

The simplifying assumptions that were used to define this cycle are the following (see fig. 1b):

- the intake process r-a takes place at a constant pressure p_a , permanently smaller than the environmental one with the value of the pressure losses Δp_a , characteristic to the intake process;
- the exhaust process d-d₁-r takes place in two stages: the free exhaust of the gases at constant volume d-d₁, and the forced exhaust at constant pressure p_r , d₁-r;
- the link between the exhaust valve and the intake one is done through the isentropic expansion of the residual gases r-r₁;
- the compression a-c of the fresh charge is assimilated with a polytropic process with a constant exponent $n_c < k_c$;
- the burning process is schematized in two evolutions: the isochoric one c-z, in which the heat input is Q_v , and the polytropic evolution of exponent $n_u < 1$, that defines the afterburning and in which the working agent receives the heat Q_u ;
- the expansion process starts when the theoretical burning ends in the point u. This process is been assimilated with a polytropic one of exponent $n_{d1} > k_d$ that leads to the give up of the heat Q_{pd} to the cylinder's walls;
- the working fluid is considered to be a perfect gas with a specific heat depending on the temperature;

It must be mentioned that the assimilation of the afterburning process is based on experimental data [2] that show that the final burning temperature T_u is higher than the temperature at the end of the constant volume temperature T_z as a result of the heat release because of the fact that remaining fuel is burned during the phase z-u (fig. 1b). After that, the temperature drops because the increasing of the burning chamber volume and because no heat is released during the process u-d (fig. 1b).

2. The calculus of the temperatures in the characteristic points of the GDI engine cycle

The first step was to calculate the parameters of the theoretical cycle (fig. 1a).

The calculus of the temperature at the end of the intake stroke was presented in a previous paper [7].

Being given the temperature at the end of the intake process $T_a = 322$ K, one can determine the temperature at the end of the compression process:

$$T_c = T_a \cdot \varepsilon^{n_c - 1} \text{ [K]}; \quad (1)$$

The polytropic exponent of the compression process n_c can be expressed depending on the adiabatic exponent k_c and on the coefficient that takes into consideration the heat ceded by the working agent to the cylinder's walls during this process:

$$n_c = 1 - \frac{k_c - 1}{\varphi_c}; \quad \varphi_c = 1.06; \quad (2)$$

where: - φ_c is a coefficient that characterizes the deviation of the working process from the adiabatic compression process;

It's obvious the fact that, because $Q_{pc} < 0$ and $\Delta U > 0$, the coefficient $\varphi_c > 1$, so that $n_c < k_c$. In the conditions in which the compression process takes place, the adiabatic exponent k_c is determined with the following relation:

$$k_c = 1.438 - 0.525 \cdot 10^{-4} \cdot (T_a + T_c); \quad (3)$$

where: - T_c [K] is the temperature of the fresh charge at the end of the compression process;

The pressure at the end of the compression process (point c in fig. 1b) is:

$$p_c = p_a \cdot \varepsilon^{n_c} \text{ [Pa]} \quad (4)$$

One can determine the temperature at the end of the main phase of the burning process (point z in fig. 1b), after the calculus of the pressure at the end of the main phase of the burning process p_z :

$$\frac{p_z}{p_c} = \frac{T_z}{T_c} = \alpha; \quad (5)$$

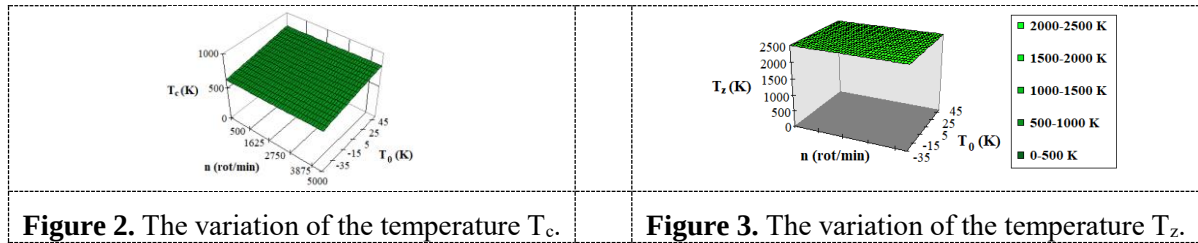
and further: $T_z = \alpha T_c$ [K] and $p_z = \alpha p_c$ [Pa];

where: - p_z [Pa] is the pressure at the end of the main burning phase;

