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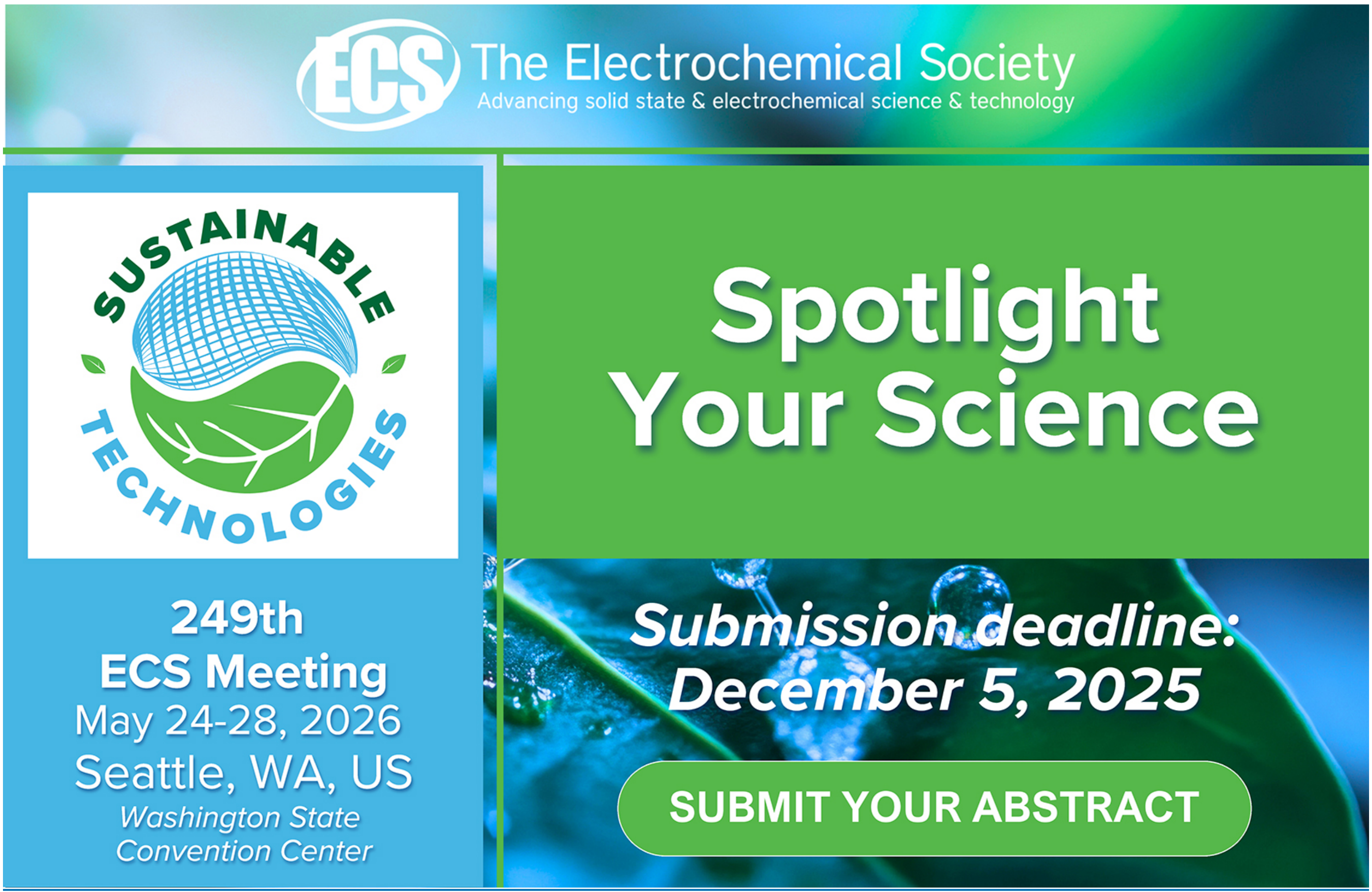

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Flow characteristics of gas injectors

