ISSN 2347 - 3983
Volume 8. No. 6, June 2020
International Journal of Emerging Trends in Engineering Research
Available Online at http://www.warse.org/IJETER/static/pdf/file/ijeter94862020.pdf
https://doi.org/10.30534/ijeter/2020/94862020

An Alternative Approach to the Assessment of Internal Combustion Engine Filling and Its Technical and Economic Parameters

Mikhail G. Shatrov[1], Leonid M. Matyukhin[2], Vladimir V. Sinyavski[3], Andrey Yu. Dunin[4]
[1]Moscow Automobile and Road Construction State Technical University (MADI), Russian Federation, dvs@madi.ru
[2]Moscow Automobile and Road Construction State Technical University (MADI), Russian Federation, panam1@mail.ru
[3]Moscow Automobile and Road Construction State Technical University (MADI), Russian Federation, sinvlad@mail.ru
[4]Moscow Automobile and Road Construction State Technical University (MADI), Russian Federation, a.u.dunin@yandex.ru

ABSTRACT

The paper offers the basis of calculating the internal combustion engine duty cycle based on the volume fraction of air in the working mixture. Such an approach makes it possible to take into account the influence of the molecular weight of the fuel used and recirculation ratio on the filling of cylinders and indicator parameters of the engine, reduce the number of independent variables and use the fraction of residual gases in the working mixture to assess the degree of internal recirculation. Comparison of calculated and experimental air consumption of the diesel and gas diesel engines was carried out which confirmed a good accuracy of the results if the proposed engine filling method is implemented.

Key words: IC engine, working mixture, volume fractions, cylinder filling

1. INTRODUCTION

With an infinitely large number of sizes and operating modes of reciprocating internal combustion (IC) engines, a certain idealization was required to study and describe the working cycle which takes place in their cylinders. As a result, the operation of the four-stroke idealized cycle was represented as the sum of the processes realized when the piston moves between the top dead center (TDC) and the bottom dead center (BDC) on the assumption that the intake and exhaust valves open and close at these points.

Such a consideration required the introduction of a number of dimensionless parameters. If refinement coefficients are used, one can transfer to real engines the results of conclusions obtained by analyzing the mentioned model. The main dimensionless quantities include, for example, such simplexes as:

Compression ratio $\varepsilon=V_a/V_c$ (the ratio of the total volume of the cylinder to the volume of the combustion chamber - Figure 1).Indicator efficiency $\eta_i=l_i/Q_c$ equal to the ratio of indicator work (performed by gases as a result of the cycle) to the amount of heat introduced with the fuel with the cycle fuel portion.

Excess air coefficient $\alpha=L/L_0$ (the ratio of the actual number of kilomoles of air participating in the combustion to the minimum amount theoretically required for complete combustion of the cycle fuel portion).

Filling factor $\eta_v=G_{air}/(G_{air \cdot th})$, which is the ratio of the amount of air actually entering the cylinder to its "theoretical" quantity, which could enter the cylinder's displacement volume $V_h=V_a-V_c$if the charge density losses during the cylinder intake process are neglected.

Coefficient of residual gases $\gamma_r=G_r/G_{fc}$- the ratio of the amount of residual gases (G_r) to the amount of fresh charge (G_{fc}) entering the cylinder.

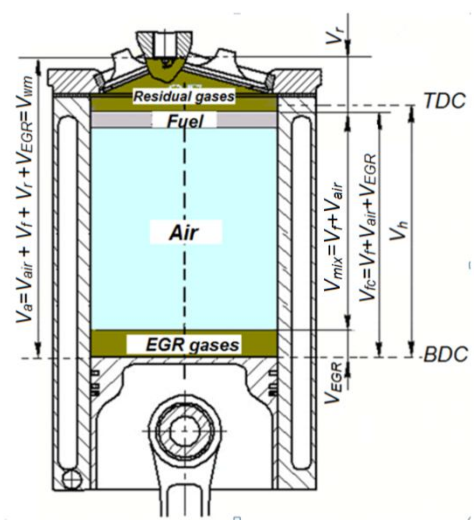

Figure 1: Composition of the working mixture as the sum of the partial volumes of its components (notations of volumes: V_h – displacement volume; V_{air}– air volume; V_f – fuel volume; V_r - residual gases volume; V_{EGR} - recirculation gases volume; V_{mix} = V_{air} + V_f - combustible mixture volume; V_{fc} = V_{air} + V_f + V_{EGR} - fresh charge volume; V_{wm} = V_{air} + V_f + V_r+ V_{REZ} - working mixture volume, equal to the total volume V_{a}

