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MULTI-RAMP FUEL INJECTION SYSTEM AUTOMATIC CONTROL

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Abstract: *This paper deals with a fuel injection system with multiple injector ramps. Fuel distribution (injection) is progressively achieved, being controlled by the injection pressure. The automatic control system operates as a follower system, realizing a proportional correspondence between throttle's position (displacement) and the injection fuel flow rate, as well an injection correction with respect to the air flow rate (air pressure behind the compressor). The author has determined a linearised adimensional mathematical model for the studied system and has built its block diagram with transfer functions; system's time behavior was also studied, based on some preformed simulations. A compared study was also discussed, between system's basic model and its simplified form, from the time behavior point of view. This kind of fuel system is useful for afterburning fuel systems, so it can be used for further studies concerning improvement possibilities, as well as for studies concerning its framing into more complex jet-engine control systems.*

Keywords: *fuel, injection, afterburning, control, pressure, jet engine, fuel-pump*

1. INTRODUCTION

Aircraft jet engines, as controlled objects, are multi-variable systems (multi-input and multi-output). The most important control parameter (input) is the fuel flow rate, both for basic engine and its afterburning system.

Fuel injection system is the most important control system, because it's accomplishing the engine's speed and/or combustor's temperature control and, eventually, engine's thrust control. Fuel injection control systems, based on different principles, are presented and studied in [8,9,10] and a fuel injection controller with correction sub-systems is studied in [11]. This kind of systems are meant to control, finally, engine's speed, using different feed-back methods and components, so fuel flow rate's level were established in order to accomplish the desired speed level, imposed by the throttle's position.

For an afterburner injection system the above-presented control possibilities are useless, because no speed-feed-back is possible and it is very difficult to design an appropriate control system and to build a reliable control scheme; afterburning system has as output parameters total thrust and afterburner's temperature, which are proportional to the fuel flow rate level; consequently, for the afterburning control, one has to use follower systems, in order to correlate the throttle's position with the desired thrust/temperature.

This paper deals with such a follower system, which establishes the correlation between the throttle's position and the injection fuel flow rate, corrected with the available air (burned gas) flow rate assure by the basic engine.

2. SYSTEM'S PRESENTATION

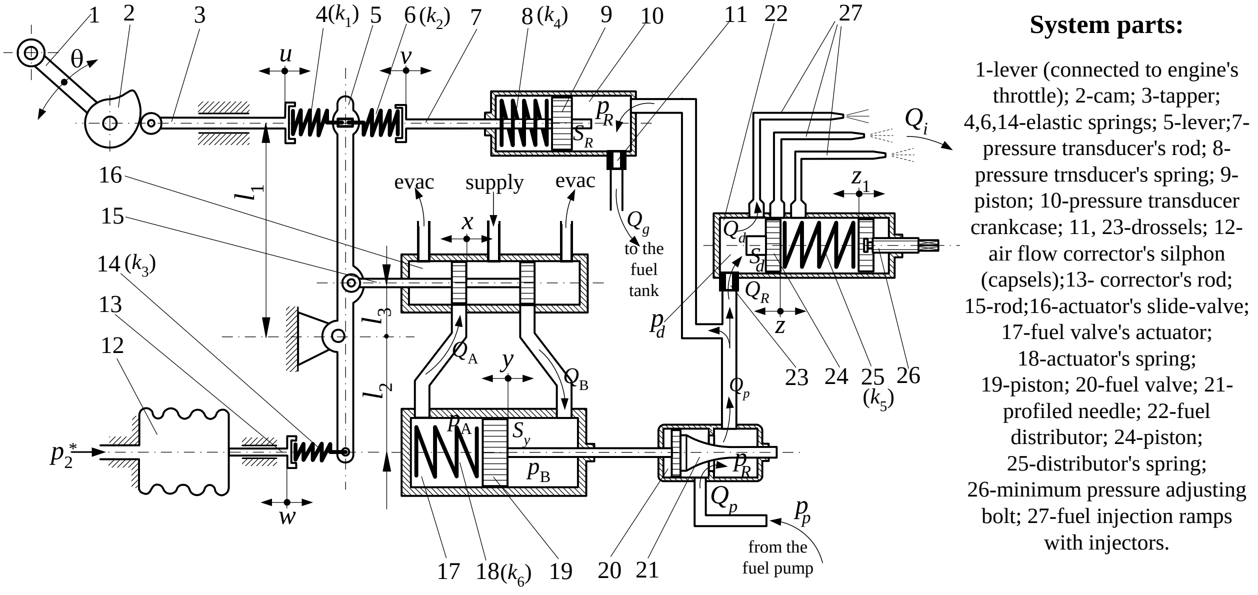

Fig. 1. Multi-ramp fuel injection automatic control system

A possible fuel injection system for an afterburner is shown in fig. 1. System's main parts are: a) throttle position transducer; b) air pressure corrector; c) fuel valve; d) fuel pressure transducer; e) fuel distributor with multi-ramp injection system.