Zdravko Ivanov[1], Stoyan Stoyanov[1], Veselin Mihaylov[1,3] and Helder Santos[2]

[1]Technical University of Varna, Department "Transport Engineering and Technologies", 9010 Varna, Bulgaria
[2]Polytechnic of Leiria, Department "Automotive Engineering", Leiria, Portugal

[3] E-mail: v_mihaylov@tu-varna.bg

Abstract. The operation of Internal Combustion Engine (ICE) with gaseous fuels is characterized by the specificity of the fuelling process. Nowadays the most common system is the one in which after lowering its pressure the gas fuel is delivered to the engine intake manifold by means of gas injectors. This leads to a difference in the laws of fuel supply relative to the original gasoline injection system that dispenses fuel in liquid phase. Theoretical research has been done and gas leakage has been found to be critical under engine normal operating conditions, that is, it depends on fuel pressure and not on the environment in which the fuel leaks (intake manifold). An experimental test stand was set-up and multiparametric characteristics of a common construction bottom feed gas injectors were determined experimentally under conditions very close to the real working ones.

1. Introduction

The fuelling process for LPG and CNG engines has features related to the delivery of fuel in gaseous phase. The injection is carried out by electromagnetic injectors – figure 1. The movement of the needle of the injector is seen as a dynamic system involving gas, inertial, elastic and electromagnetic forces – their balance in time determines its position [1, 2]. As the pressure at the outlet of the injector is variable (the pressure in the intake manifold), to ensure the normal operation of the injector, especially constant opening time, it is necessary to guarantee a constant differential pressure upstream and downstream of the injector.

The main features of gas fuel as compared to liquid fuel are, on the one hand, gas compressibility and, on the other hand, gas leakage is critical, that is, at constant upstream pressure it depends only on that pressure and not on the pressure in the environment in which it flows.

The accessible literature lacks information on the characteristics of gas injectors under engine working conditions. In most cases the characteristics are given as volume flow rate with air, in nl/h, at upstream pressure of 2 bar(a) and downstream pressure of 1 bar(a), that is differential pressure of 1bar (in the paper "bar(a)" stands for unit of absolute pressure). The changing pressure at the intake manifold, which is around 300÷1000 mbar(a), during engine operation and the constant differential pressure, ensured by the reducer, leads to change of the upstream pressure at the injector and thus fuel mass flow changes eventually with change in the manifold pressure.

2. Theoretical background

2.1. Leakage of gas through an orifice

The process of ideal adiabatic gas leakage through a tapering orifice (such as the orifice used in the gas fuel injectors), coupled to a reservoir, depends on the ratio $\beta = P_2 / P_1$. The initial parameters – pressure,

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specific volume and temperature, of the upstream gas are P_1, v_1, T_1, and the downstream pressure is P_2. There is a value of this ratio β_c where the velocity of fluid flow is equal to the local sound velocity, and the pressure in the nozzle opening reaches P_c value, below which it cannot fall [3, 4]. This pressure is called critical pressure, and the leakage of gas through an orifice can either be subcritical, at $\beta > \beta_c$ or at most critical (choked) at $\beta \leq \beta_c$ (figure 2). This means that in a critical leak, the mass of the fluid passing through the orifice will only depend on pressure P_1 and will not depend on pressure P_2 of the environment in which it flows.

