MATHEMATICAL MODEL BASED OF TRANSFER FUNCTIONS FOR DYNAMIC OPERATIONS OF SUPERCHARGED ENGINE

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Rezumat: Lucrarea prezinta modelul matematic pentru comportarea dinamica a motorului diesel supraalimentat cu turbocompresor cu rotatie libera. Modelul matematic se bazeaza pe cunoasterea caracteristicilor susbsistemelor principale, asa cum sunt motorul propriu-zis, turbocompresorul, galeria de admisie si evacuare, instalatia de injectie. Caracteristicile acestor subsisteme sunt cunoscute pentru regimurile stationare de functionare si in apropierea acestora, dupa datele experimentale, pe baza determinarilor pe stand. Modelul matematic este exprimat de ecuatia diferentiala de ordinul doi care exprima comportarea dinamica a intregului sistem. Sunt stabilite functiile de transfer, care pot fi folosite pentru reglarea automata a intregului sistem.
Cuvinte cheie: motor diesel supraalimentat, functii de transfer, functionare in regim dinamic, regimuri nestationare de functionare.

Abstract: The paper deals with the differential equation expressing the dynamic operations of turbocharged internal combustion engines using supercharger with free rotation units. Mathematical model is based on knowledge of the characteristics of subsystems, such as engine itself, turbocharger, exhaust and intake manifold and the injection system to stationary regimes. Transfer functions are determinated, functions that are used for achieving and adjusting automatic control system which controls the operation of the whole system (engine) under consideration.
Keywords: supercharged diesel engine, transfer functions, dynamic regime, unsteady working conditions.

1. INTRODUCTION

The paper presents the mathematical model for calculatind transfer function for the whole supercharged internal combustion engine, with free rotation supercharger units. Mathematical model is based on differential equations and the transfer functions established, [8], [3], [4], [5], [6], [7], [2] for dynamic operations of subsystems, as they are exhaust and intake manifold, injection system, engine itself, turbocharger [1], as part of internal combustion engines, as shown in figure 1. On the basis of these equations results the dynamical behavior of internal combustion engine. Using transfer functions can build automatic control system (controller) and make its award with engine, such as existing automatic governor for revolution.

2. DYNAMIC REGIME OF DIESEL SUPERCHARGED ENGINE

We consider the following diagram (Fig. 1) with the subsystems [8], [1], parts of the supercharged internal combustion engine, with free rotation turbocharger. In these diagram, subsystems are:

K - turbocompressor; S. Inj. - Fuel injection system; TG - gas turbine; Diesel - diesel engine itself; C.AD - intake pipe; C.EV - exhaust pipe; ω - angular speed of engine; ω_{TK} - angular speed of turbocharger; ω_{inj} - angular speed of fuel injection pump; c_{cycle} fuel consumption for a cycle; $h_{K,\ T}$ - the position of adjustment actuator of turbocompressor respectively turbine;

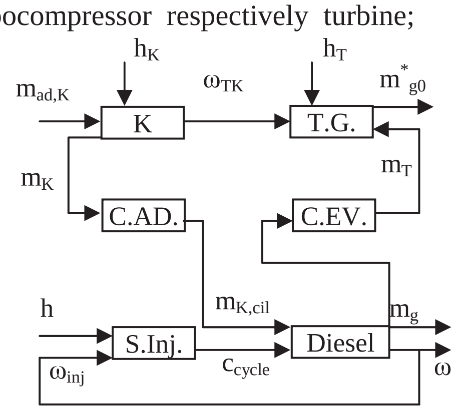

Fig.1 Diagram for supercharged engine subsystems

$m_{ad,K}$ - mass flow of air admitted into the compressor;
m_K - mass flow of air entering the intake manifold; $m_{K,cil}$ - mass flow of air entering the engine cylinders; h - actuator position of the injection pump for diesel engine; m_g – exhaust gas mass flow entering the exhaust manifold; m_T - exhaust gas mass flow entering the turbine; m^*_{g0} - exhaust gas mass flow coming out in the environment.
Using the differential equations established in [3], [4], [5], [6], [8], [2] which shape the dynamical operations of constitutive parts near by the steady working conditions, it's possible to achieve the differential equation for whole supercharged internal combustion engine therefore the dynamical behavior.
According with [8], [3], [4], [5], [6], [7] can write:
-- for Diesel engine itself:

$$T_{eng}\frac{d\left(\Delta\omega^*\right)}{d\tau}+K_{eng}\left(\Delta\omega^*\right)=\Delta c^*_{cycle}+\theta_{eng\,\Delta p_k}\Delta p^*_K-\theta_{eng\,\Delta hs}\Delta h^*_s \quad (1)$$

after use Laplace transformation:

$$\left(T_{eng}\,s+K_{eng}\right)\Delta\Omega^*(s)=\Delta C^*_{cycle}(s)+\theta_{eng\,\Delta p_k}\Delta P^*_K(s)-\theta_{eng\,\Delta hs}\Delta H^*_s(s) \quad (1')$$

-- for injection system:

$$K_{ap.inj}\,\Delta c^*_{cycle}=\Delta h^*+\theta_{ap.inj\,\Delta\omega}\Delta\omega^* \quad (2)$$

after use Laplace transformation:

$$K_{ap.inj}\,\Delta C^*_{cycle}(s)=\Delta H^*(s)+\theta_{ap.inj\,\Delta\omega}\Delta\Omega^*(s) \quad (2')$$

-- for turbocharger :

$$T_{TK}\frac{d\left(\Delta\omega_{TK}^*\right)}{d\tau}+K_{TK}\left(\Delta\omega^*_{TK}\right)=\Delta p^*_T+\theta_{TK\Delta c_{cycle}}\Delta c^*_{cycle}-\theta_{TK\Delta p_k}\Delta p^*_K+\theta_{TK\Delta h_T}\Delta h^*_T-\theta_{TK\Delta h_K}\Delta h^*_K \quad (3)$$

after use Laplace transformation:

$$\left(T_{TK}s+K_{TK}\right)\Delta\Omega^*_{TK}(s)=\Delta P^*_T+\theta_{TK\Delta C_{cycle}}\Delta C^*_{cycle}(s)-\theta_{TK\Delta p_k}\Delta P^*_K(s)+\theta_{TK\Delta h_T}\Delta H^*_T(s)-\theta_{TK\Delta h_K}\Delta H^*_K(s) \quad (3')$$

-- for intake manifold:

$$T_{c.ad}\frac{d\left(\Delta p_K^*\right)}{d\tau}+K_{c.ad}\left(\Delta p^*_k\right)=\Delta\omega^*_{TK}-\theta_{c.ad\Delta\omega}\Delta\omega^*-\theta_{c.ad\Delta T_K}\Delta T^*_K+\theta_{c.ad\Delta h_K}\Delta h^*_K \quad (4)$$

after use Laplace transformation:

$$\left(T_{c.ad}\,s+K_{c.ad}\right)\Delta P^*_K(s)=\Delta\Omega^*_{TK}-\theta_{c.ad\,\Delta\omega}\Delta\Omega^*(s)-\theta_{c.ad\,\Delta T_K}\Delta T^*_K(s)+\theta_{c.ad\,\Delta h_K}\Delta H^*_K(s) \quad (4')$$

-- for exhaust manifold:

$$T_{c.ev}\frac{d\left(\Delta p_T^*\right)}{d\tau}+K_{c.ev}\left(\Delta p^*_T\right)=\Delta\omega^*+\theta_{c.ev\Delta p_k}\Delta p^*_k+\theta_{c.ev\Delta c_{cycle}}\Delta c^*_{cycle}-\theta_{c.ev\Delta h_T}\Delta h^*_T \quad (5)$$

after use Laplace transformation :

$$\left(T_{c.ev}\,s+K_{c.ev}\right)\left(\Delta P^*_T(s)\right)=\Delta\Omega^*(s)+\theta_{c.ev\,\Delta p_k}\Delta P^*_k(s)+\theta_{c.ev\,\Delta c_{cycle}}\Delta C^*_{cycle}(s)-\theta_{c.ev\,\Delta h_T}\Delta H^*_T(s) \quad (5')$$