The injected fuel flow rate is determined by the throttle's position and it's corrected with respect to the air flow rate (i.e. to p_2^* pressure behind engine's compressor); fuel's distribution is commanded by the distribution pressure p_d, which gives both the injection pressure p_i and the injection effective area A_i (i.e. number of active injectors / ramps with injectors).

Fuel valve's opening is commanded by the 17-actuator, which slide valve's positioning is the result of the balance between throttle's command, air flow correction and pressure feed-back (all of them acting on 5-lever).

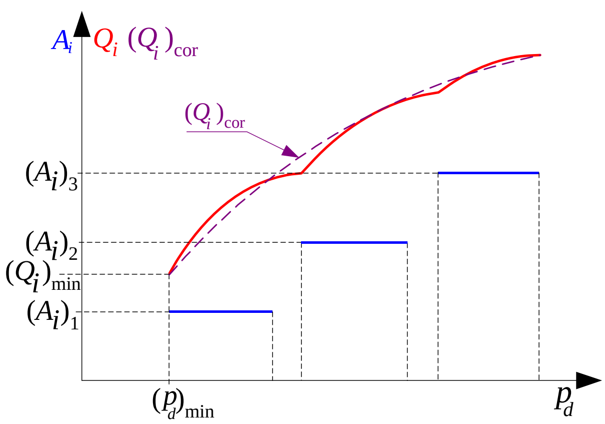

Fig. 2. Fuel flow rate versus distribution pressure

Fuel supply is achieved by a fuel pump, driven by engine's shaft; behind the fuel valve, the pressure p_R is proportional to the input parameters' level (θ and p_2^*), while fuel injection is double-commanded by the 22-distibutor's pressure p_d. Fig. 2 shows how fuel flow rate Q_i and injector effective area A_i depend on distribution pressure. As the distribution pressure increases, more injectors (injection ramps) are consecutively supplied, so the flow rate is continuously increasing. One has also estimated an equivalent (corrected) flow rate behavior $(Q_i)_{cor}$, useful for further studies, also presented. Minimum distribution pressure value $(p_d)_{\min}$ is preset, during the ground tests, by the 26-adjusting bolt, which realizes a proportional pre-compression of the 25-spring. So, minimum value of the distribution pressure is

$$(p_d)_{\min} = \frac{k_5 z_1}{S_d}, \qquad (1)$$

where z_1 is spring's pretension. This minimum value is necessary to ensure an appropriate fuel jet shape (spreading and spraying).

3. SYSTEM MATHEMATICAL MODEL

For each main system part one has to identify the motion equations, which are to be processed, in order to bring them to an optimal form to be used for further studies.

3.1. Transducer's model

Input parameter transducer consists of:

- lever-cam-taper block;
- air pressure corrector block;
- fuel pressure feed-back actuator.

Their motion equations are

$$u = f(\theta), \tag{2}$$

$$v = k_s\sqrt{p_2^*}, \tag{3}$$

$$l_1(k_1 u - k_2 w) - l_2 k_3 v = l_3\left(m_{15}\frac{\mathrm{d}^2 x}{\mathrm{d}t^2} + \xi\frac{\mathrm{d}x}{\mathrm{d}t}\right) \tag{4}$$

where u – taper's displacement, with respect to 2-cam's profile $f(\theta)$, v – air pressure corrector's rod displacement, w – feed-back's rod displacement, k_1, k_2, k_3 – springs' elastic constants, l_1, l_2, l_3 – 5-lever arms' length, x – actuator's slide valve's displacement, m_{15} – slide-valve's mass, ξ – viscous friction coefficient; k_s – silphon's (capsels') elastic constant, p_2^* – air pressure behind engine's compressor.

