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Adrian Irimescu, Dănilă Iorga, Werner Hinkel, Liviu Mihon

The Influence of Air-Fuel Ratio on Mixture Parameters in Port Fuel Injection Engines

Nowadays, research in the internal combustion engine field is focusing on detailed understanding of the processes that take place in certain parts of the aggregate, and can have a great influence on the engine's performance and pollution levels. Such research is developed in this paper, in which using a numerical method based on the i-x air-fuel diagram, one can simulate a series of values for pressure, temperature and intake air humidity before and after mixture formation takes place in a spark ignition engine inlet port. The aim is to evaluate the final temperature of the air-fuel mixture near the inlet valve and evaluating the main factors of influence on the homogeneity of the mixture.

Keywords: *air-fuel ratio, port fuel injection, heat and mass transfer, intake, temperature*

1. Introduction

In spark ignition engines with port fuel injection, the fuel mixes with air in the intake manifold, and the resulting mixture is then burned inside the engine. For a high quality combustion process, the mixture must be homogenous. Complete combustion ensures that the engine is working with minimum carbon monoxide and unburned hydrocarbon compounds being expelled into the atmosphere.

Mixture formation in fuel injection engines is a highly complex process of heat and mass transfer between the fuel and intake air. Gasoline is pulverized with injectors, atomization being the main quality factor. Fuel droplets gradually evaporate as they make contact with the intake air, a process similar to water evaporation. Thus, the idea of using an i-x diagram for air-gasoline mixtures came about (where the fuel is considered a mixture of different ideal hydrocarbon compounds), similar to Mollier's humid air diagram [1].

In the i-x diagram the actual state of the air-fuel mixture can be evaluated using different methods [2], [3]; the mixture line method is the simplest one and easy to use, with an acceptable degree of precision. All these previous methods are a combination of graphic and analytical evaluation procedures which involve drawing segments on the diagram in order to correct the value of mixture enthalpy by eliminating the enthalpy of moisture contained in the intake air. As these segments are sometimes very short in length, inevitable errors occur and precision in mixture temperature reading is lost. Thus a new and completely analytical method of mixture temperature evaluation was developed.

The studies presented in this paper follow this line of research and facilitate obtaining valid results with regard to the influence of initial air temperature, relative humidity and absolute pressure in the intake manifold, on the air-fuel mixture temperature and homogeneity.

