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Estimation of the degree of internal recirculation in Internal Combustion Engines

Evaluación del grado de recirculación interna en motores de combustión interna

Leonid Matiukhin [1*]

[1]Department of Heat Engineering and Internal Combustion Engines Faculty of the Energy and Environmental, Moscow Automobile and Road Construction University, Leningradsky Ave, 64. P. C. 125319. Moscow, Rusia.

ABSTRACT: The concepts of residual gas and admission (or volumetric efficiency) coefficients are used exclusively by specialists in the field of piston Internal Combustion Engines. However, it is preferable to apply the concepts of volume fractions of components of the working mixture consisting of air, fuel, residual, and recirculating gases, for the evaluation of filling. This simplifies and makes it more easy-to-grasp the influence of individual factors on the results of gas exchange processes. The proposed approach makes it possible to take into account the impact on the engine indicators of the molecular weight of the fuel used and the degree of external recirculation, as well as to reduce the number of independent variables. At the same time, the displacement coefficient *A* proposed by the author characterizes a decrease in filling when an engine with external mixing is switched to a gaseous fuel with a lower molecular weight. A change in the valve timing made it possible to produce an effect on the composition of the working mixture and, thereby, the environmental characteristics of the engine. In the case of external recirculation, it becomes necessary to estimate the *summary fraction* of neutral combustion products in the working mixture, on which all engine operation depended. This "overall degree of recirculation" can also be determined using the proposed approach.

RESUMEN: Las nociones de coeficientes de llenado y de gases residuales son utilizadas exclusivamente por especialistas en el campo de los motores de combustión interna de pistón. Sin embargo, se pueden utilizar las nociones técnicas generales y desde hace mucho tiempo conocidas de la termodinámica – es decir las fracciones de volumen de los componentes de la mezcla. Usándolos, es posible evaluar el llenado y la composición de la mezcla de trabajo que consiste en aire, combustible, gases residuales y de recirculación. En este caso, la fracción de gases residuales en el cilindro determina de forma única el grado de recirculación interna. Metodológicamente, el uso de fracciones no solo simplifica, sino que también hace que sea más fácil de interpretar y comprensible de analizar la influencia de factores individuales en los resultados de los procesos de intercambio de gases. Tal enfoque permite tener en cuenta la influencia del peso molecular del combustible utilizado, del grado de recirculación externa y la de la fracción total de productos de combustión neutros en la mezcla de trabajo en los índices de llenado y los de desempeño, así como reducir el número de variables independientes. Al mismo tiempo, el coeficiente de desplazamiento *A* propuesto por el autor caracteriza la disminución del llenado cuando el motor con formación de mezcla externa se transfiere a un combustible gaseoso con un peso molecular más bajo.

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1. Introduction

The systems of external recirculation are now widely used in Internal Combustion Engines. The main alternative to these systems can be the organization of internal recirculation – a change in the amount of residual gases in the working mixture (VM), that is, the air-fuel-residual gas mixture. To analyze its effect on the engine's characteristics, it is necessary to have a criterion that

* Corresponding author: Leonid Matiukhin
E-mail: panam1@mail.ru
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estimates the degree of internal recirculation.
As is known, the external recirculation is estimated by the degree of recirculation R'_c, which is the ratio of the number of kilomoles N_R or mass M_R of recirculation gases to the number of kilomoles N_{NCh} / mass M_{NCh} of new charge (NCh) [1–3]:

$$R'_c = \frac{N_R}{N_{NCh}} = \frac{N_R}{N_{\text{f-a}} + N_R} \tag{1}$$

Or $R_c = \frac{M_R}{M_{NCh}} = \frac{M_R}{M_{f-a}+M_R}$

A new charge is a mixture of air, recirculation gases and (in engines with spark ignition) fuel coming into engine cylinders.
A reciprocal recalculation of the values of R'_c and R_c is possible using expressions [1, 4, 5]:

$$R_c = \frac{\mu_r R'_c}{\mu_{NCh}} \quad \text{and} \quad R'_c = \frac{R_c}{\mu_r}\mu_{NCh}$$

where the μ_r and the μ_{NCh} are the molecular masses of combustion products and a new charge.
In fact, the degree of recirculation is a volumetric / mass fraction [1, 4] of the recirculation gases in the new charge. In this regard, as shown in [6], thermal calculation of the piston engines solely on the basis of volumetric fractions is very convenient. In this case, there is no need to use such traditionally used observables as the volumetric efficiency η_v and the residual gases coefficient γ_r.

Key points

It can be stated that volumetric efficiency assesses not the filling per se, but its deterioration in comparison with some virtual filling in the absence of residual gases, heating of new charge, and hydraulic resistance [5, 7]. At the same time, unrelated coefficients η_v and γ_r, as well as the excess-air coefficient, only indirectly characterize the composition of the working mixture (WM), which determines *all engine properties and performances*. A big drawback of volumetric efficiency is the lack of its permanent ultimate magnitude, which should be sought to achieve maximum power. In addition, it is impossible on the basis of η_v to judge the reserves of filling or loss of new charge as a result of scavenging [7–9]. The volumetric fraction of air σ_{air} or new charge σ_{NCh} in the working mixture, and the estimation the degree of air filling of the cylinder's total volume, are devoid of these deficiencies [6, 10–12].
Because the total volume of the cylinder is equal to the amount of the partial volumes of the components of the working mixture (Figure 1), one can write [10–13]:

$$V_{WM} = V_a = V_r + V_{air} + V_f + V_R$$

Here V_a is the total volume of the cylinder, V_{WM} – the volume of the working mixture, V_i – partial volumes, respectively, of residual gases (RG), air, fuel, and recirculation gases.