In order to make these equations more accessible for further operations, one has to linearise them and then bring them to non-dimensional forms, using finite differences method (as depicted in [6,8,9,10,11]). Equation system can be linearized based on the small perturbation hypothesis, considering formally any variable or parameter X as $X = X_0 + \Delta X$ and $\overline{X} = \frac{\Delta X}{X_0}$, where ΔX – parameter's deviation, X_0 – steady state regime's value and $\overline{X}$ – non-dimensional deviation.

Introducing the new form of each parameter into the above mentioned equation system and separating the steady state regime terms, one obtains a new form of the system, which becomes

$$\overline{u} = k_{\theta u}\overline{\theta}, \tag{5}$$

$$\overline{v} = k_{p2v}\overline{p_2^*}, \tag{6}$$

$$k_{ux}\overline{u} - k_{wx}\overline{w} - k_{vx}\overline{v} = s(\tau_x s + 1)\overline{x} \tag{7}$$

where the used annotations are

$$k_{\theta u} = \frac{\theta_0}{u_0}\left(\frac{\partial f}{\partial\theta}\right)_0, k_{p2v} = \frac{k_s\sqrt{p_{2_0}^*}}{v_0}, k_{ux} = \frac{l_1 k_1 u_0}{\xi l_3 x_0},$$
$$k_{wx} = \frac{l_1 k_2 w_0}{\xi l_3 x_0}, k_{vx} = \frac{l_2 k_3 v_0}{\xi l_3 x_0}, \tau_x = \frac{m_{15}}{\xi}. \tag{8}$$

3.2. Fuel valve actuator's model

As presented in [9], simplest actuator consists of a hydraulic cylinder commanded by a slide valve; its mathematical model is

$$\overline{y} = \frac{1}{\tau_y s + \lambda_y}\overline{x}, \tag{9}$$

where

$$\tau_y = \frac{S_y}{\mu_s b_s}\frac{y_0}{x_0}\sqrt{\frac{\rho_h}{p_{hs}}}, \lambda_y = \frac{k_6 y_0}{2S_y p_{hs}}, \tag{10}$$

S_y – actuator piston's area, μ_s – flow rate coefficient, b_s – slide valve's slots width, k_6 – actuator spring elastic constant, ρ_h – hydraulic fluid density, p_{hs} – hydraulic supply pressure.

3.3. Fuel valve equations

Fuel valve's profiled needle's positioning determines both the flow rate through the valve and the pressure p_R behind it, which is used in the feed-back actuator (pressure transducer 10). Valve's model, together with pressure transducer's model, consists of

$$Q_p = \mu_v A(y)\sqrt{\frac{2}{\rho}}\sqrt{p_p - p_R}, \tag{11}$$

$$Q_R = \mu_{23}\frac{\pi d_{23}^2}{4}\sqrt{\frac{2}{\rho}}\sqrt{p_R - p_d}, \tag{12}$$

$$Q_g = \mu_g \frac{\pi d_g^2}{4}\sqrt{\frac{2}{\rho}}\sqrt{p_R}, \qquad (13)$$

$$Q_p - Q_R - Q_g = \beta V_{R0}\frac{\mathrm{d}p_R}{\mathrm{d}t} + S_R\frac{\mathrm{d}w}{\mathrm{d}t}, \qquad (14)$$

$$S_R p_R = m_9\frac{\mathrm{d}^2 w}{\mathrm{d}t^2} + \xi\frac{\mathrm{d}w}{\mathrm{d}t} + k_4 w, \qquad (15)$$

where Q_p – fuel pump flow rate, Q_R – distributor's input flow rate, Q_g – discharge flow rate, μ_v, μ_{23}, μ_g – flow coefficients, d_{23}, d_g – drossels' diameters, ρ – fuel density, β – compressibility coefficient (assumed as null for fuel), S_R – transducer's piston area, m_9 – transducer's piston+rod mass, k_4 – transducer spring elastic constant, $A(y)$ – fuel valve effective area, depending on 21-profiled needle shape.