It is noteworthy that, traditionally (for example, in [1]), when assessing the filling, the main cylinder is not the full cylinder V_a, but the piston displacement volumeV_h, which is passed by the piston when it travels between the TDC and BDC. But upon completion of filling and mixing with the exhaust gases, the fresh charge received in the engine cylinders occupies the full, and not the displacemet volume. As a result, basing the calculations on the value of V_h makes it impossible to determine the value of the coefficient η_v corresponding to the optimal filling (and maximum engine power) in case of a perfect cleaning of the cylinders from residual gases. In addition, it is impossible to assess the reserves for improving filling or the fresh charge losses as a result of purging by the value of the coefficient η_v. Another drawback of this approach based onV_h is the lack of correlation between the dimensionless complexes mentioned above. These and other shortcomings of using the filling coefficient as the main evaluation criterion for the quality of engine filling were noted by various authors [2-4].

It is known that power, economic and environmental engine parameters depend on the composition of the working mixture burning in the cylinder, which is a mixture of fresh charge in general consisting of air, fuel (vapors) and recirculation gases (EGR), as well as residual gases. However, due to the basing of the main dimensionless indicators and calculations on the value of V_h, an analysis of the effect of the working mixture composition on engine performance is difficult.

2. METHOD PROPOSED

An analysis, which is based on the total volume of the cylinder is free from the indicated drawbacks. Such an approach makes it possible to evaluate the results of gas exchange of air fractions in the working mixture and to establish the relationship of this fraction with the excess air coefficient α, apparent molecular weight of the fuel, stoichiometric ratio L_0, composition of the combustible air-fuel mixture, and engine performance indicators.

Indeed, at the beginning of compression, when the piston is in the BDC (Figure 1), the total volume of the cylinder is filled with the working mixture, i.e. fuel, recirculation and residual gases. As a result of dividing by the value of $V_a = V_{wm} = V_{air} + V_f + V_{EGR} + V_r$, we obtain the sum of the volume or molar fractions of the components of the working mixture:

$$\sigma_{air} + \sigma_f + \sigma_{EGR} + \sigma_r = 1 \text{ or } \sigma_{fc} + \sigma_r = 1.$$

And if the filling and residual gas coefficients are used only in the calculations of reciprocating IC engines, the fractions are general technical concepts used in all areas of science and technology.

The value σ_{fc}, which estimates the fraction of the total cylinder volume V_a filled by the fresh charge, characterizes the filling itself, and not the degree of its degradation reflected by the coefficientη_v [5]. As can be seen from the last dependence, the maximum filling corresponding to the ideal cleaning of the cylinder from exhaust gasses, under any conditions at the inlet of the engine cylinders and on any modes of its operation, is characterized by the value of the fraction of the fresh charge $\sigma_{fc} = 1$. The values of the fraction $\sigma_{fc} < 1$ indicate the presence of reserves for filling; the values $\sigma_{fc} > 1$ indicate the loss of a part of the fresh charge as a result of purging. It is impossible to judge neither of the filling reserves, neither of the loss of fresh charge during the valves overlap period by the value of the traditionally used filling coefficient η_v.

In general, the value of the fraction of the fresh charge is found by the expression [6], [7], [8]

$$\sigma_{fc} = \frac{\varepsilon p_a T_r - p_r T_a \varphi_s}{\varepsilon p_a T_r} \quad (1)$$

Obviously, the fraction of residual gases in the working mixture is $\sigma_r = 1 - \sigma_{fc}$, and, therefore, with a known value of σ_{fc}, there is no need to introduce an additional quantity to account for the amount of exhaust gas. The value of the fraction σ_r of residual gases in the working mixture is the only evaluation criterion for the degree of internal recirculation.

In traditional assessment of filling by the value of the filling coefficient η_v, the mixture composition is only indirectly characterized by a combination of this coefficient and the coefficient of residual gases γ_r, but there is no unambiguous relationship between these two coefficients. There is also no relationship with the excess air coefficient α, which, unlike $\eta_v$and γ_r, is widely used in calculations of the combustion processes of any heat engineering devices. In addition, despite the influence of the type of fuel used on filling, which is known from experience, expressions for determining the coefficient η_v presented in publications ignore this influence. At the same time, from the definition of the coefficient α as the equality $\alpha = L/L_0$a dependence can be obtained that takes into account the influence on the filling of the excess air coefficient α, the stoichiometric ratio L_0 and the apparent molecular weight of the fuel. Really

$$\alpha = \frac{L}{L_0} = \frac{G_{air}}{l_0 G_f},$$

whereG_{air} and G_f are air and fuel consumption per hour, and l_0 is the minimal amount of air required for complete combustion of 1 kg of fuel. The last expression can be rewritten using the quantities of kilomoles and the corresponding molecular weights