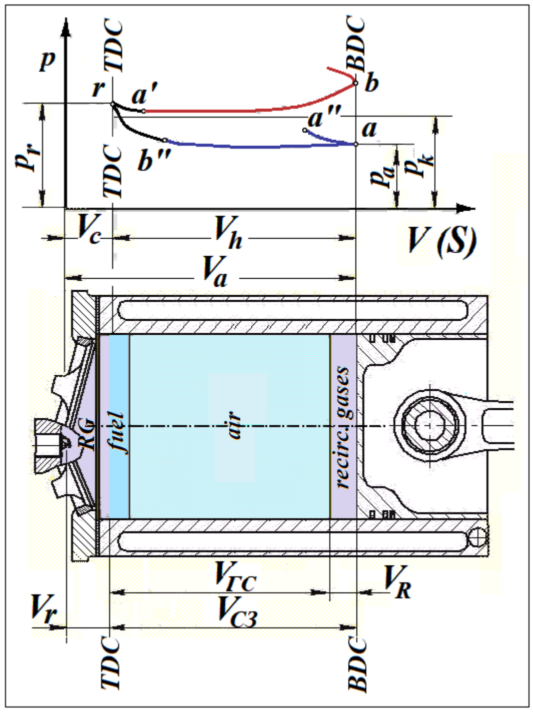

Figure 1 Total cylinder volume as the sum of partial volumes of components of the working mixture

After dividing the given equality by the volume of V_a we get:

$$1 = \sigma_r + \sigma_{air} + \sigma_f + \sigma_R = \sigma_r + \sigma_{NCh} \tag{2}$$

that allows estimating the content of RG in the working mixture by the difference $\sigma_r = 1 - \sigma_{NCh}$, since the sum of the fractions is equal to one.

At the same time, it is obvious that in the absence of RG ($\sigma_r = 0$) the volume of the cylinder is completely filled with a new charge. Therefore, the best to achieve maximum power should be considered a filling corresponding to the $\sigma_{\text{NCh}} = 1$. *The lower values of the σ_{NCh} indicate the availability of filling reserves, and the exceeding unit shows the loss of a part of the new charge as a result of the scavenging during overlapping of valves*[13].

In general, the volumetric fraction of new charge is the sum of air, fuel and recycling gases fractions. In turn, the fraction of the NCh is represented by the sum of

$$\sigma'_{\text{air}} + \sigma'_{\text{f}} + {}'\sigma_{\text{R}} = \sigma'_{\text{f-a}} + \sigma'_R = \sigma'_{\text{f-a}} + R'_c = 1$$

Therefore, the fraction of fuel-air mixture in NCh is equal to the difference

$$\sigma'_{\text{f-a}} = 1 - R'_c$$

In these equations, one stroke in the designation of the fraction corresponds to the composition of the new charge.

Consequently, according to the expression (1), $R_c' = \sigma_R'$.

An approach based on the proportion of volumetric fractions reveals under external mixture formation the effect of the molecular mass of the fuel used on the filling of cylinders with air. Indeed, the excess-air coefficient α (the relationship of the quantity of air involved in the combustion reaction to that theoretically required for the complete combustion of fuel) is the ratio:

$$\alpha = \frac{G_{\text{air}}}{G_f l_0} = \frac{N_{\text{air}}\,\mu_{\text{air}}}{N_f \mu_f l_0} = \frac{N_{\text{air}}}{N_f \mu_f L_0}.$$

Here N_{air} and N_f – are quantities of kilomoles of air and fuel; μ_{air} and μ_f – their molecular mass; l_0 and $L_0 = \frac{l_0}{\mu_{\text{air}}}$ – are the stoichiometric ratios, that are respectively the mass and the number of kilomoles of air needed for complete combustion of 1 kg of fuel. Therefore, one can record the following:

$$\frac{N_{\text{air}}}{N_f} = \alpha \frac{\mu_f}{\mu_{\text{air}}} l_0.$$

But, for ideal gases $\frac{N_{\text{air}}}{N_f} = \frac{V_{\text{air}}}{V_f}$ and $\frac{V_{\text{air}}}{V_f} = \alpha \frac{\mu_f}{\mu_{air}} l_0 = \alpha \mu_f L_0$ On the other hand, after dividing the numerator and the denominator by the total number of kilomoles of the fuel-air mixture N we get $\frac{N_{air}}{N_f} = \frac{N_{air} N}{N_f N} = \frac{\sigma_{air}}{\sigma_f}$ and then:

$$\frac{V_{\text{air}}}{V_f} = \frac{\sigma_{\text{air}}}{\sigma_f} = \alpha \mu_f L_0.$$