Introducing (10), (11) and (12) in (13), after same method applying, one obtains

$$\overline{w} = \frac{k_{wR}}{T_w^2 s^2 + 2\omega_0 T_w s + 1}\overline{p}_R, \qquad (16)$$

$$\overline{p}_R = k_{Ry}\overline{y} + k_{Ra}\overline{p}_p + k_{Rd}\overline{p}_d - \tau_w s\overline{w}, \qquad (17)$$

where

$$k_{wR} = \frac{S_R p_{R0}}{k_4 w_0}, T_w = \sqrt{\frac{m_9}{k_4}}, \omega_0 = \frac{\xi}{2\sqrt{m_9 k_4}},$$

$$k_{yy} = (k_{pp} + k_{Rp} + k_{gR})p_{R0}, k_{Ry} = \frac{k_{py} y_0}{k_{yy}},$$

$$k_{Ra} = \frac{k_{pp} p_{p0}}{k_{yy}}, k_{Rd} = \frac{k_{Rp} p_{d0}}{k_{yy}}, \tau_w = \frac{S_R w_0}{k_{yy}},$$

$$k_{py} = \mu_v\sqrt{\frac{2}{\rho}}\left(\frac{\partial A_y}{\partial y}\right)_0\sqrt{p_{p0} - p_{R0}},$$

$$k_{pp} = \frac{\mu_v A_{y0}}{\sqrt{2\rho(p_{p0} - p_{R0})}}, k_{gR} = \frac{\mu_g \pi d_g^2}{4\sqrt{2\rho p_{R0}}},$$

$$k_{Rp} = \frac{\mu_{23}\pi d_{23}^2}{4\sqrt{2\rho(p_{p0} - p_{R0})}}. \qquad (18)$$

3.4. Distributor mathematical model

Multi-ramp fuel injection distributor's motion equations are

$$Q_d = \mu_d b_d z\sqrt{\frac{2}{\rho}}\sqrt{p_d - p_i}, \qquad (19)$$

$$Q_R - Q_d = \beta V_{R0}\frac{\mathrm{d}p_d}{\mathrm{d}t} + S_d\frac{\mathrm{d}z}{\mathrm{d}t}, \qquad (20)$$

$$S_d p_d = m_{24}\frac{\mathrm{d}^2 z}{\mathrm{d}t^2} + \xi\frac{\mathrm{d}z}{\mathrm{d}t} + k_5 z, \qquad (21)$$

$$Q_i = \mu_i A_i(z)\sqrt{\frac{2}{\rho}}\sqrt{p_i}, \qquad (22)$$

$$Q_d = Q_i. \qquad (23)$$

where μ_d, μ_i – flow rate coefficients, b_d – slot width, z – distributor's piston displacement, S_d – distributor's piston area, p_i – injection pressure, m_{24} – distributor's piston mass, k_5 – distributor's spring elastic constant, $A_i(z)$ – injectors' effective area.

Fuel distributor linearised adimensional mathematical model becomes

$$\overline{p}_d = k_{di}\overline{p}_i + k_{dR}\overline{p}_R - k_{zp}(\tau_z s + 1)\overline{z}, \qquad (24)$$

$$\overline{z} = \frac{k_{zd}}{T_z^2 s^2 + 2\omega_1 T_z s + 1}\overline{p}_d, \qquad (25)$$

$$\overline{p}_i = k_{id}\overline{p}_d + k_{zz}\overline{z}, \qquad (26)$$

$$\overline{Q}_i = k_{Qi}\overline{p}_i + k_{Qz}\overline{z}, \qquad (27)$$

where $k_{ii} = (k_{ip} + k_{dp})p_{d0}$, $k_{di} = \frac{k_{dp} p_{i0}}{k_{ii}}$,

$$k_{dR} = \frac{k_{Rp} p_{R0}}{k_{ii}}, k_{zp} = \frac{k_{dz} z_0}{k_{ii}}, \tau_z = \frac{S_d}{k_{dz}},$$

$$k_{zd} = \frac{S_d p_{d0}}{k_5 z_0}, T_z = \sqrt{\frac{m_{24}}{k_5}}, \omega_1 = \frac{\xi}{2\sqrt{m_{24} k_5}},$$

$$k_{id} = \frac{k_{dp} p_{d0}}{k_{ii}}, k_{zz} = \frac{(k_{dz} - k_{iz})z_0}{k_{ii}}, k_{Qz} = \frac{k_{iz} z_0}{Q_{i0}},$$

$$k_{Qi} = \frac{k_{ip} p_{i0}}{Q_{i0}}, k_{dp} = \frac{\mu_d b_d z_0}{\sqrt{2\rho(p_{d0} - p_{i0})}},$$

$$k_{dz} = \mu_d b_d\sqrt{\frac{2(p_{d0} - p_{i0})}{\rho}}, k_{ip} = \frac{\mu_i A_{i0}}{\sqrt{2\rho p_{i0}}},$$

$$k_{iz} = \mu_i\left(\frac{\partial A_i}{\partial z}\right)_0\sqrt{\frac{2 p_{i0}}{\rho}}. \qquad (28)$$