Where:

$$\omega=\omega_0+\Delta\omega;\Delta\omega^*=\frac{\Delta\omega}{\omega_0} \quad (6)$$

$$\Delta c^*_{cycle}=\frac{\Delta c_{cycle}}{c_{cycle0}};\Delta p^*_K=\frac{\Delta p_K}{p_{K0}} \quad (7)$$

$$\Delta h^*_S=\frac{\Delta h_S}{h_{S0}}=\Delta h^*_R \quad (8)$$

ω_0=angular speed for steady working conditions;
c_{cycle0}= fuel consumption for stedy working conditions;
h_{s0}=adjustment devices at consumer for steady working conditions;
p_{K0}=the pressure furnished by compressor for steady working conditions;
p_{T0}=gas pressure at the entrance of gas turbine for steady working conditions;
Where:

$$T_{eng}=\frac{J_{eng}\cdot\omega_0}{\left(\frac{\partial M_{te}}{\partial c_{cycle}}\right)_{\omega_0,p_{k0}}\cdot c_{cycle0}} \quad (9)$$

$$\theta_{eng\,\Delta p_k}=\frac{\left(\frac{\partial M_{te}}{\partial p_k}\right)\cdot p_{k0}}{\left(\frac{\partial M_{te}}{\partial c_{cycle}}\right)_{\omega_0,p_{k0}}\cdot c_{cycle0}} \quad (10)$$

$$K_{eng}=\frac{F_{st\_eng}\cdot\omega_0}{\left(\frac{\partial M_{te}}{\partial c_{cycle}}\right)_{\omega_0,p_{k0}}\cdot c_{cycle0}} \quad (11)$$

$$\theta_{eng\,\Delta h_s} = \frac{\left(\frac{\partial M_{teR}}{\partial h_S}\right)_{\omega_0} \cdot h_{s0}}{\left(\frac{\partial M_{te}}{\partial c_{cycle}}\right)_{\omega_0 , p_{k0}} \cdot c_{cycle0}} \tag{12}$$

$$T_{TK} = \frac{J_{TK} \cdot \omega_{TK\,0}}{\left(\frac{\partial M_{tTG}}{\partial p_T}\right) \cdot p_{T0}} \tag{13}$$

$$K_{TK} = \frac{F_{st\,TK}\ \omega_{TK\ 0}}{\left(\frac{\partial M_{tTG}}{\partial p_T}\right) p_{T\,0}} \tag{14}$$

$$\theta_{TK\,\Delta p_k} = \frac{\left(\frac{\partial M_{tK}}{\partial p_k}\right) \cdot p_{k0}}{\left(\frac{\partial M_{tTG}}{\partial p_T}\right) \cdot p_{T0}} \tag{15}$$

$$\theta_{TK\,\Delta h_T} = \frac{\left(\frac{\partial M_{tTG}}{\partial h_T}\right) \cdot h_{T0}}{\left(\frac{\partial M_{tTG}}{\partial p_T}\right) \cdot p_{T0}} \tag{16}$$

$$\theta_{TK\ \ \Delta h_K} = \frac{\left(\frac{\partial M_{tK}}{\partial h_K}\right) h_{K\ 0}}{\left(\frac{\partial M_{tTG}}{\partial p_T}\right) p_{T\ 0}} \tag{17}$$

$$F_{st\_eng} = \left(\frac{\partial M_{teR}}{\partial \omega}\right)_{h_{s0}} - \left(\frac{\partial M_{te}}{\partial \omega}\right)_{p_{k0}, c_{ciclu0}} \tag{18}$$