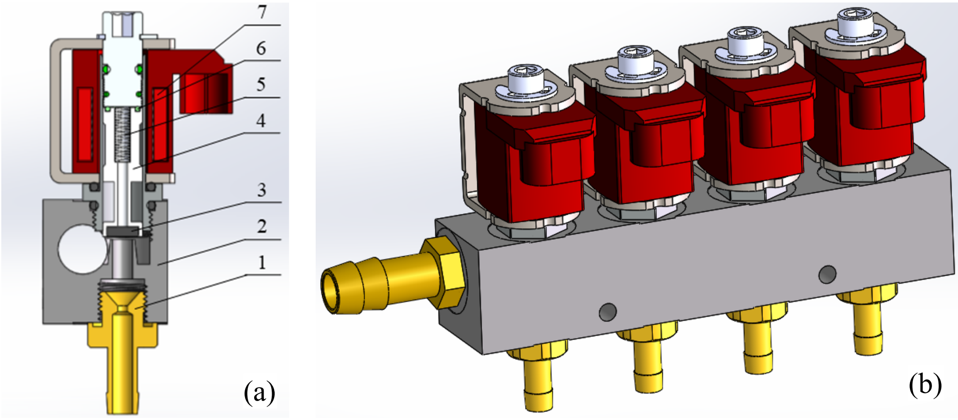

Figure 1. Section view of an injector (a) and injector assembly for 4 cylinder engine (b): 1 – Nipple with calibrated orifice; 2 – Fuel rail; 3 – Rubber insert; 4 – Needle; 5 – Spring; 6 – O-ring; 7 – Coil.

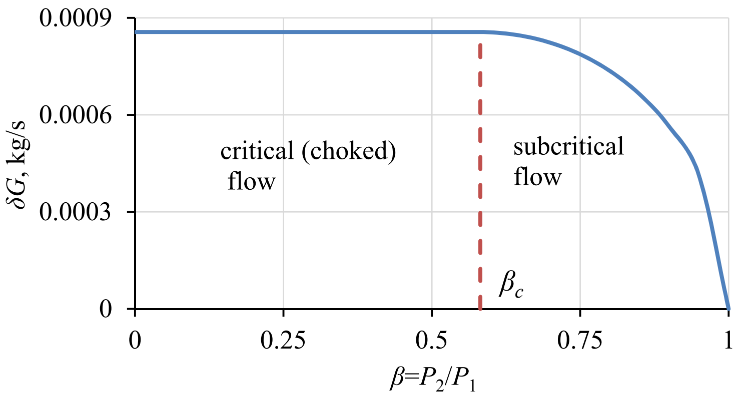

Figure 2. Leakage of gas through an orifice.

$$P_c = P_1 \, \beta_c; \tag{1}$$

$$\beta_c = \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}. \tag{2}$$

The ratio of critical pressure at the orifice exit $P_2 = P_c$ to the pressure in front of the nozzle P_1 is a constant value, depending only on the adiabatic process parameter κ, that is from the nature of the working fluid. For a mixture of 50% propane and 50% butane at 25 °C, κ= 1.1136, β_c = 0.5818.

For engine gaseous fuel delivery systems where the absolute pressure in the intake manifold is in the range of 0.3 ÷ 1.0 bar(a) and the difference with the fuel pressure is constant (about 1 bar), the ratio β is less than β_c and leakage through electromagnetic injector is always critical – figure 3.

In this case, the theoretical mass flow rate ∂G, kg/s, is determined by the following relation:

$$\partial G = A \, [\kappa \, \rho_1 \, P_1 \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{\kappa-1}}]^{1/2}, \tag{3}$$

where ∂G is theoretical mass flow rate, kg/s; A – orifice cross-sectional area, m^2; P_1 – absolute upstream total pressure of the gas, Pa; ρ_1 – density of the gas at pressure P_1 and temperature T_1 , kg/m³; κ – adiabatic index, $\kappa = \frac{C_p}{C_v}$, where c_p is specific heat of the gas at constant pressure and c_v is specific heat of the gas at constant volume.