Fuel injection control system's linear adimensional mathematical model consists of equations (5), (6), (7), (9), (16), (17), (24), (25), (26) and (27). Based on these equation, one has built system's block diagram with transfer functions, as fig. 3 shows.

4. SIMPLIFIED MATHEMATICAL MODEL. TRANSFER FUNCTIONS

Assuming that both friction effects and inertial effects are insignificant, so negligible,

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one has obtained some simpler forms for the equations, which contain terms involving mass, friction coefficients and compressibility coefficients. New forms are

$$k_{ux}\bar{u} - k_{wx}\bar{w} - k_{vx}\bar{v} = s\bar{x}\,, \tag{29}$$

$$w = k_{wR}\,\bar{p}_R\,, \tag{30}$$

$$\bar{z} = k_{zd}\,\bar{p}_d\,. \tag{31}$$

System's model becomes simpler

$$\bar{Q}_i = \frac{k_{si}\left(k_{\theta i}\bar{\theta} - k_{p2i}\bar{p}_2^*\right)}{a_4 s^4 + a_3 s^3 + a_2 s^2 + a_1 s + a_0}\,, \tag{32}$$

so, system transfer functions are

$$H_\theta(s) = \frac{k_{si}k_{\theta i}}{a_4 s^4 + a_3 s^3 + a_2 s^2 + a_1 s + a_0}\,, \tag{33}$$

$$H_{p2}(s) = -\frac{k_{si}k_{p2i}}{a_4 s^4 + a_3 s^3 + a_2 s^2 + a_1 s + a_0}\,; \tag{34}$$

coefficients' expressions are very complicated, but this new form of the mathematical model is much more accessible and usable for further applications and studies.

An observation should be made, regarding the term which contains p_p. One has assumed the pump pressure as constant, being created by the fuel pump which operates at constant regime. However, if one takes into account that the fuel pump is driven by the engine's shaft, as well as the fuel pump characteristic (dependence between engine-pump speed and supplying pressure), the transfer function must be completed with a term containing engine speed parameter, as follows

$$\bar{Q}_i = \frac{k_{si}\left[k_{\theta i}\bar{\theta} - k_{p2i}\bar{p}_2^* + \left(b_2 s^2 + b_1 s + b_0\right)\bar{n}\right]}{a_4 s^4 + a_3 s^3 + a_2 s^2 + a_1 s + a_0}\,. \tag{35}$$

5. SYSTEM'S QUALITY

Based on system's mathematical model, as well as on the above presented block diagram with transfer functions, one has performed some simulations in order to establish system's