$$F_{stTK} = \left(\frac{\partial M_{tK}}{\partial \omega}\right) - \left(\frac{\partial M_{tTG}}{\partial \omega}\right) \tag{19}$$

$$K_{ap.inj} = \frac{c_{cycle\,0}}{\left(\frac{\partial c_{cycle}}{\partial h}\right) \cdot h_0} \tag{20}$$

$$\theta_{ap\ \_\ inj\ \Delta\omega} = \frac{\left(\frac{c_{cycle}}{\partial \omega}\right) \omega_{\ 0}}{\left(\frac{\partial c_{cycle}}{\partial h}\right) h_0} \tag{21}$$

$$\theta_{TK\ \ \Delta C_{cycle}} = \frac{\left(\frac{\partial M_{tTG}}{\partial c_{cycle}}\right) c_{cycle\ \ 0}}{\left(\frac{\partial M_{tTG}}{\partial p_T}\right) p_{T\ 0}} \tag{22}$$

$$K_{c.ad} = \frac{F_{st\ c.ad}\ p_{k\,0}}{\left(\frac{\partial \dot{m}_k}{\partial \omega_{TK}}\right) \omega_{TK\ 0}} \tag{23}$$

$$\theta_{c.ad\ \Delta\omega} = \frac{\left(\frac{\partial \dot{m}_{k,cil}}{\partial \omega}\right) \omega_{\,0}}{\left(\frac{\partial \dot{m}_k}{\partial \omega_{TK}}\right) \omega_{TK\ 0}} \tag{24}$$

$$\theta_{c.ad\ \Delta h_k} = \frac{\left(\frac{\partial \dot{m}_{k,cil}}{\partial h_k}\right) h_{k\,0}}{\left(\frac{\partial \dot{m}_k}{\partial \omega_{TK}}\right) \omega_{TK\ 0}} \tag{25}$$

$$\theta_{c.ad\ \Delta T_k} = \frac{\left(\frac{\partial \dot{m}_{k,cil}}{\partial T_k}\right) T_{k\,0}}{\left(\frac{\partial \dot{m}_k}{\partial \omega_{TK}}\right) \omega_{TK\ 0}} \tag{26}$$

$$T_{c.ad} = \frac{V_{c.ad} \cdot p_{k\,0}}{n_{TK} \cdot p_k \left(\frac{\partial \dot{m}_k}{\partial \omega_{TK}}\right) \omega_{TK\,0}} \tag{27}$$

$$F_{st\ \_\ c.ad} = \left(\frac{\partial \dot{m}_{K,cil}}{\partial p_k}\right) - \left(\frac{\partial \dot{m}_K}{\partial p_k}\right) \tag{28}$$

$$K_{c.ev} = \frac{F_{st\ c.ev}\ p_{T\ 0}}{\left(\frac{\partial \dot{m}_g}{\partial \omega_{TK}}\right) \omega_{TK\ 0}} \tag{29}$$

$$\theta_{c.ev\,\Delta Ccycle} = \frac{\left(\frac{\partial \dot{m}_T}{\partial c_{cycle}}\right) c_{cycle\ 0}}{\left(\frac{\partial \dot{m}_g}{\partial \omega_{TK}}\right) \omega_{TK\ 0}} \tag{30}$$

$$\theta_{c.ev\,\Delta h_T} = \frac{\left(\frac{\partial \dot{m}_T}{\partial h_T}\right) h_{T\,0}}{\left(\frac{\partial \dot{m}_g}{\partial \omega_{TK}}\right) \omega_{TK\ 0}} \tag{31}$$

$$\theta_{c.ev\,\Delta p_k} = \frac{\left(\frac{\partial \dot{m}_T}{\partial p_k}\right) p_{k0}}{\left(\frac{\partial \dot{m}_g}{\partial \omega_{TK}}\right) \omega_{TK\,0}} \quad (32)$$

$$T_{c.ev} = \frac{V_{c.ev} \cdot p_{T0}}{n_{TK} \cdot p_T \left(\frac{\partial \dot{m}_g}{\partial \omega_{TK}}\right) \omega_{TK0}} \quad (33)$$

$$F_{st\_c.ev} = \left(\frac{\partial \dot{m}_T}{\partial p_T}\right) - \left(\frac{\partial \dot{m}_g}{\partial p_T}\right) \quad (34)$$