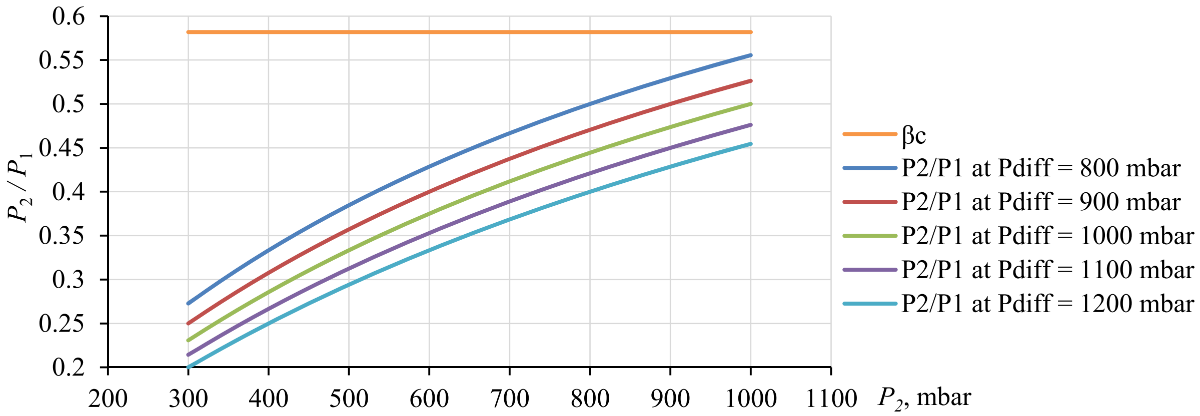

Figure 3. Ratio of P_2/P_1 at different settings of the differential pressure at the reducer.

Formula (3) refers to a theoretical leakage and does not take into account the influence of the valve construction. For this purpose, a discharge coefficient C_d is introduced which indicates the ratio of the actual mass fuel flow rate to the theoretically determined one. Thus the formula to estimate the flow rate becomes (4), where C_d – discharge coefficient:

$$\partial G = C_d\, A\, [k\, \rho_1\, P_1 \left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}]^{1/2}. \qquad (4)$$

The exact values of the discharge coefficient C_d can be determined experimentally.

2.2. Flow simulations

Computational fluid dynamics (CFD) is a branch of fluid mechanics that uses numerical analysis and data structures to analyse and solve problems that involve fluid flows. It can be used in various applications to predict the thermal behaviour, aerodynamic performance, the gas flow through or around a structure [5-7].

Simulations were conducted to study the effect of aerodynamic resistance of the construction on the gas flow. The results show that structural elements affect the leakage of the fuel. The pressure drop of the flowing fuel through the injector is determined by its internal aerodynamic drag and the presence of a calibrated orifice at the outlet nipple.

For the given bottom feed fuel injector design with stroke h = 0.45 mm, calibrated orifice diameter d = 2 mm, parametric simulations were performed at downstream pressure of 300 mbar to 1000 mbar abs., at differential pressure p_{diff} = 1 bar, figure 4 (the coil is not shown). The internal resistance of the injector is characterized by the transition area between points 1-2, figure 5. This drop in pressure is less than the drop in the pressure at transition area 2-3 at the exit of the injector.

The simulations performed show that for this design the main resistance of the injector is at the orifice at the injector outlet nipple.

3. Experimental set-up

To determine the actual characteristics, a non-engine test stand is used. It simulates the actual fuelling conditions in the injector. The stand makes it possible to explore the fuel delivery and to study the operation of the gas injector at different pressure values and at different pressures of the environment in which the fuel leak occurs [1, 8-12]. The stand is shown in figure 6.