Thus, in the case of the stoichiometric mixture, this ratio for gasoline vapors is equal to $\frac{V_{\text{air}}}{V_f} = \alpha \mu_f L_0 = 1 \cdot 113,3 \frac{14.9}{28.9} = 58,4$ and for hydrogen – only $\frac{V_{\text{air}}}{V_f} = 1 \cdot 2 \frac{34.8}{28.9} = 2.4$ (Here $\mu = 113.3$ is the value of the apparent molecular weight of gasoline, corresponding to a mixture of 95% isooctane C_8H_{18} and 5% n-heptane C_7H_{16}). In other words, in the latter case, under *external mixture formation* and the excess-air coefficient α = 1 about a quarter of the cylinder's volume is occupied by hydrogen. At $\alpha = 0.8$,, the partial volume of air, exceeds the volume of hydrogen by less than 2 times, which, with the constant cylinder's volume means a reduction in the volume of air due to a corresponding increase in the fraction of hydrogen. This means that the impact of the type of fuel used on the results of gas exchange must be taken into account [13–20].

As we know from experience, the converting transfer of engines to power gas fuel is accompanied by a change in its performances [21–26], including volumetric efficiency.

In the case of spark ignition engines with external mixture formation, the link between the fractions of the new charge and the air is easily found with the so-called "displacement factor" A [13].

$$A = \frac{\alpha L_0 \mu_f}{\alpha L_0 \mu_f + 1} \tag{3}$$

In the given relation, L_0 is a stechiometric ratio in kilomoles of air per kilogram of fuel, $\alpha-$ excess-air coefficient and $\mu_{\text{f}}-$ fuel's molecular mass. The dimensionless complex A inevitably appears, including when the volumetric efficiency is deduced on the basis of the partial volumes of working mixture's components [13].

If the displacement factor is presented as a $A = \frac{\alpha L_0}{\left(\frac{1}{\mu_f} + \alpha L_0\right)}$, then its physical meaning becomes clear, it is equal to the volumetric fraction of air in the fuel-air mixture. The volumetric fraction of fuel in the fuel-air– , as well as in the working mixture, under the constant value of excess-air coefficient α, increases as the molecular mass of the fuel used decreases.

Similarly, the fraction of air in the fuel-air mixture (i.e., factor A) decreases in the case of enrichment of the fuel mixture (fig. 2). Therefore, it can be concluded that the volumetric (molar) fractions of air and fuel in the fuel-air– and in the working mixture are linked by the excess-air coefficient and the (seeming) molecular fuel mass. Consequently, any change in the excess-air coefficient or switching to fuel with a significantly different (seeming) molecular mass in engines with external mixture formation should affect the fraction of air σ_a in the WM, i.e., the filling. In turn, the average indicated mean pressure is a function of the σ_{air}, which is illustrated by equations [6, 13]:

$$p_i = \frac{\varepsilon}{\varepsilon - 1} \frac{H_u}{l_0} \frac{\eta_i}{\alpha} \sigma_{air} \rho_{\text{air}}^a \quad \text{and} \quad p_i = \frac{1}{\varepsilon - 1} \frac{H_u}{l_0} \eta_i \frac{\varepsilon p_a T_r - p_r T_a \varphi_s}{287 T_r T_a} \frac{A}{\alpha} \left(1 - R'_c\right).$$

Here, $\mathcal{E}$ is the compression ratio, H_u is the lower working combustion heat, η_i is the indicator efficiency, ρ_{air}^a is the air density at point "a" parameter of the indicator diagram and φ_s is the coefficient of purging, which is the ratio of the actual quantity of kilomoles of the RG to the calculated one. The product $\sigma_{air} \rho_{air}^a$ is the density of air at its partial pressure in the working mixture.

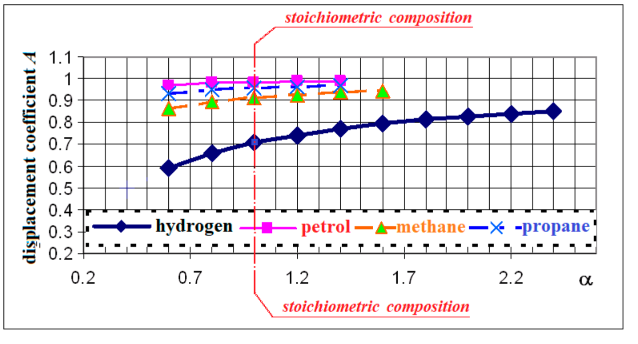

Figure 2 Effect of the excess-air coefficient on the value of displacement factor A [13]

From the last expression, it can be concluded that the efficiency of qualitative regulation of engines with external mixture formation depends on the magnitude of the ratio A/α. The smaller the molecular mass of the fuel, the lower the displacement factor A (3) and the greater the effect of the enrichment of the fuel-air mixture on the reduction of the air fraction in the working mixture (figure 3). As a result, the qualitative power governing in gas engines with spark ignition is less efficient than in gasoline engines [6, 13].