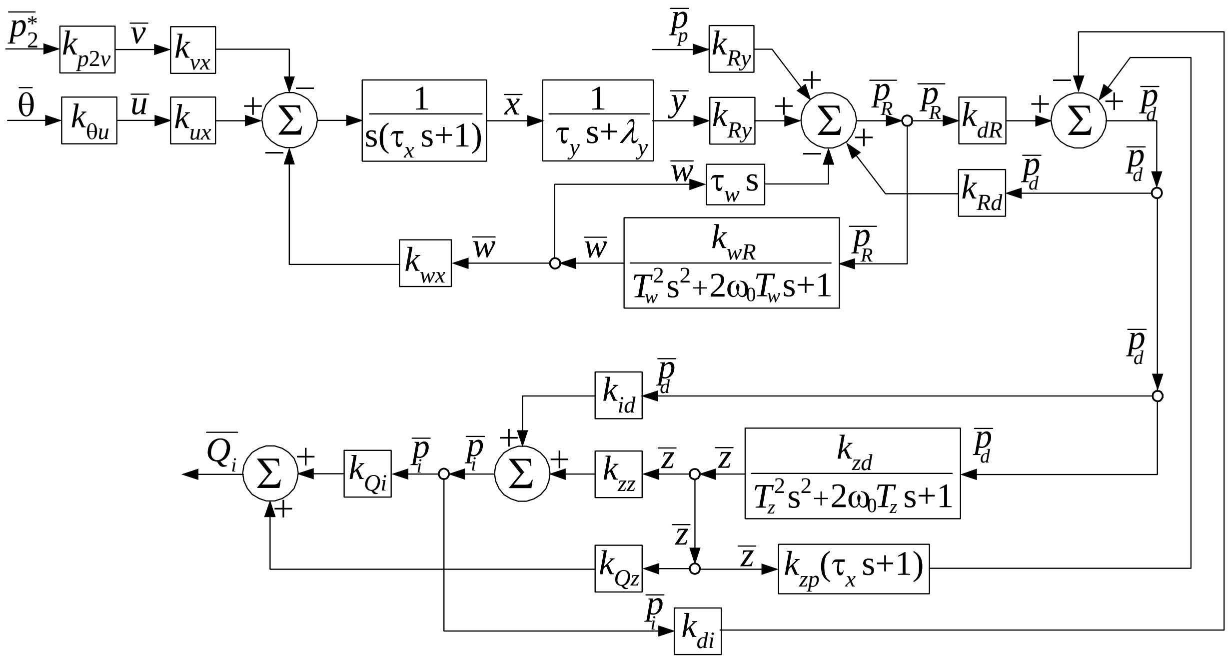

Fig. 3. System's block diagram with transfer functions

time behavior for step input(s).

Equations' coefficients were calculated for a hypothetically usage of the fuel system on a VK-1F type jet engine and implemented in the simulation scheme based on the above-presented block diagram with transfer functions. For this simulation one has neglected the fuel pump influence, considering that fuel pressure is kept constant (e.g. by using a constant pressure valve), so $\overline{p}_p = 0$.

The simulation was performed in two cases

a) step input for throttle's position and constant air flow (air pressure);

b) step input for air pressure and fixed throttle position.

Main output parameter is the fuel flow rate $\overline{Q}_i$, but one has also studied secondary output parameters, such as transducer lever dis-