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

After dividing the numerator and denominator by the total number of kilomoles of combustible mixture *N*" (consisting of air and fuel), we obtain

$$\alpha = \frac{\sigma''_{air}\mu_{air}}{l_0 \sigma''_f \mu_f} = \frac{\sigma''_{air}}{L_0 \sigma''_f \mu_f} = \frac{\sigma''_{air}}{L_0 (1 - \sigma''_{air}) \mu_f} \quad (2)$$

In this expression, σ''_i is the fraction of air and fuel in the air-fuel mixture, and $L_0 = l_0/\mu_{air}$ is the number of kilomoles of air theoretically required for the complete combustion of 1 kg of fuel.From the last equation, we have:

$$\alpha L_0 \mu_f = \frac{\sigma''_{air}}{(1-\sigma''_{air})}$$

This is the relationship between the excess air coefficient α, composition of the fuel-air mixture (expressed as $\frac{\sigma''_a}{\sigma''_f} = \frac{\sigma''_a}{1-\sigma''_a}$), stoichiometric ratio L_0 and characterizes the type of fuel with a molecular mass of μ_f. It follows from this relation that the proportion of air in the air-fuel mixture σ''_{air}, as well as in the working mixture, should in all cases depend on the excess air coefficient and the properties of the fuel used. This dependence is reflected by the dimensionless complex, called the “displacement coefficient” *A* [6], [8]. The proposed complex has a clear physical meaning. Indeed, from equality (2) we obtain

$$A = \sigma''_{air} = \frac{\alpha\mu_f L_0}{(\alpha\mu_f L_0 + 1)} = \frac{\alpha L_0}{\left(\alpha L_0 + \frac{1}{\mu_f}\right)},$$

Since for a fixed value of the excess air coefficient α, the power indicators (power, torque, mean effective pressure *et al.*) of the engine depend on the amount of air (oxygen)which remained in the cylinders after the intake valves closure, it is important to determine the amount of air entering the cylinder. For this reason, one should know its fraction, since $G_{\text{air}} = \sigma_{\text{air}} V_a \rho_k$. Here ρ_k is the air density at the engine inlet (before the intake valves). The shares of all other components of the working mixture can be represented through the value of σ_{air} (Table 1). According to the results of the experiment, the value of this fraction is found by the expression presented in [4]-[6]:

$$\sigma_{air} = \frac{(\varepsilon-1)V_a 10^3}{30n\varepsilon i V_h}\frac{p_k}{p_a}\frac{T_a}{T_k}.$$

Table 1: The formulas for the ratios of the components of the working mixture

$\boldsymbol{\sigma_i}$	$\boldsymbol{\sigma_{air}}$	$\boldsymbol{\sigma_f}$	$\boldsymbol{\sigma_{mix}}$	$\boldsymbol{\sigma_{fc}}$	$\boldsymbol{\sigma_{EGR}}$
σ_{air}	1	$\sigma_{\text{air}} = \alpha L_0 \mu_f \sigma_f$	$\sigma_{air} = A\sigma_{mix}$	$\sigma_{air} = \sigma_{fc} A(1-R'_c)$	$\sigma_{air} = \frac{A\sigma_{EGR}(1-R'_c)}{R'_c}$
σ_f	$\sigma_f = \frac{\sigma_{\text{air}}}{\alpha\mu_f L_0}$	1	$\sigma_f = \frac{\sigma_{cm}}{(1+\alpha\mu_f L_0)}$	$\sigma_f = \frac{\sigma_{fc}(1-R'_c)}{(1+\alpha L_0\mu_f)}$	$\sigma_f = \frac{\sigma_{EGR}(1-R'_c)}{R'_c(1+\alpha L_0\mu_f)}$
σ_{mix}	$\sigma_{mix} = \frac{\sigma_{air}}{A}$	$\sigma_{mix} = \sigma_f(1+\alpha L_0\mu_f)$	1	$\sigma_{mix} = \sigma_{air}(1-R'_c)$	$\sigma_{mix} = \frac{\sigma_{EGR}(1-R'_c)}{R'_c}$
σ_{fc}	$\sigma_{fc} = \frac{\sigma_{air}}{A(1-R'_c)}$	$\sigma_{fc} = \frac{\sigma_f(1+\alpha L_0\mu_f)}{(1-R'_c)}$	$\sigma_{fc} = \frac{\sigma_{cm}}{(1-R'_c)}$	1	$\sigma_{fc} = \frac{\sigma_{EGR}}{R'_c}$
σ_{EGR}	$\sigma_{EGR} = \frac{R'_c\sigma_{air}}{A(1-R'_c)}$	$\sigma_{EGR} = \frac{R'_c\alpha L_0\mu_f\sigma_f}{A(1-R'_c)}$	$\sigma_{EGR} = \frac{R'_c\sigma_{cm}}{(1-R'_c)}$	$\sigma_{EGR} = R'_c\sigma_{fc}$	1