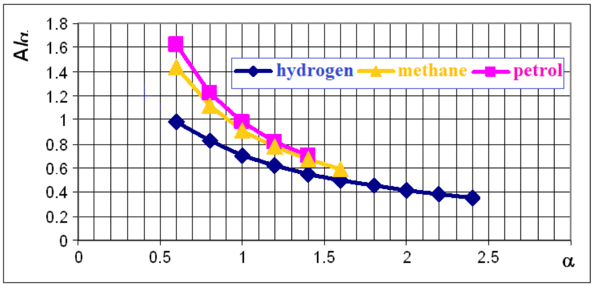

Figure 3 The plot of the effectiveness of quality regulation, which is determined by the ratio of displacement– and excess air factor A/α in the function of the coefficient α[6]

The partial volume of air and its fraction in the air-fuel-residual gases WM are determined by expressions [6, 13]:

$$V_{air} = V_c\left(\frac{\varepsilon p_a T_r - \varphi_s p_r T_a}{p_a T_r}\right) A\left(1 - R'_c\right) \text{ and} \qquad (4)$$
$$\sigma_{air} = \left(\frac{\varepsilon p_a T_r - \varphi_s p_r T_a}{\varepsilon p_a T_r}\right) A\left(1 - R'_c\right)$$

Unlike the expression of the volumetric efficiency η_v traditionally used in calculations (e.g., $\eta_v = \varphi_1 \frac{\varepsilon}{\varepsilon-1}\frac{p_a}{p_o}\frac{T_o}{(T_r+\Delta T+\varphi\gamma_r T_r))}$ [5, 27], the formula deduced to determine the air fraction σ_{air} contains fewer variables. In addition, the value of $\sigma_{\text{air}} = 1$ indicates the maximum possible filling in the absence of loss of fresh charge. The corresponding value of the η_v at which maximum power is reached cannot be called due to its dependence on constantly changing atmospheric conditions (from p_0 and T_0).

In (4) ε is the compression ratio, V_c is the volume of the combustion chamber, the p_a and T_a are the parameters of the working mixture at the beginning of the compression, p_r and T_r – parameters of residual gases at the end of the exhaust.

Since all characteristics of piston ICE are determined by the composition of the ignition working mixture, it is important to know the proportion of individual components obtained during the gas exchange process.

Knowing the value of the air fraction σ_{air} , it is easy to find the volumetric fractions of the remaining components of the working mixture (table 1).

At a known frequency of rotation of the crankshaft and hourly air consumption, its volume in the WM can be determined by the expression

$$\sigma_{\text{air}}^{\text{actual}} = \frac{(\varepsilon-1)}{\varepsilon}\frac{V_{\text{air}}\cdot 10^3}{30\cdot niV_h}\frac{p_k}{p_{a''}}\frac{T_{a}n}{T_k}.$$

The required pressure values (see Fig. 1) are determined by the results of the indication of processes. Volume-based calculations make it easier to obtain equations to determine the number of kilomoles of new charge and recirculation gases [6, 10, 13]. The expression below shows the link between the recirculation degree and the composition of the fuel-air mixture (with the displacement factor A)

$$R'_C = \frac{AN_R}{N_B + AN_R}.$$

From this expression, it follows [6, 13] that, all other things being equal, the convert an engine to another fuel, as well as the variation in excess-air coefficient, are accompanied by a change in the recirculation degree (see fig. 4).

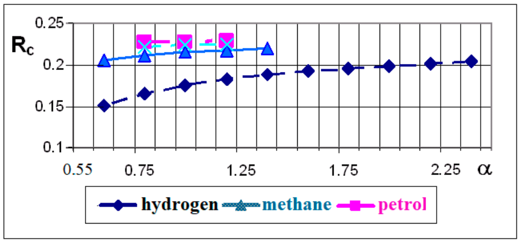

Figure 4 Effect of excess-air coefficient and fuel type (its molecular mass) on the value of recirculation degree R'_c under constant amounts of air and recirculation gases (the values of "N_{aire}" and "N^R" remain unchanged)

2. Estimation of internal recirculation

Currently, engines with variable phases of gas distribution have been widely distributed. By variable valve timing, neutral combustion products can be thrown into the intake tract and therefore increase their fraction in the working mixture. In other words, modern technologies allow for internal recirculation.

The volume of residual gases is determined by the assumption that the working mixture at the beginning of the compression stroke contains the amount of RG equal to their content in the combustion chamber at the time of the piston was found in the top dead center (TDC).

Thus, the partial volume V_r of residual gases can be found by its reduction to the conditions at the point "a" indicator diagram: $\frac{V_c p_r}{T_r} = \frac{V_r p_a}{T_a}$. Hence, the volume of RG at parameters of the point "a" is $V_r = V_c \frac{p_r}{p_a}\frac{T_a}{T_r}$ or, given the coefficient of purging φ_S, is equal to

$$V_r = V_c \frac{p_r}{p_a}\frac{T_a}{T_r}\varphi_s$$

After dividing the partial volume V_r by the volume of the WM, equal to the total volume of the cylinder V_a, we get an

Table 1 Proportions of the components of the working mixture [13]