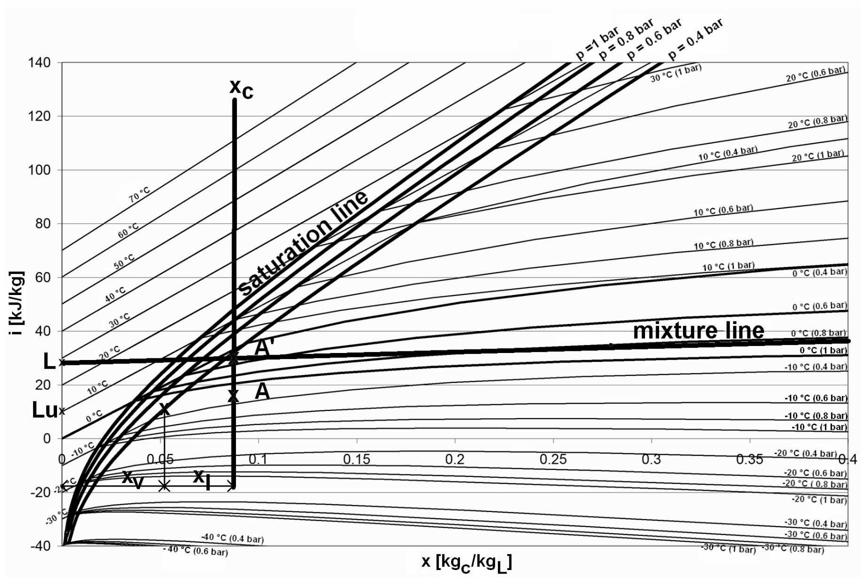

Figure 1. Air-fuel i-x diagram

2. Air-fuel i-x diagram

When plotting the air-fuel i-x diagram for spark ignition engines fueled with gasoline, several hypotheses were used, just as for the Mollier diagram for humid air. The air is considered a mixture of ideal gases because of the low pressure levels and therefore, gas mixture laws of Dalton and Amagat can be used. Gasoline is a mixture of hydrocarbons with different boiling points. An equivalent mixture of three compounds hexane-heptane-octane was used for plotting the i-x diagram.

Air-fuel mixture enthalpy can be calculated with the following equations, in the diagram's non-saturated as well as the saturation domain:

$$i = i_c + i_{L_u}\,, \tag{1}$$

$$i_c = t \cdot (\sum x_{v_i} \cdot c_{pv_i} + \sum x_{l_i} \cdot c_{l_i}) + \sum x_{v_i} \cdot r_i\,, \tag{2}$$

$$i_{L_u} = c_{p_{L_u}} \cdot t\,, \tag{3}$$

where:

i_c – fuel enthalpy;

i_{L_u} – dry air enthalpy;

t – air-fuel mixture temperature;

x_{v_i}, x_{l_i} – fuel mass fraction in the mixture (v-vapor, l-liquid);

c_{pv_i} – fuel vapor specific heat at constant pressure;

c_{l_i} – liquid fuel specific heat;

r_i – fuel enthalpy of vaporization;

$c_{p_{Lu}}$ – dry air specific heat at constant pressure.

In the diagram's nonsaturated domain calculation for determining the fuel mass fraction – enthalpy (x,i) is easily done as no fuel is in liquid state and the molar fractions of the fuel vapor coincide with the molar fractions determined through the distillation curve. On the saturation line (figure 1), the fuel vapor pressure reaches saturation value given by the absolute mixture pressure.

Enthalpy values for the M points on the saturation line are calculated as well as for different constant temperatures lines given by the Q points. The diagram is plotted by linking the N, P points in the non-saturated domain, the M points on the saturation lines and the constant temperature line points Q calculated for different absolute manifold pressure values, 0.4, 0.6, 0.8 and 1 bar. The gap between the constant pressure lines shrinks when the pressure increases, which has an important influence on mixture homogeneity.

3. Methods of mixture temperature calculation in port fuel injection engines

Mixture line method

Air-fuel mixture state near the intake valve is given by the final mixture temperature and the fuel vapor fraction. In order to have a homogeneous mixture, the fuel has to be completely vaporized.

For evaluating the mixture temperature and fuel vapor fraction, the mixture line method was employed. The mixing process takes place in a very short time and can be considered adiabatic. Evaluating the air-fuel mixture state can be done by plotting the "mixture line" from the initial state L (figure 1) of the intake air. This line's gradient is given by the following equation:

$$\frac{\Delta i}{\Delta x_c} = i_c = c_c \cdot t = tg\alpha, \qquad (4)$$

where:
t – temperature of the liquid fuel, before mixing with air;
c_c – liquid fuel specific heat for temperature t;
x_c – fuel mass fraction in the mixture.

Point L_u (figure 1) represents the dry air enthalpy, without any water or fuel vapor, for temperature t_L. By adding the enthalpy of moisture (contained in the air), point L can be plotted, which is correspondent to the state of humid air (i_L), and

Error! Objects cannot be created from editing field codes., (5)

Point A' is given by the intersection of the mixture line with the line for constant fuel mass fraction x_c. In order to obtain the temperature of the mixture after fuel evaporation (t_A), at the corresponding pressure value (p_A), moisture enthalpy (represented by the A'A segment) must be subtracted.