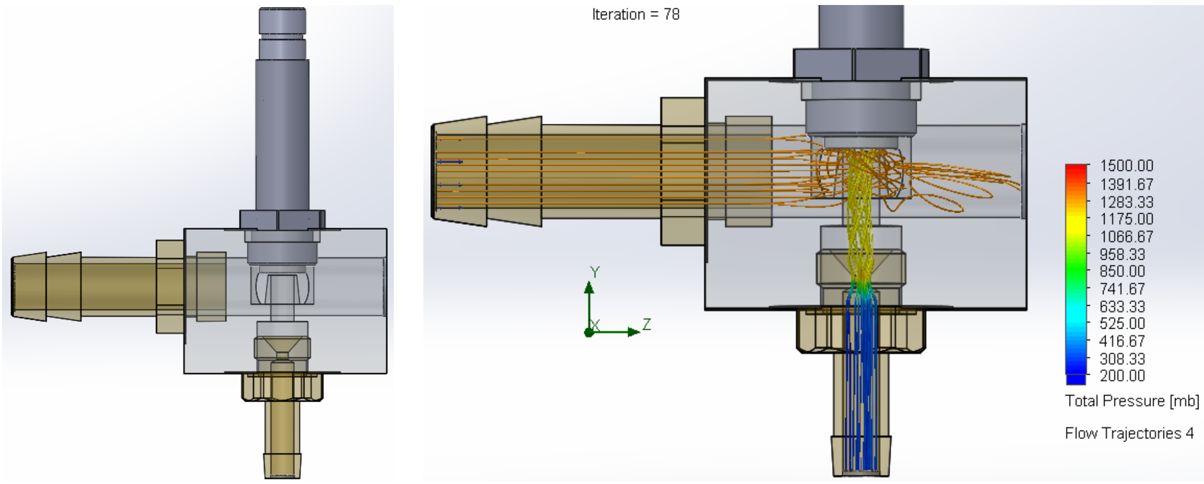

Figure 4. Flow trajectories and pressure distribution at boundary conditions – upstream pressure 1300 mbar and downstream pressure 300 mbar (absolute).

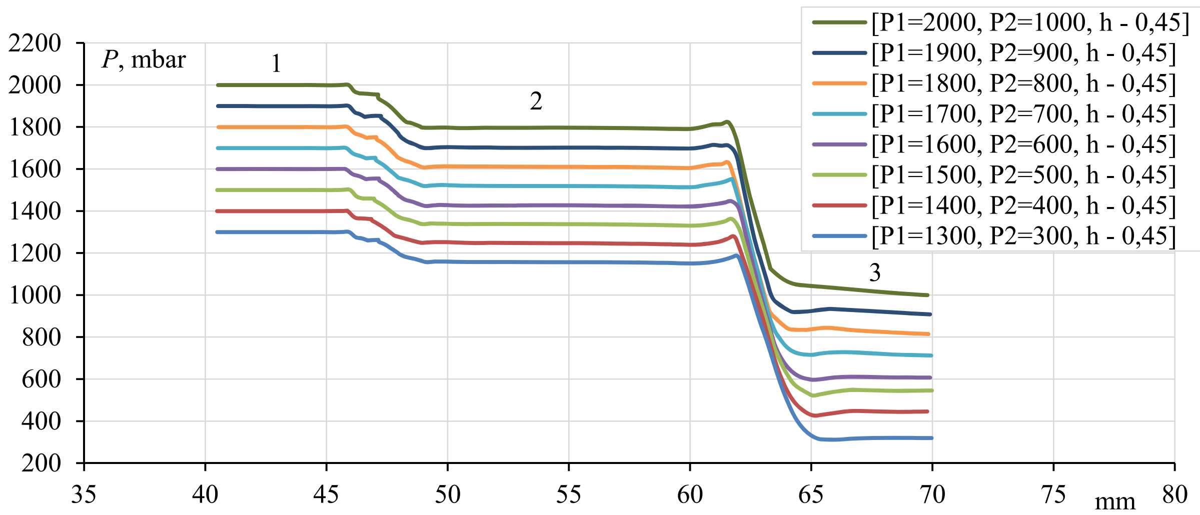

Figure 5. Pressure distribution along the fluid trajectory.

4. Results and discussions

The change of mass of injected fuel per cycle of electromagnetic injector of the discussed type is examined. The diameter of the calibrated orifice at the exit of the injector, which is defining for the fuel leakage, has a diameter $d = 2$ mm. At the inlet and outlet of the injector a constant differential pressure $p_{diff} = 1$ bar is maintained, provided by the reducer balancing system. The fuel temperature passing through the injector is maintained within the operating range of the engine.

The characteristics of the injected fuel per cycle q_c, mg, at different pressures in the engine intake manifold and for different duration of the control electric impulse are given in figure 7. In this case, the experimental results outlined in the graph are processed with first degree polynomials.