σ_i	σ_R	$\boldsymbol{\sigma}_{\mathrm{C3}}$	$\boldsymbol{\sigma}_{\Gamma\mathrm{C}}$	$\boldsymbol{\sigma}_{\mathrm{T}}$	σ_{B}	σ_i
$\boldsymbol{\sigma}_{\mathrm{B}}$	$\sigma_R = \frac{R'_c\sigma_{\mathrm{B}}}{A(1-R'_c)}$	$\sigma_{\mathrm{C3}} = \frac{\sigma_{\mathrm{B}}}{A(1-R'_c)}$	$\sigma_{\Gamma\mathrm{C}} = \frac{\sigma_{\mathrm{B}}}{A}$	$\sigma_{\mathrm{T}} = \frac{\sigma_{\mathrm{B}}}{\alpha\mu_{\mathrm{T}}L_0}$	1	σ_{B}
$\boldsymbol{\sigma}_{\mathrm{T}}$	$\sigma_R = \frac{R'_c\alpha L_0\mu_{\mathrm{T}}\sigma_{\mathrm{T}}}{A(1-R'_c)}$	$\sigma_{\mathrm{C3}} = \frac{\sigma_{\mathrm{T}}(1+\alpha L_0\mu_{\mathrm{T}})}{(1-R'_c)}$	$\sigma_{\mathrm{TC}} = \sigma_{\mathrm{T}}\,(1+\alpha L_0\mu_{\mathrm{T}})$	1	$\sigma_{\mathrm{B}} = \alpha L_0\mu_{\mathrm{T}}\sigma_{\mathrm{T}}$	σ_{T}
$\boldsymbol{\sigma}_{\Gamma\mathrm{C}}$	$\sigma_R = \frac{R'_c\sigma_{\Gamma\mathrm{C}}}{(1-R'_c)}$	$\sigma_{\mathrm{C3}} = \frac{\sigma_{\Gamma\mathrm{C}}}{(1-R'_c)}$	1	$\sigma_{\mathrm{T}} = \frac{\sigma_{\Gamma\mathrm{C}}}{(1+\alpha\mu_{\mathrm{T}}L_0)}$	$\sigma_{\mathrm{B}} = A\sigma_{\Gamma\mathrm{C}}$	$\boldsymbol{\sigma}_{\Gamma\mathrm{C}}$
$\boldsymbol{\sigma}_{\mathrm{C3}}$	$\sigma_R = R'_c\sigma_{\mathrm{C3}}$	1	$\sigma_{\Gamma\mathrm{C}} = \sigma_{\mathrm{C3}}\,(1-R'_c)$	$\sigma_{\mathrm{T}} = \frac{\sigma_{\mathrm{C3}}(1-R'_c)}{(1+\alpha L_0\mu_{\mathrm{T}})}$	$\sigma_{\mathrm{z}} = \sigma_{\mathrm{C3}}A\,(1-R'_c)$	$\boldsymbol{\sigma}_{\mathrm{C3}}$
σ_R	1	$\sigma_{\mathrm{C3}} = \frac{\sigma_R}{R'_c}$	$\sigma_{\Gamma\mathrm{C}} = \frac{\sigma_R(1-R'_c)}{R'_c}$	$\sigma_{\mathrm{T}} = \frac{\sigma_R(1-R'_c)}{R'_c(1+\alpha L_0\mu_{\mathrm{T}})}$	$\sigma_{\mathrm{B}} = \frac{A\sigma_R(1-R'_c)}{R'_c}$	σ_R

expression to determine the volumetric fraction of residual gases [13, 28]

$$\sigma_r = \varphi_1\frac{p_rT_a}{\varepsilon p_aT_r} \tag{5}$$

The coefficient of additional charging φ_1 is the ratio of the number of moles of fresh charge actually received in the cylinder to the calculated one. It takes into account the increase in filling as a result of better cleaning during the period of overlapping of valves, as well as due to inertial phenomena during the delay in closing the intake valves.

It follows from the last equation that the fraction σ_r increases when covering the throttle (when the p_a is lowered) and the pressure of residual gases goes up.

As shown in the [24, 25], the additional charging- and purging coefficients [28] are interlinked in the result that the sum of all partial volumes of working mixture components is equal to the total volume of the cylinder V_a, and the sum of their volumetric fractions is equal to unit. The relationship between these coefficients is determined by expressions.

$$\varphi_S = \frac{1-\varphi_1\sigma_{\mathrm{C3}}}{1-\sigma_{\mathrm{C3}}} \text{ and } \varphi_1 = \frac{1-\varphi_s\sigma_r}{1-\sigma_r}$$

It is an expression (5) that can describe the influence of various factors on the quantity of combustion products remaining in the cylinder, i.e., *the volume of residual gases σ_r is the sole estimated criterion for the degree of internal recirculation.* As the temperature T_r increases, RG reduce their density and mass. As a result of cooling by the subsequent mixing with a new charge, there is a decrease in the volume of RG, which results in a slight decrease in the fraction of RG and a corresponding increase in the fraction of the new charge in the working mixture.

According to expression (5), the volumetric fraction of RG in the working mix is proportional to their pressure. In this regard, the increase in pr means an increase in the density and mass of RG, and therefore (as the example suggests in fig. 5), after mixing with a new charge, their fraction in the working mix increases by 0.011, while the fraction of new charge, defined by the expression $\sigma_{\mathrm{NCh}} = 1-\sigma_r$, it decreases by the same value.