Error! Objects cannot be created from editing field codes., (6)

Once the value for mixture temperature near the inlet valve t_A is obtained (by completing several iterations), the fuel vapor mass fraction (x_v) can be evaluated by graphical means – x_v, as well as the liquid fuel mass fraction x_l are given by the intersection point of the constant temperature line with the saturation line corresponding to the pressure value p_A. The closer the fuel vapor mass fraction is to x_c, the closer the mixture is to a homogeneous state.

Humid air enthalpy (i_w) can be calculated with one of the equations below [9]:

Error! Objects cannot be created from editing field codes., for **Error! Objects cannot be created from editing field codes.** (7)

Error! Objects cannot be created from editing field codes., for **Error! Objects cannot be created from editing field codes.** and *t ≥ 0°C* (8)

Error! Objects cannot be created from editing field codes., for **Error! Objects cannot be created from editing field codes.** and *t < 0°C* (9)

where:
x_w – moisture mass fraction;

r_w – water enthalpy of vaporization;
c_{pw} – water vapor specific heat at constant pressure;
c_w – liquid water specific heat;
l_g – ice water standard enthalpy of fusion;
c_g – ice water specific heat.

The mixture line method requires a large volume of calculations, due to the repeated iterations (up to 15-16) necessary for obtaining a high degree of precision when evaluating the mixture temperature t_A. Results obtained using this method are presented in table 1.

Table 1. Comparative values for different methods

Temperature		
initial air t_L [ºC]	after fuel evaporation t_A [ºC]	
	Mixture line method	Numerical method
-30	-37.4	-37.42
-25	-33.7	-33.66
-20	-29.9	-29.9
-15	-26.4	-26.37
-10	-22.7	-22.71
-5	-19.1	-18.98
0	-15.2	-15.27
5	-11.4	-11.35
10	-7.2	-7.27
15	-3.1	-2.99
20	1.5	1.49
25	7.2	7.19
30	12.6	12.6
35	17.8	17.77
40	22.7	22.75
45	27.5	27.59
50	32.3	32.3

Mixture temperature calculation using the numerical method

For illustrating the numerical method, the humid intake air is considered an open thermodynamic system. The mixing of air with the liquid fuel (figure 2) is considered an adiabatic process and the only variation of this system's enthalpy is due to the added fuel (enthalpy variation will equal the enthalpy of injected fuel).

In most cases, the intake air does not contain any fuel, thus the initial fuel mass participation is null. Also, the gasoline comes from the fuel tank, which has a temperature value equal to that of the ambient air (t_L), and temperature changes along the fuel lines are not significant. It must be noted that the equations of this method allows the use of any liquid fuel temperature.

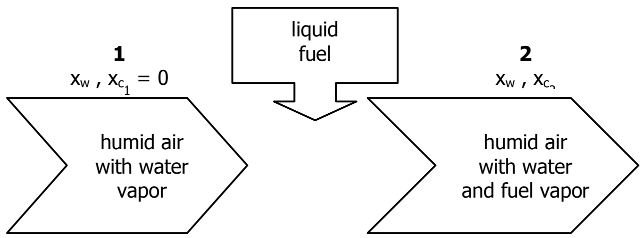

Figure 2. Schematic representation of air-fuel mixture formation

Mixture enthalpy before (i_1) and after the fuel mixes with air (i_2) can be calculated as shown below:

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Error! Objects cannot be created from editing field codes., (11)

Injected liquid fuel enthalpy is given by the equation (12)

Error! Objects cannot be created from editing field codes., (12)

Energy balance equation for x_{c1}=0, x_{c2}=x_c and t_c=t_L will give:

Error! Objects cannot be created from editing field codes., (13)

Combining equations (10, (11) and (13):

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and rearranging the terms with the unknown temperature to the right side:

Error! Objects cannot be created from editing field codes., (15)

This final equation is based on the numerical method used for calculating the mixture temperature t_A after partial or complete fuel evaporation. The terms on the left side of the equation are all known for a given initial air temperature t_L, while dry air and moisture enthalpy can be easily calculated for any mixture temperature with equations (3), (7), (8) and (9). Fuel enthalpy is obtained by linear interpolation between two pairs of (x_c, i_c) values calculated when the i-x diagram was plotted. Thus, the mixture enthalpy can be calculated after the fuel evaporation process takes place (right side of the equation) for different estimated temperature values t_A until equation (15) is true. The results obtained by using either of the two algorithms presented show that both methods are valid.