The temperature at the end of the afterburning process (point u in fig. 1b) is:

$$T_u = T_z \delta^{1-n_u} \quad ; \quad n_u = 0,9 \quad [2] \quad (6)$$

In fig. 2 is presented the variation of the temperature at the end of the compression process T_c , depending on the environmental temperature and on the rotational speed for an air-excess factor $\lambda=1$.

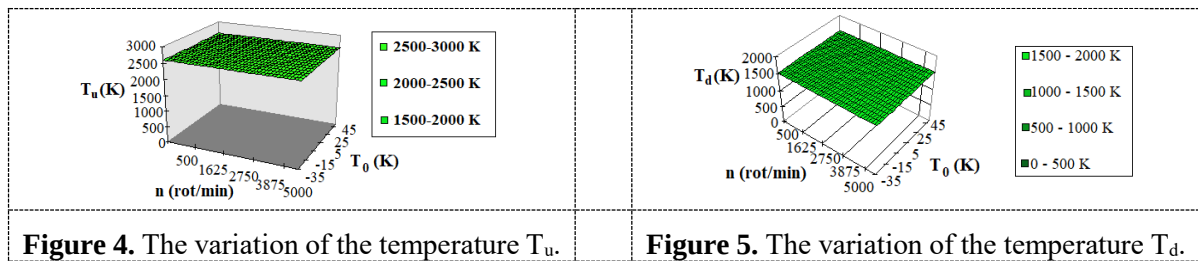


In fig. 3 is presented the variation of the temperature at the end of the main phase of the burning process T_z , depending on the environmental temperature and on the rotational speed for an air-excess factor $\lambda=1$.

The temperature at the end of the burning process (the point u in fig. 1b) is [2]:

$$T_u = T_z \cdot \delta^{1-n_u} \quad [K]; \quad (7)$$

In fig. 4 is presented the variation of the temperature at the end of the afterburning T_u depending on the rotational speed and on the environmental temperature.



The temperature at the end of the expansion process (point d in fig. 1b):

$$T_d = T_u \cdot \left(\frac{\delta}{\varepsilon} \right)^{n_d-1} \quad [K]; \quad (8)$$

In fig. 5 is presented the variation of the temperature at the end of the expansion process T_d depending on the rotational speed and on the environmental temperature.

The temperature at the end of the free exhaust phase (point d_1 in fig. 1b) is:

$$T_{d_1} = T_d \cdot \left[\frac{(\varepsilon / \delta)^{n_d} - \delta^{n_u}}{\psi \cdot \alpha \cdot \varepsilon^{n_c}} \right]^{\frac{k_e - 1}{k_e}} [K]; \quad (9)$$

In fig. 6 is presented the variation of the temperature at the end of the forced exhaust phase T_{d1} depending on the rotational speed and on the environmental temperature.

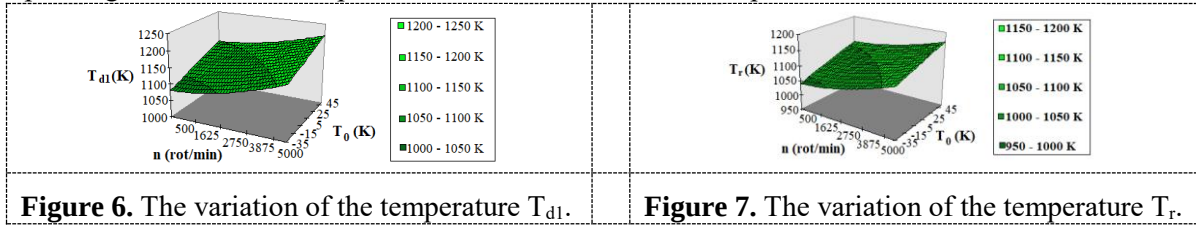


Figure 6. The variation of the temperature T_{d1} .