The results in figure 7 show the dependence of the injected fuel per cycle on the pressure at the intake manifold at constant duration of the control pulse. With a pulse duration of 10 ms, the injected fuel per cycle q_c increases from 9.4 mg at 300 mbar to 14.6 mg at 1000 mbar, representing an increase in the cyclic portion value of about 55%. Accordingly, for a duration of 30 ms, the increase is from 29.5 mg at 360 mbar to 46 mg at 1000 mbar, or 56% increase of fuel delivery for the same duration of the control signal. The change for intermediate values of the duration of the control pulse are shown in the graph.

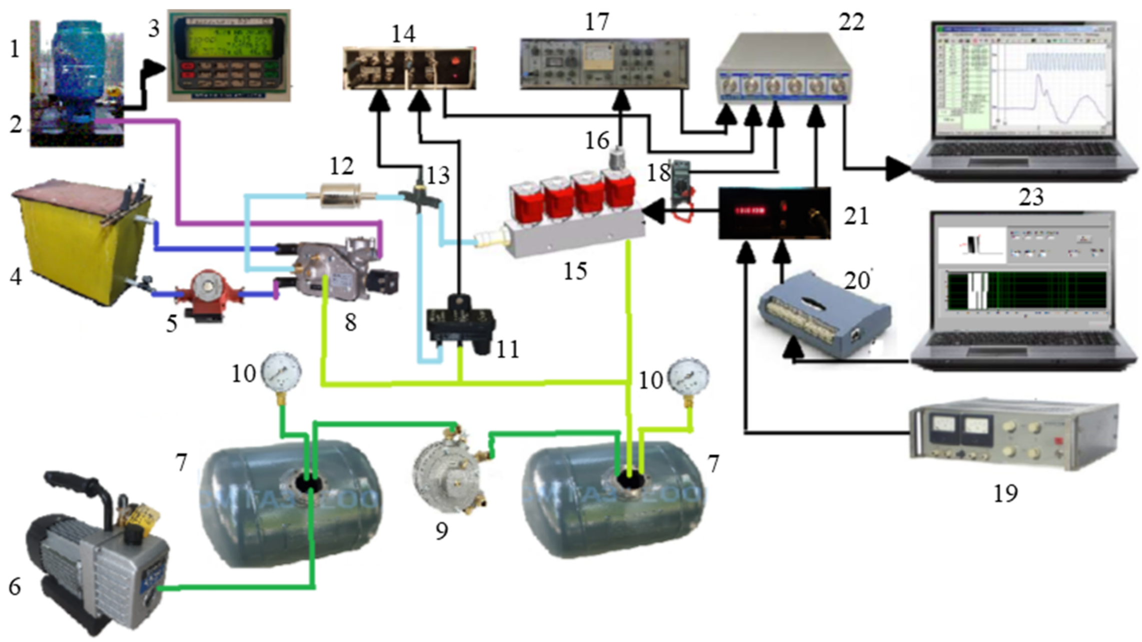

Figure 6. Experimental test stand: 1 – fuel tank; 2 – scale; 3 – scale controller with display; 4 – heated water tank; 5 – water pump; 6 – vacuum pump; 7 – tanks; 8 – gas reducer; 9 – modified reducer; 10 – pressure gauges; 11 – 2-Channel MAP pressure sensor; 12 – fuel filter; 13 – 3-Way nipple with temp. sensor; 14 – module for temperature and MAP sensors; 15 – injectors; 16 – accelerometer; 17 – chassis with charge amplifier module; 18 – current clamp; 19 – power supply; 20 – DAC for control signal generation; 21 – amplifier with counter; 22 – DAQ device; 23 – laptops.