4. Results and discussions

As previously stated, proper air-fuel mixture homogeneity is a factor of great importance for spark ignition engine performance and reduced pollution levels.

The main factors that influence mixture state are intake air temperature (t_L), relative humidity (φ), manifold absolute pressure (p) and air-fuel ratio (λ).

Pollutant emissions limits and low fuel consumption require spark ignition engines to work with air-fuel ratios as close as possible to the stoechiometric mixture value. However, under certain conditions like acceleration and deceleration, the engine is working with various air-fuel ratios ranging from 10:1 to 19:1 ($0,7<\lambda<1,3$). The numerical method for mixture temperature evaluation allows determining vapor fuel mass fraction for rich as well as lean mixtures. Studies regarding the influence of intake air parameters on mixture homogeneity can be done by programming the numerical method on computers which allows development of research with a very high degree of precision, in a short period of time, over a wide range of air temperatures, humidity and pressure values.

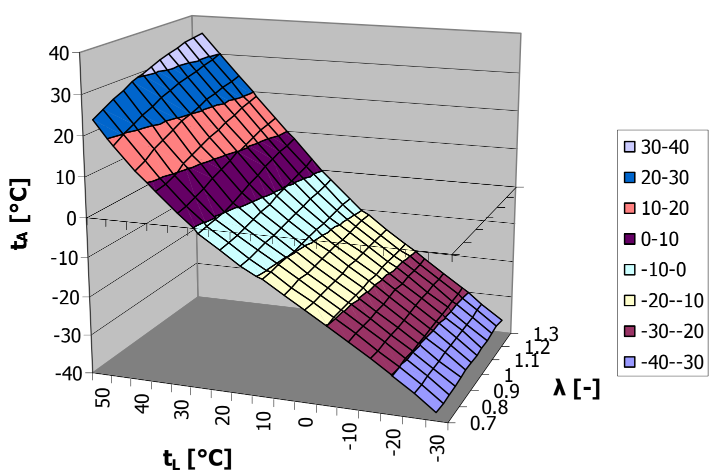

Figure 3. Mixture temperature after fuel evaporation for p=0,4 bar, φ=50 %

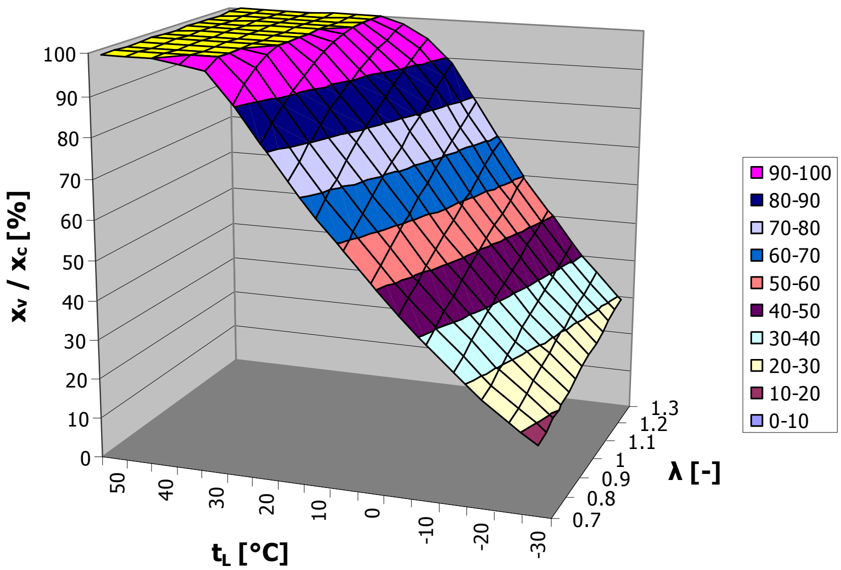

Figure 4. Evaporated fuel ratio for p=0,4 bar, φ=0 %

