr} a_g}{a_m} \cdot \frac{\delta_1}{\lambda_{Al}} \cdot (T_g - B) \right] + \frac{a_{lpr}}{a_m} \cdot T_{aer} + \frac{a_l}{a_m} \cdot B - T_0$$

Finally, the function that allows the calculation of the filling coefficient in dependence on the change in the coolant temperature from inside the cylinder head in accordance with the principle of zonal cooling of the engine (the zonal thermal regime of the engine) is obtained.

$$\eta(T_{lr}) = c_p \cdot c_{tm} \cdot c_e \cdot c_{up2}^{-1} \cdot c_{es} \cdot \left[ T_p2 - \left[ \frac{a_l}{a_m} \cdot A \left( 1 + \alpha_g \frac{\delta_1}{\lambda_{Al}} \right) - \alpha_{lpr} \frac{a_l}{a_m} \cdot T_{lr} - \frac{\alpha_{lpr} a_g}{a_m} \cdot \frac{\delta_1}{\lambda_{Al}} \cdot (T_g - B) \right] \right] + \frac{a_{lpr}}{a_m} \cdot T_{aer} + \frac{a_l}{a_m} \cdot B - T_0$$

in which:

$c_p = \frac{p_2}{p_0}$ – pressures constant;
$c_{tm} = T_0$ – temperatures constant;
$c_e = \frac{\varepsilon}{\varepsilon+1}$ – constructive engine constant;
$c_{up2}^{-1} = 1 - \varphi_{p2} + \gamma_r$ – the second constant that influences the process of filling the cylinder with the fresh charge;
$c_{es} = 1 + \gamma_r$ – the constant that characterized the exhaust gases from the cylinder.
Considering the transfer of heat from the combustion chamber to the coolant and then from it to the intake channel in the cylinder, the proposed model finally determines the dependence of the filling coefficient on the engine's thermal regime through the coolant temperature. In order to attenuate the phenomenon of reducing the filling coefficient value and implicitly the performance of the engine with the heating of the fresh load, with the proposed model, optimization of the engine's thermal regime can be made.

**Figure 3.** Variation of the heat exchange inside the cylinder head intake path

**Figure 4.** Filling coefficient dependence on the coolant temperature
2. Results
The synthetically presented physico-mathematical model assures first of all the emphasis of the heat exchange inside the cylinder head intake path dependence on the variation of the coolant temperature. Thus, the establishment and the graphical representation of this dependence, shown in figure 3, indicate a correlation between coolant temperature and the heat exchange inside the cylinder head, thus highlighting the advantages of a zonal thermal engine. On the other hand, the model proposed by the authors highlights the improvement of the cylinder filling coefficient by reducing the cylinder head thermal regime. This dependence reveals a satisfactory increase of the filling coefficient, as shown in figure 4, in the conditions of an average decrease in the thermal regime of the engine.

3. Conclusions
Through the proposed model in this study it is realizing an analysis of the influence of the engine thermal regime on the cylinder filling phenomenon, based on the variation of the filling coefficient. An original physico-mathematical model is proposed and developed to ensure that this influence is established. The model provides significant results that can be used to improve and to optimize the performance of the spark ignition engine. In this idea, compared to previous studies realized by the authors, this paper contains a physico-mathematical model that is more complex, in which intervene the thermal regime of the engine. We noted that the theoretical model proposed by the authors in this paper does not include the engine load modification.

4. References
[1] Allen D J Lasecki M P 2001 Thermal management evolution and controlled coolant flow SAE paper 2001–01-1732

[2] Alkidas A C 1980 Heat transfer characteristics of a spark – ignition engine J Heat transfer vol. 102(2) pp 189–193

[3] Eberth J F Wagner J R Afshar B A. and Foster R C 2004 Modelling and Validation of Automotive Smart Thermal Management System Architectures SAE paper 2004-01-0048

[4] Incropera F P and DeWitt D P 2002 Introduction to Heat Transfer John Wiley & Sons Inc.

[5] Popa B Aradau D Biris I Iosifescu C 1986 Thermo Technician's Engineering Handbook vol 1 Technical Publishing House Bucharest

[6] Popa B. Bataga N Madarasan T Marinescu M 1978 Thermal Stress in Machinebuilding Technical Publishing House Bucharest