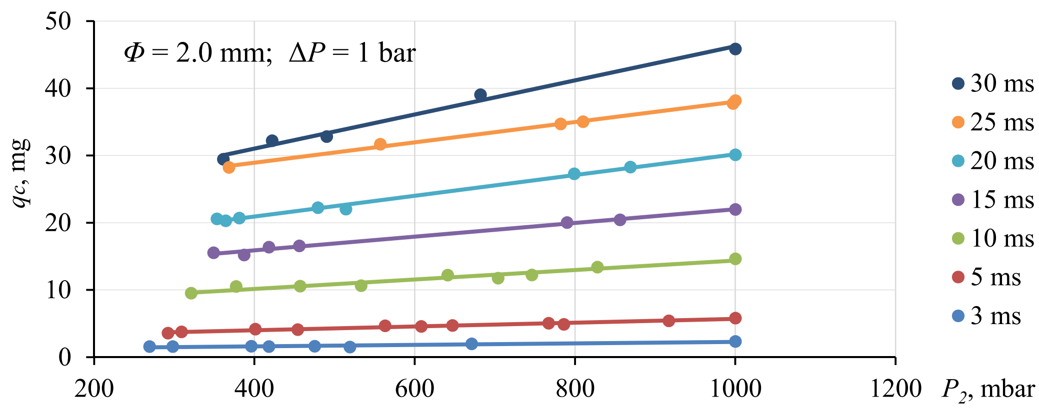

Figure 7. Mass of injected fuel per cycle vs downstream pressure (absolute), p_{diff} =1 bar, at different duration of electrical control signal.

The previous graph shows the full map of the engine operation depending on the duration of the control pulse and the pressure in the environment in which the gas is injected. In real conditions, the injector works in part of this map. Low engine loads are characterized by small values of manifold absolute pressure (MAP) and a small fuel portion per cycle determined by the corresponding duration of the control pulse. At high loads and at high mean effective pressure, ICE operates with an open throttle valve, which increases the absolute pressure in the intake manifold, and the fuel portion per cycle approximates the maximum values for the specific engine. Maximum loads are characterized by high peak values and an extended period of control pulse. For areas of the injector operation map, characterized by low pressures and long pulse duration and high pressure and short pulse values, the magnitude of the fuel portion per cycle for these modes is irrelevant or the gas injectors are excluded from the engine management for environmental considerations. The zone of possible operation of the gas injectors in real conditions is shown in figure 8 and is indicated by a blue colour.

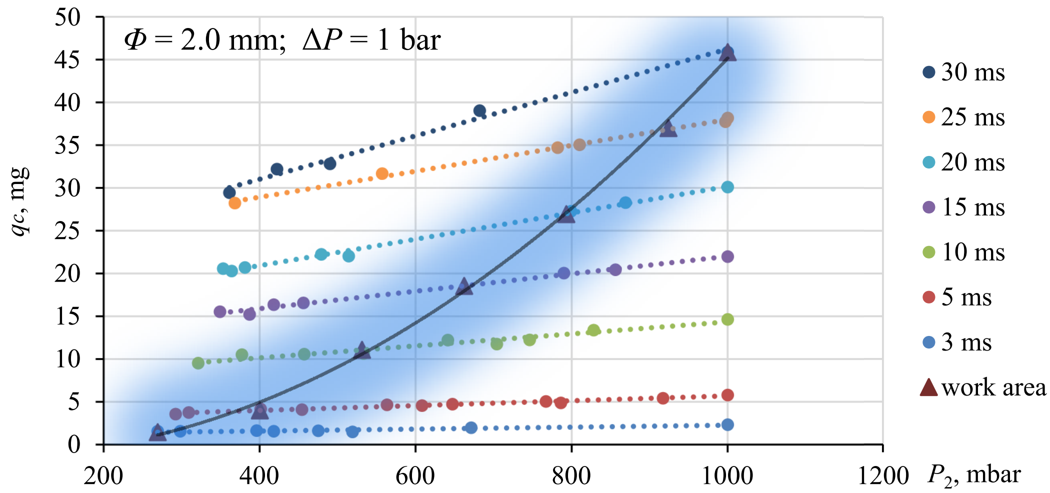

Figure 8. Working area of injector during normal engine operation.

Although the delivery of fuel with gas injectors is a non-continuous pulsed process, if we divide the injected fuel per cycle by the duration of the control signal, we could receive the mass air flow. Figure 9 shows the results for injection duration of 15 ms. It can be seen, that there is good correspondence between the experimentally determined points with the results from simulation.