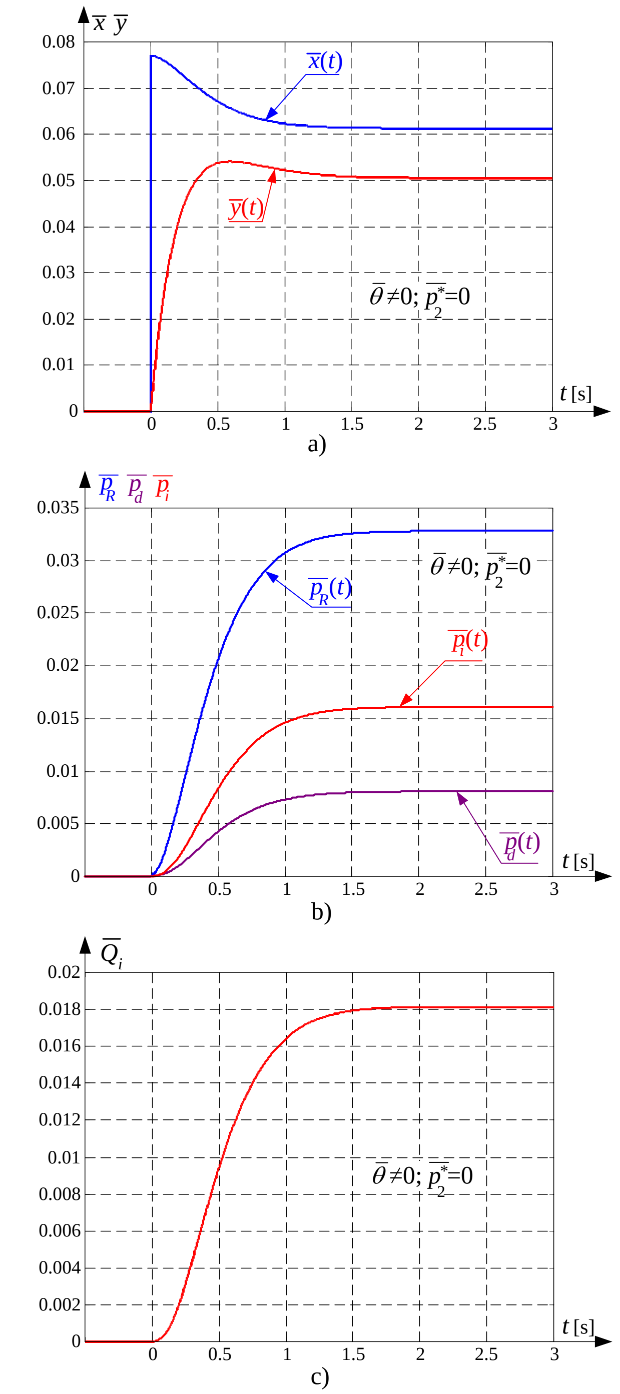

Fig. 4. System step response for $\overline{\theta}$ throttle position parameter step input and constant $\overline{p}_2^*$ air pressure parameter

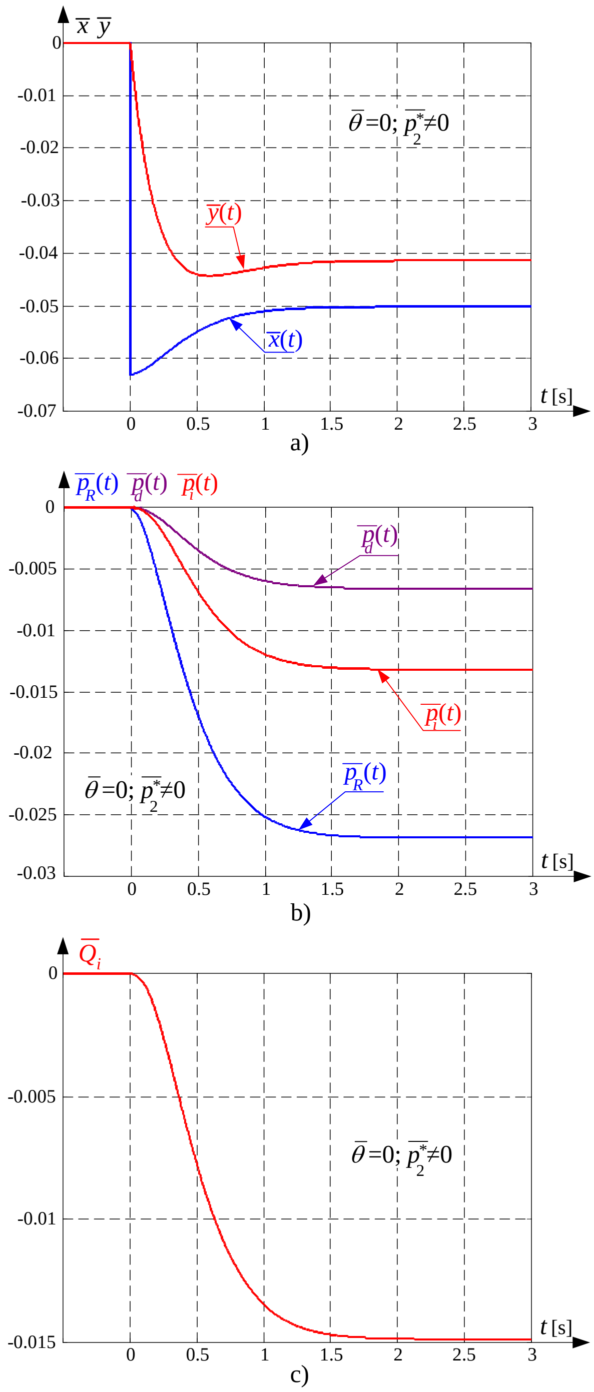

Fig. 5. System step response for $\overline{p}_2^*$ air pressure parameter step input and constant $\overline{\theta}$ throttle position parameter

placement $\bar{x}$ and profiled needle displacement $\bar{y}$, as well as fuel pressures $\bar{p}_R, \bar{p}_d$ and $\bar{p}_i$.

Simulation results, based on system's mathematical model, are presented in fig. 4 (case *a*) and 5 (case *b*), containing step responses for all the above-mentioned parameters.

As fig. 4.a and 5.a show, both $\bar{\theta}$ and $\bar{p}_2^*$ step input determine similar behavior for $\bar{x}$ and $\bar{y}$; after an initial "jump", transducer's lever displacement $\bar{x}$ asymptotically stabilizes with an important static error ($5\% \div 6\%$); fuel valve's profiled needle's displacement $\bar{y}$ asymptotically stabilizes (static error between 4% and 5%), but after a small initial override. Response time, for both parameters and for both of cases is around $(1.5 \div 2)$ s, which is an acceptable value.