As will be recalled, the worsening of clearing is the main way of organizing internal recirculation. By increasing the pressure and organizing the ingress of RG into the inlet

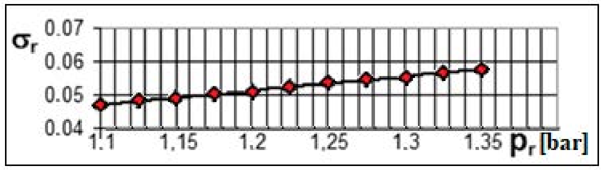

Figure 5 The effect of residual gases pressure p_r [bar] on the fraction σ_r

manifold during the valves overlap period, it is possible to increase the proportion of neutral combustion products in the working mixture in order to reduce the toxicity of exhaust gases.

By influencing the RG parameters at the end of the exhaust process, which appear in the expression (5), and therefore on the parameters of the working mixture at the beginning of the compression stroke, it is possible to change the composition of the working mixture and influence the properties of the engine - environmental, economic, as well as its power [6, 13, 19].

Thus, the thermal calculation of the cycle based on the composition of the working mixture allows evaluating the degree of internal recirculation quantitatively and, because of that, controlling the composition of the working mixture.

As it follows from figure 5, due to the high degrees of compression and the small volumes of the combustion chamber of modern engines, it is impossible to achieve any significant values of the fraction of σ_r only by increasing the pressure of the combustion products in the combustion chamber. In order to implement internal recirculation, it is necessary to organize the backstreaming of residual gas from the exhaust to the intake manifold during the valve overlap period. Re-entering the cylinder and mixing with a fresh charge, these combustion products increase the fraction of RG in the working mixture.

The actual value of the fraction σ_r, at a known value of the new charge fraction σ_{NCh}^{actual}, will be determined by the expression $\sigma_r^{\text{actual}} = 1-\sigma_{NCh}^{\text{actual}}$.

With the known value of air hourly consumption G_{air}, the consumption per minute is $G_{\text{air}}/60[\text{kg/min}]$ and the cycle consumption in grams is $G_{\text{air}}^{\text{cycle}} = \frac{G_{\text{air}}}{0{,}03\text{ in}}[\text{g/cycle}]$. Here «i» is the number of cylinders, and «n» - the frequency of rotation of the crankshaft.

In view of the fact that $G_{\text{air}}^{\text{cycle}} = V_{\text{air}}^{k}\rho_k = V_{\text{air}}^{a}\rho_a$, the volume that air takes up in the cylinder upon the parameters of the working mixture at the point "a" of the indicator chart, is determined by the expression.

$$V_{\text{air}}^{a} = \frac{G_{\text{air}}^{\text{cycle}}}{\rho_a}$$

In this equality $\rho_a = \rho_k \frac{p_a}{p_k}\frac{T_k}{T_a}$ and, therefore,

$$V_{\text{air}}^{a} = \frac{G_{\text{air}}}{0{,}03\text{ in}\,\rho_k}\frac{p_k T_a}{p_a T_k} \quad \text{or} \quad V_{\text{air}}^{a} = \frac{V_{\text{air}}}{0{,}03in}\frac{p_k T_a}{p_a T_k}$$

Here V_{air} is a hour's volume air consumption.
And, after dividing by the volume of the working mixture equal to the total volume of the cylinder V_a, we get the actual value of the air fraction in the working mixture

$$\sigma_{\text{air}}^{\alpha ctual} = \frac{G_{\text{air}}}{0{,}03\rho_k niV_a}\frac{p_k T_a}{p_a T_k}$$

or, more conveniently,

$$\sigma_{\text{air}}^{\alpha ctual} = \frac{V_{\text{air}}}{0{,}03niV_a}\frac{p_k T_a}{p_a T_k}.$$

We use the displacement factor A to determine the fraction of the new charge.

$$\sigma_{NCh}^{\alpha ctual} = \frac{V_{air}}{0{,}03AniV_a}\frac{p_k T_a}{p_a T_k}.$$

Consequently, the total fraction of RG in the cylinder by the beginning of the *compression stroke* will
be determined by the difference $\sigma_r^{\alpha ctual} = 1 - \sigma_{NCh}^{\alpha ctual} = 1 - \frac{V_{\text{air}}}{0{,}03\text{ AniV}_a}\frac{p_k T_a}{p_a T_k}$ or, finally,

$$\sigma_r^{\alpha ctual} = \frac{0{,}03AniV_a p_a T_k - V_{\text{B}} p_k T_a}{0{,}03AniV_a p_a T_k}$$

But the total volume of the cylinder V_a can be represented by displacement volume V_h as $V_a = V_c + V_h = \frac{\varepsilon V_h}{(\varepsilon - 1)}$, because $V_c = \frac{V_h}{(\varepsilon-1)}$. In this case

$$\sigma_r^{\alpha ctual} = \frac{(\varepsilon-1)}{\varepsilon}\frac{0{,}03AniV_a p_a T_k - V_{\text{B}} p_k T_a}{0{,}03AniV_h p_a T_k}$$

where iV_h is the total displacement volume of an engine. The difference in the values of $\sigma_r^{\alpha ctual}$ and σ_r, that is,$\Delta\sigma = \sigma_r^{\alpha ctual} - \sigma_r$, equals the increase in the fraction of residual gases as a result of their backstreaming from the exhaust to the intake manifold. This value, in fact, characterizes a change in the degree of internal recirculation.
The degree of internal recirculation is the volume / molar fraction of RG in the working mixture

$$R'_{ci} = \frac{V_r^{\alpha ctual}}{V_{WM}} = \frac{V_r^{\alpha ctual}}{V_a} = \frac{N_r^{\alpha ctual}}{N} = \sigma_r^{\alpha ctual}$$

and (under the absence of external recirculation) is equivalent to the total fraction of neutral combustion products in the working mixture.
In the case of external recirculation, the total fraction of combustion products in the working mixture is determined by the expression.