The degree of impact of the molecular weight of the fuel on the engine filling process can be illustrated by the example of hydrogen. So, when it is used as a fuel, the number of kilomoles in one kilogram of fuel is 0.5, that is, it exceeds the number of kilomoles of gasoline $\left(N_f = \frac{1}{\mu_f} \approx \frac{1}{110} = 0.009\right)$ by more than 55 times! But, in accordance with the law of Avogadro, the volume of onekilomole of any substance under given thermodynamic conditions is a constant value. Thus, even if we imagine gasoline vapors as an ideal gas, the volume occupied by them will not exceed $22.4 \cdot 0.009 \approx 0.202 \text{m}^3$ under normal physical conditions. In fact, its volume will be significantly smaller. At the same time, hydrogen under the same conditions should occupy a volume of 11.2 m^3. In this regard, the proportion of air in its stoichiometric mixture with hydrogen is significantly lower than in the petrol-air mixture, and this means a decrease in filling by air of the engine cylinders.

The dependence of the displacement coefficient on the molecular mass of the fuel and the excess air coefficient is shown in Figure 2.

3. RESULTS OF CALCULATIONS

Thus, a change in the proportion of air in a combustible air-fuel mixture results in a corresponding change in its share in the working mixture. Enrichment of the mixture means an increase in the fraction of fuel vapor with a decrease in the fraction of air (since $\sigma''_f = 1 - \text{A} = 1 - \sigma''_{air}$).

In case of external mixture formation, the enrichment of the mixture is accompanied by the replacement of a part of thecylinder volume with fuel - gaseous or vaporized liquid. As a result, a decrease in the displacement coefficient *A* unambiguously means a corresponding decrease in the proportion of air in combustible and working mixtures (as well as a decrease in the filling coefficient), which indicates a deterioration of the mass filling and, as a result, leads to a decrease in the power-to-volume ratio.

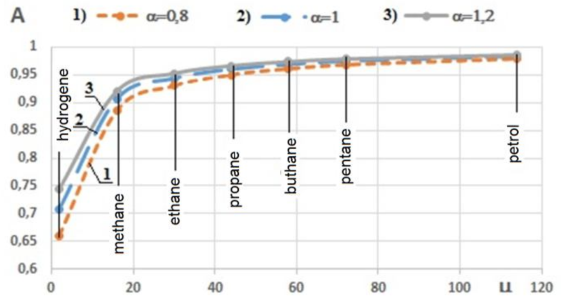

Figure 2: The influence of the molecular mass of the fuel and the air excess coefficient on the displacement coefficient *A*

Since the exhaust gas recirculation rate is their volume fraction in the fresh charge $\left(R'_c = \frac{N_{EGR}}{N_{mix}+N_{EGR}} = \sigma'_{EGR}\right)$, consisting of the air-fuel mixture and neutral combustion products, their introduction into the intakesystem also should lead to a decrease in the amount of air entering the cylinder [9, 10]. As a result ([4], [10]), the fraction of air in the working mixture decreases, since the difference $(1 - R'_c)$represents the fraction of the air-fuel mixture in the fresh charge. In this case, the product $M = A(1 - R'_c)$characterizes a certain total displacement coefficient, which allows to find the fraction of air in the working mixture

$$\sigma_{air} = \sigma_{fc}A(1 - R'_c) = M\sigma_{fc}.$$

The analysis shows that the degree of recirculation depends on the displacement coefficient

$$R'_c = \frac{N_{EGR}}{N_{fc} + N_{EGR}} = \frac{N_{EGR}}{\frac{N_{air}}{A} + N_{EGR}} or$$

$$R'_c = \frac{AN_{EGR}}{N_{air}+AN_{EGR}} \tag{3}$$

It follows from this equality that, with a constant number of kilomoles of recirculation gases and air, transfer to a lighter fuel (which is accompanied by a decrease in the displacement coefficient *A*, all other things being equal, leads to a decrease in the degree of recirculation (Figure3). Accordingly, the enrichment of the mixture accompanied by a decrease in the displacement coefficient at constant values of N_{EGR} and $N_a$should also lead to a decrease in the degree of recirculation. The reason for such a decrease is an increase in both cases of the number of fuel moles$N_f$in a fresh charge, which results in an increase in the denominator in (3).