J_{TK} (kgm^2) - mechanical moment of inertia of the rotating mechanical turbocharger components, reduced to its revolution axis:
ω_{TK} (s^{-1}) - common angular speed of the compressor and turbine;
ω (s^{-1}) - angular engine speed;
J_{eng} (kgm^2) = mechanical momentuum inertia of the mobile mechanical engine, reduced to it's revolution axis;
M_{te} (Nm) = torque of engine;
$M_{te\,R}$(Nm) – torque to drive, the couple must overcome the brake resistance brake;
M_{tK} (Nm)-- brake torque of compressor ;
M_{tTG} (Nm)-- torque of gas turbine;
$\dot{m}_k$ (kg/s) - compressor mass flow rate;
$\dot{m}_T$ (kg/s) - gas turbine mass flow rate;
h = the position of adjustment devices for injection pump;
h_T – actuator turbine control position;
h_K – actuator compressor control position;
$c_{cycle,0}$(kg fuel/cycle) – fuel consumption on cycle at stationary running;
p_T(N/m^2) - gas pressure in the turbine entry;
p_K(N/m^2) - fluid pressure at the compressor exit;
T_k(K) - fluid temperature at the exit of the compressor;
T_T(K) - gas temperature at the entrance of gas turbine;
c_{cycle}(kg fuel/cycle) – fuel consumption on cycle;
'0' index indicates stationary operating regimes;
h_{K0} = actuator compressor control position at steady working conditions;
h_{T0} = actuator turbine control position at steady working conditions;
ω_{TK0} (s^{-1}) - common speed of the compressor and turbine at stationary running;
M_{teTG0} (Nm) - turbine shaft torque at stationary running;
M_{teK0} (Nm) - compressor shaft torque at stationary running;
p_{T0}(N/m^2) - gas pressure in the turbine entry at stationary running;
p_{K0}(N/m^2) - fluid pressure at the compressor exit at stationary running;

If the turbocharger has not adequate adjustable devices (actuators), in the equations (1,2,3,4,5) particular condition is $\Delta h_T^* = 0, \Delta h_K^* = 0$ respectively (1',2',3',4',5') $\Delta H_T^* = 0, \Delta H_K^* = 0$ and results equations:

$$\left(T_{eng} \cdot s + K_{eng}\right) \cdot \Delta\Omega^*(s) - \Delta C^*_{cycle}(s) - \theta_{eng\Delta p_k} \cdot \Delta P_K^*(s) = -\theta_{eng\Delta hs} \cdot \Delta H_s^*(s) \quad (35)$$

$$-\theta_{ap.inj\,\Delta\omega} \cdot \Delta\Omega^*(s) + K_{ap.inj} \cdot \Delta C^*_{cycle}(s) = \Delta H^*(s) \quad (36)$$

$$-\theta_{TK\Delta C_{cycle}} \cdot \Delta C^*_{cycle}(s) + \theta_{TK\Delta p_k} \cdot \Delta P_K^*(s) - \Delta P_T^*(s) + \left(T_{TK} \cdot s + K_{TK}\right) \cdot \Delta\Omega^*_{TK}(s) = 0 \quad (37)$$

$$\theta_{c.ad\Delta\omega} \cdot \Delta\Omega^*(s) + \left(T_{c.ad} \cdot s + K_{c.ad}\right) \cdot \Delta P^*_K(s) - \Delta\Omega^*_{TK}(s) = 0 \quad (38)$$

$$-\Delta\Omega^*(s) - \theta_{c.ev\Delta c_{cycle}} \cdot \Delta C^*_{cycle}(s) - \theta_{c.ev\Delta p_k} \cdot \Delta P_k^*(s) + \left(T_{c.ev} \cdot s + K_{c.ev}\right) \cdot \left(\Delta P^*_T(s)\right) = 0 \quad (39)$$

If we want to obtain the differential equations of the internal combustion engines without the adequate adjustment devices for T and K , it is necessary to solve the particular equations system resulting from the general equations 35, 36, 37, 38, 39.