Figure 7. The variation of the temperature T_r .

The temperature at the end of the exhaust process (point r in fig.1.b) is:

$$T_r = T_{d1} / \varphi_r [K]; \quad (10)$$

In fig. 7 is presented the variation of the temperature at the end of the free exhaust phase T_r depending on the rotational speed and on the environmental temperature.

The temperature at the beginning of the intake process (the point r_1 in fig. 1b):

$$T_{r_1} = T_r \cdot \psi^{k_e - 1/k_e} [K]; \quad (11)$$

In fig. 8 is presented the variation of the temperature at the beginning of the intake process T_{r1} depending on the rotational speed and on the environmental temperature.

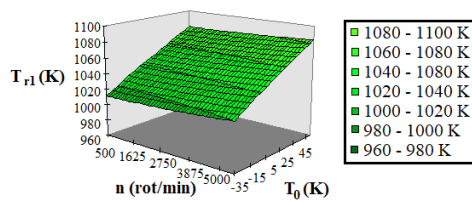


Figure 8. The variation of the temperature T_r

3. Conclusion

The computation results for the temperatures in the characteristic points of the engine's cycle were compared to those experimentally obtained. The experimental data were acquired at the environmental temperature $T_0=293$ K and at the maximum power rotational speed $n=5250$ rot/min.

The calculated temperature at the end of the intake stroke increases as the rotational speed rises. The calculated temperature is $T_a=343.5$ K, and the one determined from the experimental date is $T_{am}=353$ K.

The temperature calculated in the point c is $T_c=762.5$ K, and the determined from the experimental data is $T_{cm}=783$ K. The calculated temperature is with 2.7% smaller than the experimental one.

The temperature calculated in point z is $T_z=2500$ K, and the one determined from the experimental data is $T_{zm}=2547$. The difference is of 1.88% and appears as a result of the fact that the temperature in the point c was underestimated in the calculus.

The temperature calculated in the point u is $T_u=2600$ K, and the one determined from the experimental data is $T_{um}=2680$ K. The error is of 3.077% and appears as a result of the underestimation of the temperature in the point c.

The temperature calculated in the point d is $T_d=1538$ K, and the one determined from the experimental data is $T_{dm}=1564$ K. The error is of 1.7%.

The temperature calculated in the point d1 is $T_{d1}=1193$ K, and the one determined from the experimental data is $T_{d1m}=1220$ K. The error is of 2.26%.

The temperature calculated in this point is $T_r=1147$ K, and the one determined from the experimental data is $T_{rm}=1140$ K. The error is of 0.6%, that is insignificant.

The temperature calculated in the point r1 is $T_{r1}=1147$ K, and the one determined from the experimental data is $T_{r1m}=1064$ K. The error is of 1.5%.

One can observe that the errors are between 0.6 and 3%.

The greatest error is registered in the point u, at the end of the burning process (approx. 3%). That's an excellent value taken into consideration the fact the estimation of the heat exchange process inside the cylinder are very difficult to be estimated. This means that the model proposed by the authors is sufficiently accurate to estimate the temperatures in the characteristic points of the engine's cycle.

In conclusion, the model proposed by the authors, through small calibrations, can be a useful tool for the determination of the maximum pressure of the engine's cycle and can give information about the formation of the polluting emissions.

A good management of the thermal regime of the engine can reduce pollutant emissions [8,9] so, the model proposed in this paper can offer useful information about the thermal regime of the GDI engines.

References

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