The number of moles of fresh charge can be determined by the universal dependence [13]

$$N_1 = \left(\alpha L_0 + \frac{1}{\mu_f}\right)\left(\frac{1}{1 - R'_c}\right).$$

The real molecular change coefficient μ_{real}characterizes the increase in the number of kilomoles (or the volume of the working mixture) as a result of combustion and is found from the expression indicated in [4]

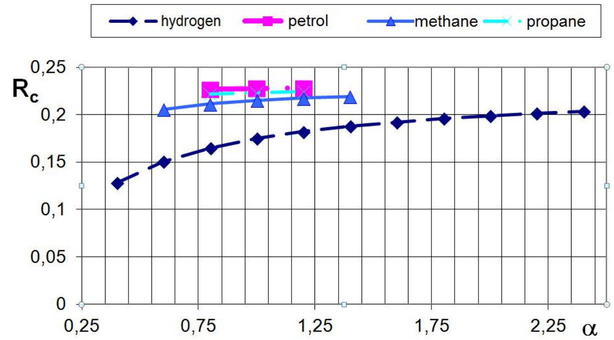

Figure 3: The effect of the fuel used and the excess air coefficient on the degree of recirculation

$$\mu_{real} = \frac{\varphi_1 N_2 + \varphi_1 N_{real} + \varphi_s N_r}{\varphi_1 N_{mix} + \varphi_1 N_{real} + \varphi_s N_r}$$

In the above expression, φ_1 is the additional charge coefficient and φ_s is the cleaning coefficient. After dividing the numerator and denominator by the number of kilomoles of combustible mixture we get:

$$\mu_{real} = \varphi_1\mu_0\sigma_{mix} + \sigma_r^{\Sigma} \text{or} \mu_{real} = \mu_0\sigma_{mix}^{\text{real}} + \sigma_r^{\Sigma},$$

since in this case the denominator turns into a unit

$$\varphi_1\sigma_{mix} + \varphi_1\sigma_{EGR} + \varphi_s\sigma_r = 1.$$

Here μ_0 is the theoretical coefficient of molecular change, taking into account the change in the number of kilomoles of combustible air-fuel mixture as a result of combustion, and$\sigma_r^{\Sigma} = \sigma_{EGR}+\sigma_r$.It is the total fraction of combustion products. Thus, the actual coefficient of molecular change is numerically equal to the sum of the shares of neutral combustion products in the working mixture and the fraction of the combustible mixture increased by μ_0 times. Since the sum of the fractions of the components of the working mixture is always equal to unity ($\sigma_{cm}^{real} + \sigma_r^{\Sigma} = 1$), the value of the actual coefficient of molecular change is $\mu_{real} > 1$. As shown in [6], [12], in the case of assessing the filling of air shares, the average indicator pressure is determined by the expression

$$p_i = \frac{\varepsilon}{\varepsilon - 1}\frac{H_u}{l_0}\frac{\eta_i}{\alpha}\sigma_{air}\rho_{air},$$

explicitly taking into account the effect of the compression ratio on p_i. This dependence can also be represented as

$$p_i = \frac{\eta_i\rho_k}{\varepsilon - 1}\frac{H_u}{l_0}\frac{\eta_i}{\alpha}\frac{\varphi_1(\varepsilon p_a T_r - p_r T_a)}{p_k T_r}\frac{T_k}{T_a}\frac{A}{\alpha}(1 - R'_c).$$

The last expression, in particular, shows the dependence of p_i on the A/αratio, which characterizes the effectiveness of a quality power control in the case of external mixture formation [4, 13]. The value $\frac{A}{\alpha} = \frac{L_0}{\left(\alpha L_0+\frac{1}{\mu_f}\right)}$is equal to the ratio of the number of kilomoles of air, theoretically required for complete combustion of 1 kg of fuel, to the actual number of moles of a combustible mixture consisting of 1 kg of fuel and used to burn air.