$$R'_{c_\Sigma} = \frac{V_r^{\alpha ctual} + V_R}{V_{WM}} = \sigma_r^{\alpha ctual} + \sigma_R = R'_{ci} + R'_c = 1 - (\sigma_{air} + \sigma_f)$$

Thus, from (2) follows:

$$R'_{c_\Sigma} = 1 - (\sigma_{\text{air}} + \sigma_f).$$

Because the combustion process of the working mixture is affected by the value of R'_{c_Σ}, it seems preferable to analyze the environmental, economic, and power performance data of the engine in the function of this quantity.
The author's approach to estimation filling by volume fractions (in contrast to the traditionally based on the concepts of the volumetric efficiency and coefficient of residual gases) equally applies to the calculation of supercharged engines, two-stroke ICE [28], and engines operating on the Miller-Atkinson cycle [29]. It allows deriving equations for calculating the composition of the working mixture, taking into account the actual, rather than geometric, compression ratio.

3. Conclusions

The totality of all WM components determines its composition. All engine performance data are a function of the WM composition. At the same time, the fraction of new charge / air unequivocally characterizes the state of fullness of total cylinder volume V_a, that is, its filling.

To determine the optimal magnitudes of the degree of internal recirculation, a certain quantitative value is needed to estimate this degree. This is the fraction of residual gases in the working mixture. If the volumetric composition of the air-fuel-residual gases mixture is known, the fraction of residual gases is defined as the difference $\sigma_r = 1 - \sigma_{NCh}$.

The displacement factor A, based on the ratio of the partial volumes of the components in the new charge, allows analyzing and considering the effect on the filling of each particular fuel (its molecular mass), which is important in the case of gaseous fuels and, above all, hydrogen. Is it clear from the expression $\sigma_r = 1 - \sigma_{NCh}$, that in order to achieve maximum capacity without loss of the new charge as a result of the scavenging, it is necessary to tend to $\sigma_{NCh} = 1$. At the same time, the magnitude of the fraction of σ_r unequivocally characterizes the reserves for filling, and its negative value indicates that there are losses of part of the new charge as a result of the scavenging.

Basing the thermal calculation of the engine on the composition of the working mixture makes it easier to analyze the impact of any characteristics of the mixture on the main indicators of the cycle and on the degree of internal recirculation estimated by the volumetric fraction of residual gases. An additional advantage of this approach is the opportunity to visualize the effects of different factors on the composition of the working mixture, which is very important for didactic purposes.

4. Declaration of competing interest

I declare that I have no significant competing interests, including financial or non-financial, professional, or personal interests interfering with the full and objective presentation of the work described in this manuscript.

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7. Data Availability Statement

The authors confirm that the data supporting the findings of this study are available within the article [and/or] its supplementary materials.

References

[1] M. G. Shatrov1, L. M. Matyukhin, and V. V. Sinyavski, "An alternative approach to the assessment of internal combustion engine filling and its technical and economic parameters," *International Journal of Emerging Trends in Engineering Research*, vol. 8, no. 16, Jun. 2020. [Online]. Available: http://www.warse.org/IJETER/static/pdf/file/ijeter94862020.pdf

[2] R. van Basshuysen and F. Schäfer, Eds., *Handbuch Verbrennungsmotor*, ser. ATZ/MTZ-Fachbuch. Washington, DC: Springer Fachmedien Wiesbaden, 1964, pp. 32–33.

[3] V. N. Lukanin, *Internal Combustion Engines*. Moscow: Mir, 2005.

[4] M. G. Shatrov, *Automobile Engines*. Moscow, RU: Akademia, 2010.

[5] V. N. Lukanin, *Motores de combustión interna*. Moscú: Mir, 1985.

[6] L. M. Matyukhin, "El método alternativo de la evaluación de calidad de los resultados del intercámbio de gases en los motores de combustión interna," *Revista Ingeniería, Universidad de Carabobo*, vol. 25, no. 1, Ene-Abr 2018. [Online]. Available: https://www.redalyc.org/journal/707/70757668005/70757668005.pdf

[7] B. A. Sharoglazov, M. F. Farafontov, and V. V. Klementiev, *Internal combustion engines: Theory, modeling and process calculation*. Russia: South Ural State University, 2005.

[8] Y. L. Kovylova and D. A. Uglanov, "The influence of various factors on the filling ratio of a piston engine," *Bulletin of the Samara State Aerospace University*, vol. 2, pp. 114–117, 2007.

[9] B. A. Sharoglazov and V. A. Povalyaev, "Estimated quality assessment of fresh charge filling of piston engine cylinders at the design stage," *Bulletin of the National Research South Ural State University*, vol. 23, pp. 20–24, 2008.