Fuel pressure parameters, as fig. 4.b and 5.b shows, have aperiodic behavior and very small static error values (between 0.5% and 3.2%), especially for the distribution pressure parameter; response times, acceptable as values, are the same, between 1.5 s and 2 s.

Main parameter's $\bar{Q}_i$ behavior, in both of studied cases, is aperiodic, with acceptable

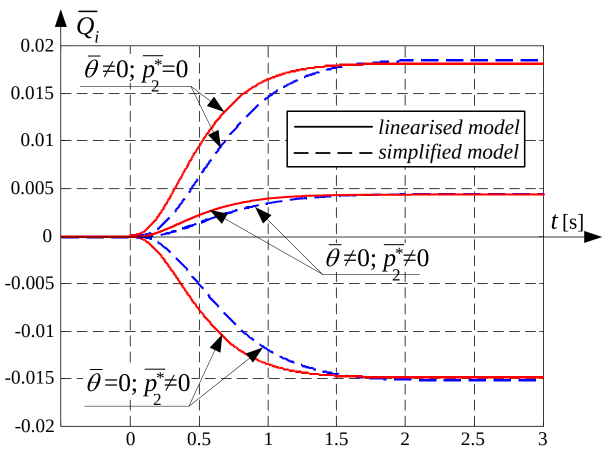

Fig. 6. Fuel flow rate behavior given by the complete model and by the simplified model

static errors (around 1.8%) and acceptable response time values (1.5 to 2.5 s).

In order to facilitate system mathematical model for more complex control schemes simulation, one has determined system's simplified model. A comparison between the complete model and the simplified model was realized, from their time behavior point of view. The simulation results were presented in fig. 6.

Simplified mathematical model expression, determined for the above-mentioned condition, which was used for the graphics in fig. 6 is

$$\bar{Q}_i = \frac{4.957\left(0.448\bar{\theta} - 0.397\,\bar{p}_2^*\right)}{s^4 + 15.01s^3 + 88.93s^2 + 172.11s + 119.94} \tag{36}$$

Time behavior for the simplified system is also presented in fig. 6 (dashed line), while time behavior for the complete model is presented with continuous line.

Some observations can be made, concerning the situation in fig. 6. One has studied system's quality for both already discussed situations (first $\bar{\theta}$, second $\bar{p}_2^*$ step input), as well as for a combined step input of $\bar{\theta}$ and $\bar{p}_2^*$. No matter the situation were, system's time behavior is a stable one; the difference between the static errors is insignificant and the dynamic regime in either case is very similar. Consequently, the simplified mathematical model provides sufficient accuracy to be used instead of the complete model.

When the fuel pump influence must be revealed, instead of eq. (32) (particularly (36)), one has to use eq. (35) as follows

$$\bar{Q}_i = \frac{4.957}{s^4 + 15.01s^3 + 88.93s^2 + 172.11s + 119.94} \times \\ \times \left[0.448\bar{\theta} - 0.397\,\bar{p}_2^* + \right. \\ \left. + \left(0.023s^2 + 0.062s + 0.076\right)\bar{n}\right] \tag{37}$$

The above-presented form is useful for embedded engine control system studies.

CONCLUSION

This paper has presented a fuel injection system, working as follower system, with respect to the engine's throttle's position and being assisted by an air pressure corrector. Fuel injection is assured by a fuel distributor with multiple injection ramps, which are progressively activated as distribution pressure grows and distributor's piston forwards.

One has built system's mathematical model, which was linearised and brought to an adimensional form, appropriate for further studies. Based on it, system's block diagram with transfer functions was built. Assuming some simplifying hypothesis, a much simpler system form was issued, more appropriate for study and for further similar applications.

The performed simulations have shown an appropriate system quality, which means asymptotic output parameters' stabilization, small static errors and small response time.

This fuel system can't be implemented for jet engines control, because its lack of an effective speed feed-back possibility, but it's recommended for afterburning control; it may be included in a more complex control system, for example for an embedded propulsion system (jet engine+afterburning).

The employed method and some of the obtained results could be extended for similar further studies (such as follower systems, complex fuel injection systems, embedded jet engine control systems etc.).

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