Figures 2, 3, 4, 5, 6 show the structural diagrams for the turbocharger engine itself, the fuel injection system, the intake manifold, the exhaust manifold and the turbocharger. In these diagrams the transfer functions are set out in rectangles, the input signals on the left and the output signals on the right. The index number added to some input signals represents the number of the structural diagram where that signal comes out.

We consider the independent parameters Δh_s^* and Δh^* [8], and the unknown quantities $\Delta\omega^*, \Delta c^*_{cycle}, \Delta p_K^*, \Delta p_T^*, \Delta\omega^*_{TK}$.

For the internal combustion engine as the subject of automatic adjustment abiding by the revolution, $\Delta\omega^*$ is considered to be the parameter which must be pursued in time:

$$\Delta\Omega^*(s)=\frac{\Delta_{\Delta\Omega^*}}{\Delta};\ \Delta C^*_{cycle}(s)=\frac{\Delta_{\Delta C^*_{cycle}}}{\Delta}$$

$$\Delta P^*_K(s)=\frac{\Delta_{\Delta P^*_K}}{\Delta};\ \Delta P^*_T(s)=\frac{\Delta_{\Delta P^*_{TK}}}{\Delta} \qquad (40)$$

$$\Delta\Omega^*_{TK}(s)=\frac{\Delta_{\Delta\Omega^*_{TK}}}{\Delta}$$

Where:

$$\Delta=T^2_{eng\_s_2}\cdot s^2+T_{eng\_s_1}\cdot s+K_{eng\_s}$$

$$\Delta_{\Delta\Omega^*}=\left(T_{act\_inj\_pump}\cdot s+\theta_{act\_inj\_pump}\right)\cdot \Delta H^*(s)-\left(T_S\cdot s+\theta_S\right)\cdot\Delta H^*_S(s) \qquad (41)$$

$$T_{eng\_s_2}=T_{eng}T_{TK}K_{ap.inj}K_{cad}K_{cev} \qquad (42)$$

The differential equation written in the operational form, where the rotation is considered as the object of automatic adjustment, is as follows:

$$\Delta\cdot\Delta\Omega^*(s)=\Delta_{\Delta\Omega^*} \qquad (43)$$

Or:

$$\left(T^2_{eng\_s_2}\cdot s^2+T_{eng\_s_1}\cdot s+K_{eng\_s}\right)\cdot\Delta\Omega^*= =\left(T_{act\_inj\_pump}\cdot s+\theta_{act\_inj\_pump}\right)\cdot\Delta H^*(s)- -\left(T_s\cdot s+\theta_s\right)\cdot\Delta H^*_s(s) \qquad (44)$$