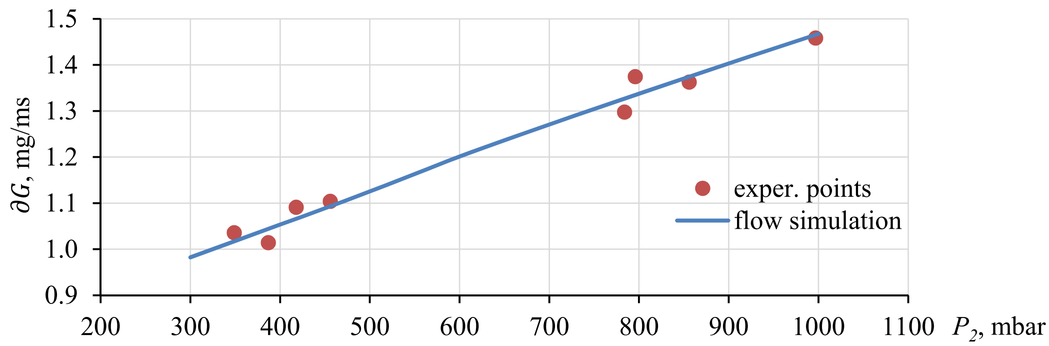

Figure 9. Mass flow rate.

5. Conclusion

Theoretical research has been done and gas leakage has been found to be critical under engine normal operating conditions.

Multi-parameter characteristics of a bottom feed type gas injectors are experimentally determined, that show the dependence of the duration of the electrical control signal on the injected fuel per cycle at varying pressure at the intake manifold and constant differential pressure.

It can be concluded that at increasing load (increasing intake manifold pressure), the proportional coefficient in the software of the gas control unit, that is used to relate the fuel flow of the gasoline and gas injectors should be corrected in negative direction.

References

[1] Ivanov Z, Mihaylov V and Mersinkov H 2014 Optical method for measurement needle lift of electromagnetic valves for LPG/CNG injection *XX Proc. Tech. Sci. Conf. ECO Varna 2014* 415-22

[2] Mitukiewicz G, Dychto R and Leyko J 2015 Relationship between LPG fuel and gasoline injection duration for gasoline direct injection engines *Fuel* **153** 526-34
[3] Potter M, Wiggert D and Ramadan B 2010 *Mechanics of Fluids* 5th ed. (Cengage) chapter 9
[4] Miller R 1996 *Flow Measurement Engineering Handbook* 3rd ed (McGraw-Hill)
[5] Matsson J 2018 *An Introduction to SolidWorks Flow Simulation* (SDC Publications) chapter 6, 9
[6] Szpica D 2018 Investigations of the flow characteristics of low-pressure gas-phase injectors under the conditions of numerical simulation *Proc. 23rd Int. Sci. Conf. Mechanika 2018* 164-68
[7] Zlateva P, Yordanov K and Dimitrov D 2014 Numerical simulation of temperature separation phenomena inside vortex tube *Acta Technica Corvininesis-Bulletin of Engineering* **7(4)** 23-26
[8] Ivanov Z and Mihaylov V 2012 Vibrational state of gas injection valves for spark-ignited (SI) engines *XVIII Proc. Tech. Sci. Conf. ECO Varna 2012* 502-10
[9] Dimitrov R, Magdziak – Tokłowicz M, Sitnik L, Reksa M, Tkaczyk M and Wróbel R 2014 A comparison of fuel consumption between the new European driving cycle test and the natural operation of a vehicle *22nd Int. Sci. Tech. Conf. Trans&Motauto* 14–16
[10] Belchev S 2019 Method for determining the parameters of injection of common rail injectors *XXVI Proc. Tech. Sci. Conf. ECO Varna 2019*
[11] Szpica D 2016 The influence of selected adjustment parameters on the operation of LPG vapor phase pulse injectors *J. Natural Gas Science and Engineering* **34** 1127–36
[12] Majerczyk A and Radzimirski S 2012 Effect of LPG gas fuel injectors on the properties of low emission vehicles *J. KONES Powertrain and Transport* **19** 401–10