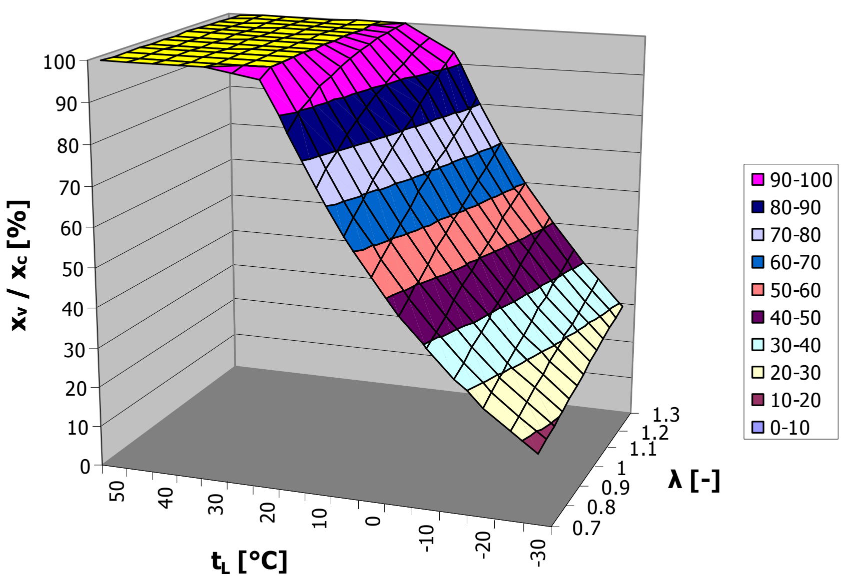

Figure 5. Evaporated fuel ratio for p=0,4 bar, φ=100 %

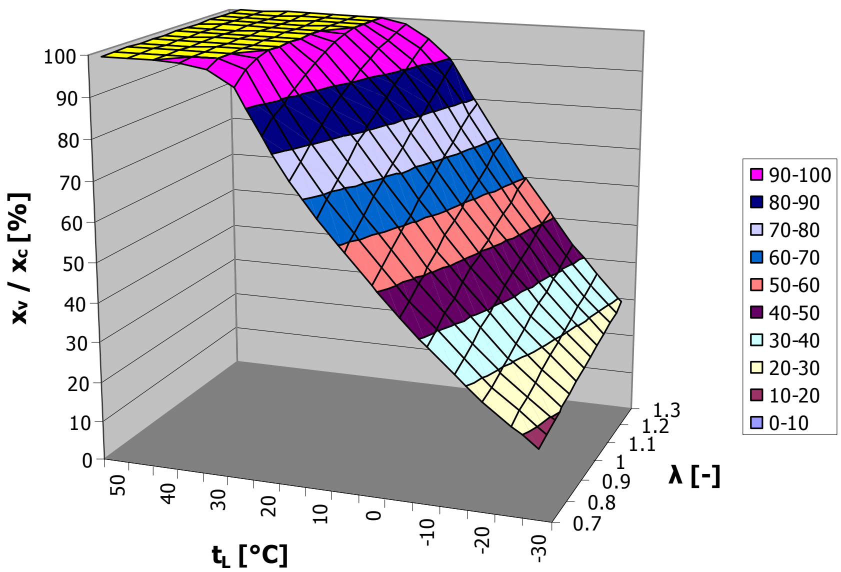

Figure 6. Evaporated fuel ratio for p=0,4 bar, φ=50 %

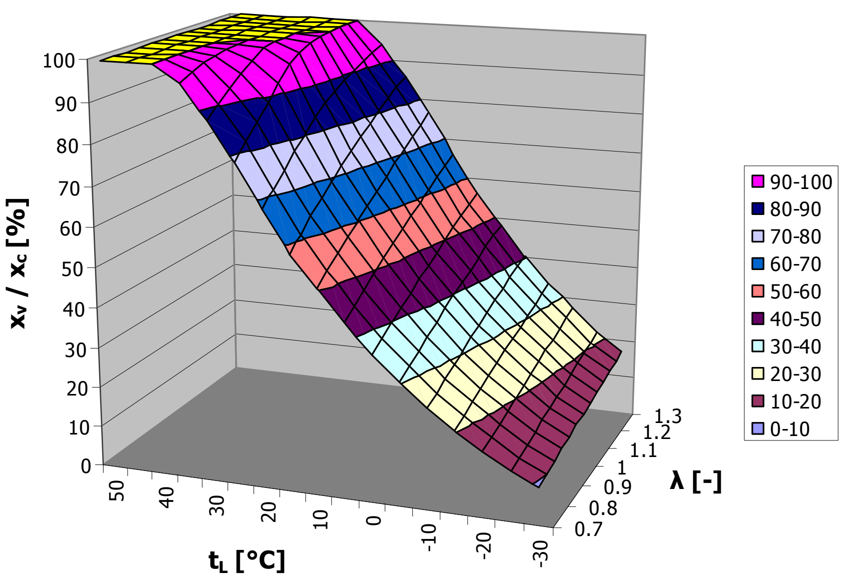

Figure 7. Evaporated fuel ratio for p=1 bar, φ=50 %

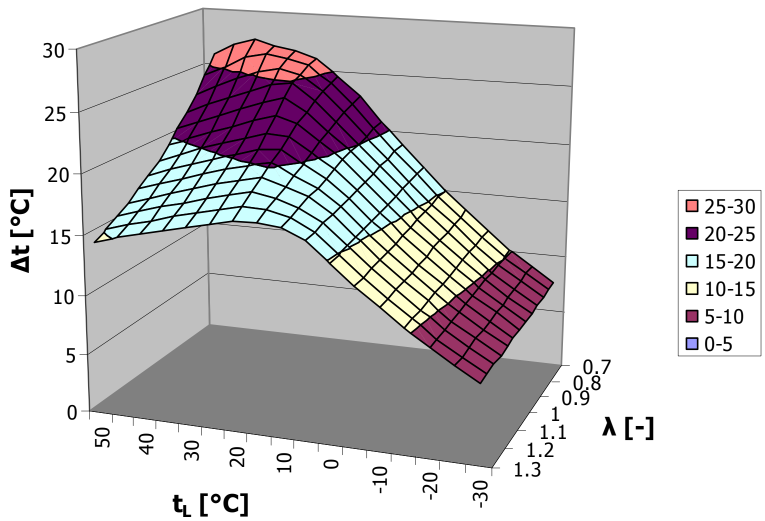

Figure 8. Temperature reduction after fuel evaporation for p=0,4 bar, φ=50 %

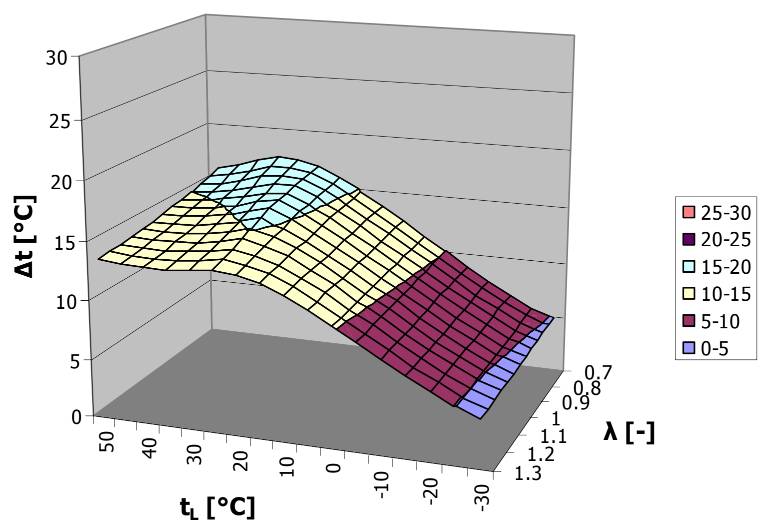

Figure 9. Temperature reduction after fuel evaporation for p=1 bar, φ=50 %

Figure 3 shows the mixture temperature after the fuel is evaporated, for absolute a manifold pressure value of 0,4 bar and relative humidity of 50%, conditions encountered when the engine is idling. The temperature curve is nonlinear and changes shape as the air-fuel ratio increases. A small influence of the air-fuel ratio is observed at low ambient temperatures, while at high temperatures a difference of up to 10% can be observed in the range of 0,7 to 1,3 for λ. As figures 8 and 9 show, the temperature drop due to fuel evaporation is high for pressure levels of 0,4 (when the engine is idling) and lower for 1 bar (wide open throttle operation).