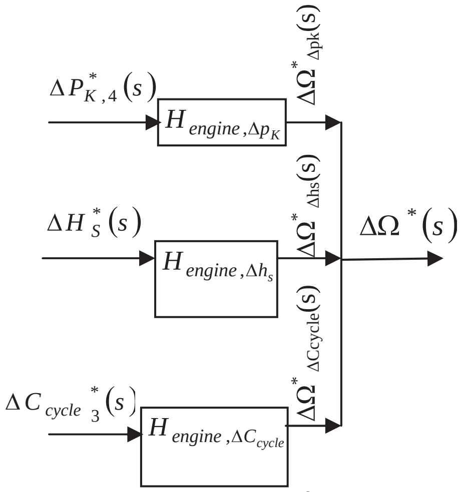

Fig. 2 Structural diagram for the turbocharger engine itself

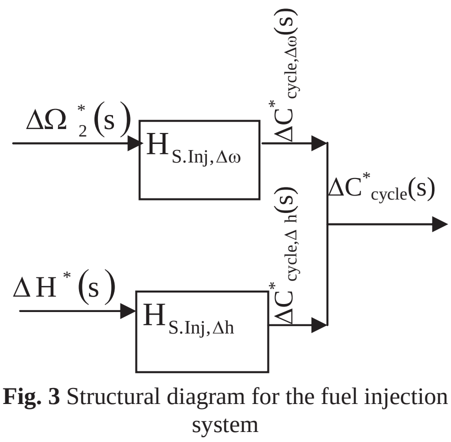

Fig. 3 Structural diagram for the fuel injection system

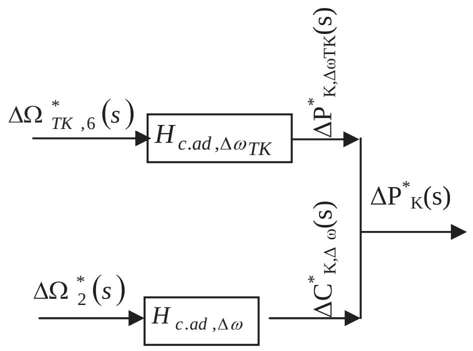

Fig. 4 Structural diagram for the intake manifold

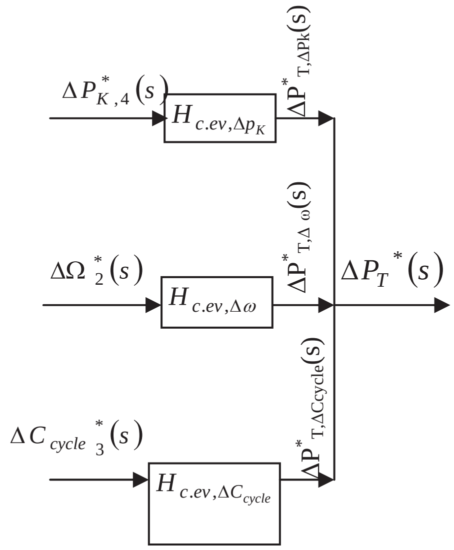

Fig. 5 Structural diagram for the exhaust manifold

It therefore leads to:

$$\Delta\Omega^*(s)=\frac{\left(T_{act\_inj\_pump}\cdot s+\theta_{act\_inj\_pump}\right)\cdot\Delta H^*(s)}{\left(T^2_{eng\_s_2}\cdot s^2+T_{eng\_s_1}\cdot s+K_{eng\_s}\right)}- -\frac{\left(T_S\cdot s+\theta_S\right)}{\left(T^2_{eng\_s_2}\cdot s^2+T_{eng\_s_1}\cdot s+K_{eng\_s}\right)}\cdot\Delta H^*_S(s)= =H_{engine\_\Delta h}(s)\cdot\Delta H^*(s)-H_{engine\_\Delta h_S}(s)\cdot\Delta H^*_S(s); \qquad (45)$$

where:

$H_{engine\ \Delta h}(s)$ = the transfer function determined by the actuator of the injection pump;

$H_{engine\ \Delta hs}(s)$ = the transfer function determined by the load engine;

If Laplace^{-1} is applied to the differential equation (44) written above, the result is the differential equation expressing the dynamic engine operation in relation to time (equation 46), for the supercharged internal combustion engine without adjustment actuators for the turbine and compressor.