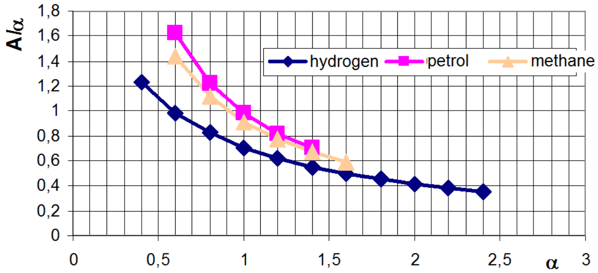

Figure 4: The dependence of the ratio A/α on the value of the excess air coefficient α

As follows from Figure4, the lighter is the fuel used, the less efficient is quality control of power in the case of external mixture formation.The specific effective fuel consumption and indicator efficiency can be found using the dependencies indicated in [14]-[16].From the above formula, for finding the average indicator pressure, it follows that the engine power indices cannot depend on the degree of recirculation R'_c and the displacement coefficient A determined by the type of fuel. A decrease in the latter when switching to lighter grades of fuel should lead to a decrease of the value of $p_i$and, consequently, to a deterioration in engine power indicators. This drop is successfully compensated by the use of boost [17].

$$\eta_i = \frac{1}{H_u \sigma_f}\frac{\mu_{air}}{\mu_f}\frac{(\varepsilon-1)}{\varepsilon}\frac{p_i}{\rho_a} \text{and} g_i = 3600\frac{\varepsilon}{\varepsilon-1}\frac{\sigma_f \mu_f}{\mu_{air}}\frac{\rho_a}{p_i}.$$

It follows from the above expressions that the indicator efficiency $\eta_i$increases, and the specific indicator fuel consumption $g_i$decreases when the engine is switched to lighter fuel, that is, as is known from experience, the economic performance of the engine improves.

It should be noted that using the proposed approach to the analysis of gas exchange processes, a universal dependence was obtained for calculating the traditionally used excess air coefficient, both in the presence and absence of recirculation [17]. Using this technique, it was also convenient to analyze the effect of the pre-opening/delayed closing of the intake valves on the efficiency of engines with a shortened intake/compression [18].

4. EXPERIMENTAL RESULTS

In Figure5, the load characteristic of the Cummins 6.7l gas diesel engine which was converted from the base diesel engine in MADIis presented [19]. The use of specially developed gas supply and electronic engine control systems jointly with the Common Rail diesel fuel supply system enabled to get a very high substitution of diesel fuel by natural gas. Here the following parameters are presented: p_s – boost air pressure, p_t– pressure before the turbine, η_e – effective efficiency, G_d – diesel fuel consumption, G_g – gas fuel consumption, α – excess air efficiency, G_a – air consumption, T_s – boost air temperature.The gas fuel consumption decreased smoothly as the load decreased. The diesel fuel consumption decreased to the value 0.85 kg/h which corresponded to the fuel rate 2 mg/cycle and further remained constant. The reason is that the fuel rate 2 mg/cycle was the lowest value which could be reliably ensured by the injectors used. In addition to experimental curves, modeling results are shown. Calculations were carried out using a one-zone model of MADI [20, 21]. In Figure6, the curves of the air consumption of diesel and gas diesel versions of the Cummins 6.7l engine by the load characteristic are presented.

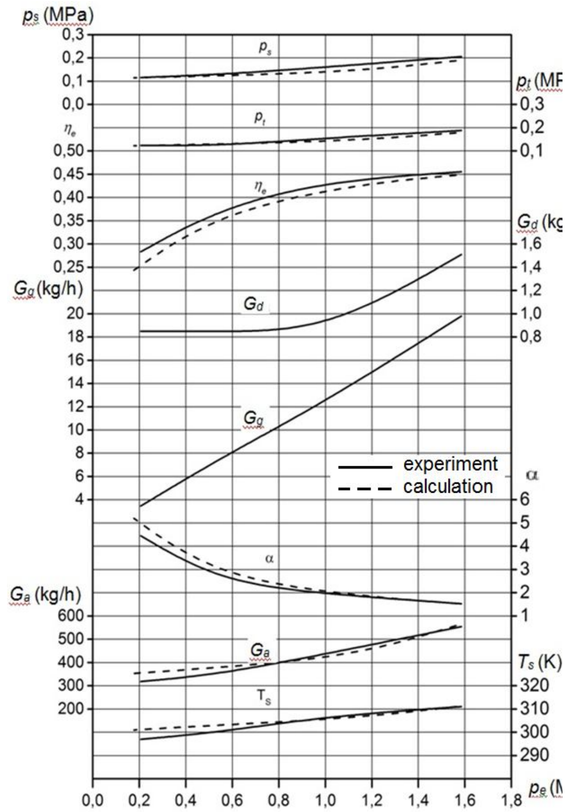

Figure 5: Load characteristic of the Cummins 6.7l gas diesel engine at n=1420 rpm