[10] A. Jante, *Leitfaden der technischen Thermodynamik*. Germany, LE: Teubner, 1950.

[11] A. P. Baskakov, *Termotecnia*. Moscú: Mir, 1985.

[12] W. Christian, *Volume 1 of Technische Wärmelehre für Studierende des Bergbaus, des Hüttenwesens und der Aufbereitung*. Germany, LE: VEB Dt. Verlag für Grundstoffindustrie, 1966.

[13] L. M. Matiukhin, "Analysis of filling and thermal calculation of ice based on the composition of the working mix," *LAP LAMBERT Academic Publishing GmbH & Co*, 2011. [Online]. Available: http://www.techweb.com/se/index.html

[14] P. Kovalenko, S. N. Devyanin, E. A. Ulyukina, and A. V. Todoriv, "Prospects for the operation of agricultural machinery on compressed natural gas," *Autogas filling complex+ Alternative fuel*, vol. 16, no. 7, pp. 313–315, 2017.

[15] V. V. Sinyavski, I. V. Alekseev, I. Y. Ivanov, S. N. Bogdanov, and Y. V. Trofimenko, "Physical simulation of high-and medium-speed engines powered by natural gas," *Pollution Research*, vol. 36, no. 4, pp. 798–804, 2017.

[16] G. G. Ter-Mkrtichyan, A. M. Saikin, K. E. Karpukhin, A. S. Terenchenko, and Y. G. Ter-Mkrtichyan, "Diesel-to-natural gas engine conversion with lower compression ratio," *Pollution Research*, vol. 36, no. 4, pp. 925–930, 2017.

[17] V. V. Sinyavski, M. G. Shatrov, V. V. Kremnev, and G. Pronchenko, "Forecasting of a boosted locomotive gas diesel engine parameters with one- and two-stage charging systems," *Reports in Mechanical Engineering*, vol. 1, no. 1, Dec. 30, 2020. [Online]. Available: https://doi.org/10.31181/rme200101192s

[18] M. G. Shatrov, V. V. Sinyavski, A. Y. Dunina, I. G. Shishlov, and A. V. Vakulenko, "Forecasting of a boosted locomotive gas diesel engine parameters with one- and two-stage charging systemmethod of conversion of high- and middle-speed diesel engines into gas diesel engineiss," *Facta Universitatis*, vol. 15, no. 3, 2017. [Online]. Available: https://doi.org/10.22190/FUME171004023S

[19] V. I. Erokhov and A. L. Karunin, *Gazodizel'nye avtomobili (konstruktsiya, raschet, ekspluatatsiya) [Gas-diesel cars (design, calculation, operation)*. Russia: Graf-press, 2005.

[20] K. Schmillen, "Nuzung von biogas in gaszundungstrahlmotoren," *MTZ Motortechnische Zeitschrift*, vol. 7, no. 9, pp. 351–357, 1989.

[21] V. A. Markov, S. N. Devyanin, L. I. Bykovskaya, I. G. Markova, and I. N. Afteni, "Biogas is a promising for engines," *Gruzovik Press*, vol. 5, pp. 29–39, 2018.

[22] V. L. Chumakov, S. N. Devyanin, F. N. Bizhaev, and A. V. Kapustin, "Experimental studies to improve the toxic characteristics of gas diesel," in *Readings of academician*. Russia, MO: V. N. Boltinsky, 2021, pp. 104–112.

[23] V. A. Markov, S. Bowen, and S. N. Devyanin, "Emission performance of a diesel engine running on petroleum diesel fuel with different vegetable oil additives," in *2020 International Multi-Conference on Industrial Engineering and Modern Technologies, FarEastCon*, A. Jante, Ed. Germany, B.G: Teubner Verlagsgeselschaft, 1956.

[24] V. L. Chumakov and S. N. Devyanin, "Oxide emissions reduction from combustion control in a diesel engine," *Sel□skohozâjstvennye mašiny i tehnologii*, vol. 15, no. 1, 2021. [Online]. Available: https://doi.org/10.22314/2073-7599-2021-15-1-48-56

[25] V. L. Chumakov, S. N. Devyanin, F. N. Bizhaev, and A. V. Kapustin, "Optimization of gas diesel regulation as a method of improving its environmental characteristics," *Agroinzheneriya*, vol. 4, no. 104, pp. 28–32, 2021.

[26] S. N. Devyanin, V. A. Markov, and A. A. Savastenko, "Use of biogas as a motor fuel," in *Readings of academician V.N. Boltinsky (115 years*

since birth). Collection of articles of the seminar, M. N. Erokhin, Ed. Russia, MO: Megapolis, 2019.

[27] V. I. Erokhov and A. L. Karunin, *Motores de automóvil*. Russia, MO: MIR, 1982.

[28] L. M. Matiukhin and G. G. Ter-Mkrtich'yan, "Thermodynamic fundamentals for calculating the duty cycle of engines with a shortened intake or shortened compression based on an analysis of the composition of the working mixture," *Bulletins of NAMI*, vol. 263, pp. 35–44, 2015.

[29] L. M. Matiukhin, "Evaluation of the results of gas exchange in the 4- and 2-stroke piston ice based on the analysis of the ratios of individual components in the working mixture," *Bulletin of MADI*, vol. 6, pp. 36–41, 2006.