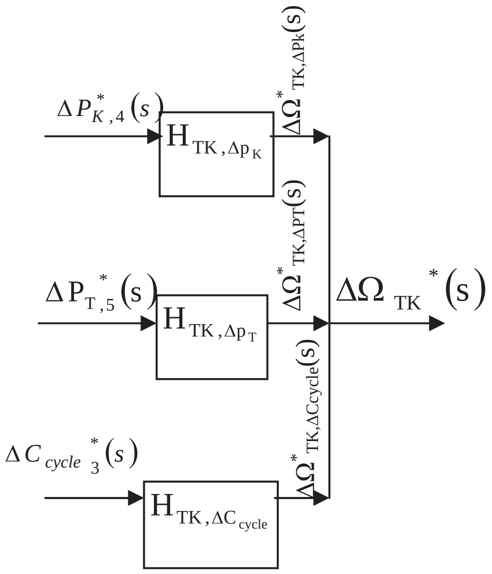

Fig. 6 Structural diagram for the turbo-charger

$$T_{eng\_s_2}\frac{d^2\left(\Delta\omega^*\right)}{d\tau^2}+T_{eng\_s_1}\frac{d\left(\Delta\omega^*\right)}{d\tau}+ K_{eng\_s}\Delta\omega^* = T_{act\_inj\_pump}\frac{d\left(\Delta h^*\right)}{d\tau}+ \theta_{act\_inj\_pump}\Delta h^* - T_s\frac{d\left(\Delta h^*{}_s\right)}{d\tau}-\theta_s\Delta h_s^* \quad (46)$$

$$T_{eng\_s_2} = T_{eng}T_{TK}K_{ap.inj}K_{cad}K_{cev} \quad (47)$$

$$T_{act\_inj\_pump} = T_{TK}K_{c.ad}K_{c.ev} \quad (48)$$

$$T_S = T_{TK}K_{ap.inj}K_{c.ad}K_{c.ev}\theta_{eng\Delta h_s} \quad (49)$$

$$\theta_S = K_{ap.inj}\theta_{eng\Delta h_s}\cdot \left(K_{TK}K_{c.ad}K_{c.ev}+K_{c.ev}\theta_{TK\Delta p_K}-\theta_{c.ev\Delta p_K}\right) \quad (50)$$

$$\theta_{act\_inj\_pump} = \left(K_{TK}K_{c.ad}+\theta_{TK\Delta p_K}\right)K_{c.ev} + \left(K_{c.ev}\theta_{TK\Delta C_{cycle}}-\theta_{c.ev\Delta c_{cycle}}\right)\theta_{eng\Delta p_K}-\theta_{c.ev\Delta p_K} \quad (51)$$

Figure 7 shows the structural diagram of the diesel supercharged engine [8], considered as the rotation of the automatic adjustment object.

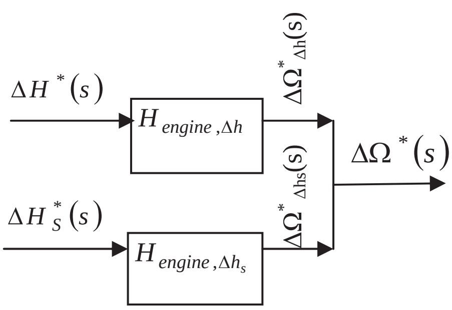

Fig. 7 Structural diagram for the supercharged internal combustion engine

Taking into account the different charge schemes and mathematical models of the engine subsystems, the result yields different mathematical models represented by differential equations up to grade 5. Based on these models, the dynamic engine operations of supercharged engines can be obtained.

3. CONCLUSION

On the basis of the mathematical model of the dynamic regimes of the supercharged engine and the transfer functions established, the following conclusions can be specified:

-- Transfer functions would allow the achievement of an automatic system control located on the engine, which can control all dynamic and stationary regimes and can ensure the working of the engine on optimum regimes with minimum possible fuel consumption.

-- Using an automatic system control for engine, the system will ensure quality indices for all the dynamic regimes of the whole engine and a high system performance on stationary regimes of operation, with minimum errors for speed and acceleration. The automatic system control will ensure performance for the whole engine and all its subsystems, during transitory regimes of operation.

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