5. DISCUSSION

As seen from Figure 5, the results of the calculations demonstrate a high accuracy of the model. At full load, the experimental and calculated parameters practically coincide, but at low loads, the difference reaches 3-5% which may be partly explained by not very good description of experimental compressor and turbine maps used in the model in the area of the engine low load operation. In Figure 6, it is seen that the gas diesel engine air consumption is by 4-7% lower than that of the base diesel engine due to the substitution of a portion of intake air by natural gas. It correlates pretty well with calculations by formula (3) showing how much the air consumption G_{air} is lower than the working mixture (air-gas mixture) consumption G_{wm}. Formula (3) is based on the universal dependence [13] and fractions of working mixture offered:

$$G_{air} = G_{wm}\left(\frac{\alpha L_0}{\mu_{air}} + \frac{1}{\mu_f}\right) \quad (3)$$

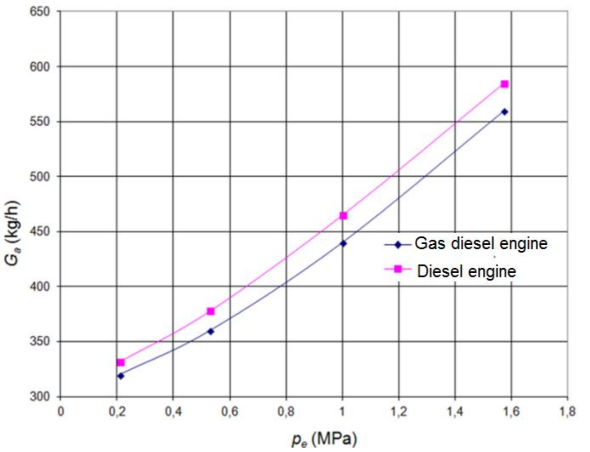

Figure 6: Experimental comparison of air consumption of the diesel and gas diesel versions of the Cummins 6.7*l* engine by the load characteristic at *n*=1420 rpm

Calculations by formula (3) show that the values of the air consumption of the diesel engine and the gas-air mixture consumption of the gas diesel engine are close to the experimental values 4-7% mentioned above especially at high loads where the percentage of diesel fuel is pretty low (about 6%). At low loads, the percentage of the diesel fuel increases and of natural gas – decreases. Therefore, less air is substituted by gas and the difference between the air consumption and gas-air consumption becomes lower.

6. CONCLUSION

The main importance of the present work is using the fraction of air/fresh charge in the working mixture as the basis for calculating the working cycle of reciprocating IC engines. This enables us to obtain dependences that quantitatively evaluate the effect of the molecular weight of the fuel used on the main engine indicators. The fractions of the remaining components of the working mixture are expressed in terms of the fraction of air, which makes it possible to determine the composition of the working mixture, which influences on all the IC engine parameters.

A possible future work will be the comparison of experimental and calculated parameters of the filling processes of more IC engines operating at different modes using the presented method.

REFERENCES

[1]G. Küntscher and Volkmar .(1989). ***Kraftfahrzeugmotoren − Auslegung und Konstruktion*** .VEBVerlag Technik, 720 p.

[2] Yu.L. Kovylov, D.A. Uglanov. **The influence of various factors on the filling ratio of a piston engine.** *Bulletin of the Samara State Aerospace University n.a. S.P.Korolev (VestnikSamarskogoaerokosmicheskogouniversitetaim . S.P.Koroleva*), Vol. 2, pp. 114-117, 2007

[3] B.A. Sharoglazov, 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 (VestnikYuznoUralskogogosudarstvennogouniversitet a*), Vol. 23, pp. 20-24, 2008.

[4] L.M. Matyukhin. **Analysis of filling and thermal calculation of IC engines based on the composition of the working mixture**.Saarbrücken: LAP LAMBERT Academic Publishing GmbH & Co. KG, 2011, 170 p.

[5] L.M. Matyukhin. **Features of the use of filling coefficients "by air" and "by mixture" to evaluate the results of gas exchange in gas engines**.*Transport on Alternative Fuel (Transport naAlternativnomToplive)*, Vol. 14 (40), pp. 38-46, 2014.