The ratio of evaporated fuel is presented in figures 4, 5, 6 and 7 for the air-fuel ratio range of 0,7 to 1,3 and different pressure and relative humidity levels. Leaner mixtures are closer to homogeneity at any given ambient temperature, and completely homogenous mixtures are obtained above 40°C regardless of other conditions. Obviously, more fuel evaporates when absolute manifold pressure levels are lower (figures 6 and 7).

Temperature variation is very much influenced by the pressure level (figures 8 and 9). When idling, the engine is running with absolute manifold pressure around 0,4 bar and temperature variations are very different on the ambient temperature range (form -30°C to 50°C), while at full throttle (when pressure levels hit 1 bar) the temperature variations are much smoother. Another interesting fact is that when idling, the air-fuel ratio has a major effect on mixture temperature, while at wide open throttle, the quantity of fuel injected has little effect.

The main factor to be considered is that air-fuel mixtures with completely evaporated gasoline can be obtained above 40°C regardless of other intake air parameters and air-fuel ratios in the range of 10:1 to 19:1 ($0,7<\lambda<1,3$). This is one of the main reasons why engine manufacturers ensure proper heating of the intake system (at nominal working engine coolant temperature), even if it means that lower levels of volumetric efficiency are obtained.

5. Conclusions

The numerical method for calculating air-fuel mixture temperature in port fuel injection spark ignition engines is much easier to use and more precise. While it is based upon the equations used for plotting the air-fuel i-x diagram, it does not require any segments being plotted on the diagram.

This new method allows complete studies of all the main factors which influence the mixture temperature after fuel evaporation, like intake air temperature, manifold absolute pressure and relative humidity. Thus the mixture homogeneity can be studied for a wide range of air parameters values.

Using the numerical methods ideal mixture state conditions can be evaluated for obtaining optimum combustion inside the cylinders. It is easily noted that intake air parameters, manifold pressure levels and air-fuel ratio have a major influence and only for a narrow window of values homogenous mixtures are obtained.

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Addresses:

- Ph. D. St. Eng. Adrian Irimescu, "Politehnica" University of Timişoara, Blv. Mihai Viteazu, nr. 1, 300222, Timişoara, iamotors@yahoo.com
- Prof. Dr. Eng. Dănilă Iorga, "Politehnica" University of Timişoara, Blv. Mihai Viteazu, nr. 1, 300222, Timişoara, diorga@mec.upt.ro
- Eng. Werner Hinkel, EWD Gmbh, Estererstraße 12, 84503 Altötting, Germany, info@ewd.de
- Conf. Dr. Eng. Liviu Mihon, "Politehnica" University of Timişoara, Blv. Mihai Viteazu, nr. 1, 300222, Timişoara, mihon@mec.upt.ro