[6] L.M. Matyukhin. **The alternative method of the estimate of the quality of gas–exchange processes in the internal combustion engine,***RevistaIngenieria UC*, Vol. 25(1), pp. 31-43, 2018.

[7] L.M. Matyukhin. **Assessment of filling and the basis for calculating the working cycle of a piston IC engine based on the proportion of air in the working mixture.** *Science and Technique of Transport (Naukaitehnikatransporta)*, Vol. 2, pp. 8-16, 2018.

[8] L.M. Matyukhin. **The concept of displacement coefficient and optimal filling**, *Electronic journal Avtomobili-Dorogi-Infrastruktura*, 1(11), p. 22, 2017.
adi.madi.ru/index.php./madi/article/view/3217

[9] L.M. Matyukhin. **Thermodynamic fundamentals of accounting for recirculation in thermal calculation**, *Bulletin of MADI (VestnikMADI)*, Vol. 2 (21), pp. 10-14, 2010.

[10] L.M. Matyukhin. **Features of the assessment of filling the volume fraction of air in the working mixture,** *(Science-Production-Society Naukograd (Nauka-proizvodstvo-obshchestvoNaukograd).*Vol. 2(16), pp. 64-69, 2018.

[11] L.M. Matyukhin. **Evaluation of results for the gas exchange processes by the use of volumetric ratios of air-fuel-residual gases-mixture**. *Science and Education, in Proc. VIII International Research and Practice Conf.* Vela-Verlag, Waldkraiburg–Munich, 2015, pp. 337-345

[12] L.M. Matyukhin, **Universal expression for determining the number of moles of fresh charge.***Electronic journal Automobiles-Roads-Infrastructure (Avtomobili-Dorogi-Infrastruktura)*, Vol. 2(4), 2015.Retrieved December 20, 2015 from: adi.madi.ru/index.php./madi/article/view/3217

[13] L.M Matyukhin. **The dependence of the indicator parameters of IC engines on the type of fuel used.** *Bulletin of MADI (VestnikMADI)*, Vol. 4 (27), pp. 26-29, 2011.

[14] L.M. Matyukhin. **Dependence of indicator indicators of internal combustion engines on the type of fuel used. Assessment of filling and indicator indicators of gas IC engines.** *Vehicle Gas Filling Complex + Alternative Fuel*

(AvtogazozapravochnyyKompleks + Al'ternativnoyetoplivo), Vol. 1(61), pp. 15-18, 2012.

[15] L.M. Matyukhin. **Preliminary forecasting of engine performance when it is converted to gaseous fuel, as well as when using recirculation.***Vehicle Gas Filling Complex + Alternative Fuel (AvtogazozapravochnyyKompleks + Al'ternativnoyetoplivo)*, Vol. 4 (73), pp. 14-18, 2013.

[16] L.M. Matyukhin. **The universal formula for the fill efficiency of a four-stroke IC engine,***Bulletin of MADI (VestnikMADI)*, Vol. 3(22), pp. 39-43, 2010.

[17] L.M. Matyukhin. **Thermodynamic principles of alternative assessment of filling and analysis of piston IC engine cycles.***Truck (Gruzovik)*, Vol. 4, pp. 24-33, 2018.

[18] L.M. Matyukhin, 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,***Bulletin of NAMI (VestnikNAMI)*, Vol. 263, pp. 35-44, 2015.

[19] M.G. Shatrov, V.V. Sinyavski, A.Yu.Dunin, I.G. Shishlov, A.V. Vakulenko. **Method of conversion of high- and middle-speed diesel engines into gas diesel engines.** *FactaUniversitatis, Series: Mechanical Engineering*. Vol. 15(3), pp. 383-395, 2017.
https://doi.org/10.22190/FUME171004023S

[20] V.V. Sinyavski, I.V. Alekseev, I.Ye.Ivanov., S.N. Bogdanov, Yu.V. Trofimenko. **Physical Simulation of High- and Medium-Speed Engines Powered by Natural Gas.***Pollution Research*, Vol. 36 (3), pp. 684-690, 2017.

[21] A. S. Khatchijan, V.V. Sinyavskiy, I.G. Shishlov, D.M. Karpov.**Modeling of parameters and characteristics of engines fed with natural gas**,*Transport on Alternative Fuel (Transport naAlternativnomToplive)*, 3 (15), pp. 14-19, 2010.