TECHNICAL UNIVERSITY OF CLUJ-NAPOCA

ACTA TECHNICA NAPOCENSIS

Series: Applied Mathematics, Mechanics, and Engineering
Vol. 68, Issue IV, November, 2025

MODELING OF OPERATIONS IN DYNAMIC REGIMES OF SUPERCHARGED DIESEL ENGINES USING TRANSFER FUNCTIONS

Salvadore-Mugurel BURCIU

Abstract: *In the paper is presented a mathematical model based on the transfer functions which is used for establishing the differential equation which expresses the dynamic behavior on working of supercharged internal combustion engines. Mathematical model is based on knowledge of the working characteristics for the stationary regimes and non-stationary ones in their vicinity, for the engines subsystems, such as engine without auxiliary systems, engine turbocharger, exhaust and intake manifold and also the fuel injection system. Are presented transfer functions for each engine subsystems and also like whole supercharged engine, functions that are used for adjusting automatic command and control systems of the engines.*

Key words: *dynamic regimes of working, transfer functions, supercharged engine.*

1. INTRODUCTION

In the paper is presented the mathematical model for calculate the transfer function for the whole supercharged internal combustion engine. According to the paper [1], for the dynamic behavior on working of a supercharged diesel engine, there is a need for a model that is more advanced then a first order system.

The results of the model presented in [1] show that the diesel engine behaves like a second order system when operating in the power regimes (governor area and in vecinity) and more like a first order system in the regimes with constant torque (overload regimes) area and in their vicinity.

Mathematical model presented in this paper is based on knowledge of the working characteristics for the stationary regimes and non-stationary ones in their vicinity, the differential equations and the transfer functions established in papers [2], [3], [4], [5], [6], [7], [8] for dynamic behavior of supercharged diesel engine subsystems, as they are exhaust and intake manifold, injection system, engine itself, turbocharger [8], [9], as shown in Figure 1.

On the basis of these equations results the dynamical behavior of whole supercharged internal combustion engine. Using transfer functions we can adjusting automatic command and control systems of the engines, such as automatic governor for revolution or another automatic controllers.

2. DYNAMIC CHARACTERISTICS AND TRANSFER FUNCTIONS OF DIESEL SUPERCHARGED ENGINE

Consider the next diagram (Figure 1) with the component parts of the supercharged internal combustion engine [8], [9], supercharged with free rotation turbocharger.

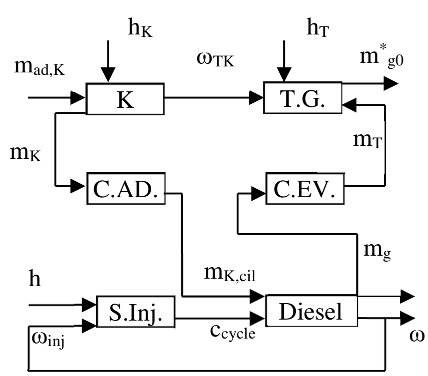

Fig.1. Diagram for supercharged engine subsystems

Received: 30.09.25; **Similarities**: 21.10.25: **Reviewed**: 22.10./26.10.25: **Accepted**:12.11.25.

In these diagrams, subsystems are:
K - turbocompressor; S. Inj. - Fuel injection system; TG - gas turbine; Diesel - diesel engine itself; C.AD. - intake manifold; C.EV. - exhaust manifold; ω - angular speed of crankshaft; ω_{TK} - angular speed of turbocharger shaft; ω_{inj} - angular speed of fuel injection pump shaft; c_{cycle} fuel consumption for a cycle; $h_{K,T}$ - the position of adjustment actuator of compressor respectively turbine;
Where: $m_{ad,K}$ - mass flow of air aspirated into the compressor; m_K - mass flow of air on pressure entering the intake manifold; $m_{K,cil}$ - mass flow of air on pressure which entering the engine cylinders; h - actuator position of the injection pump for engine; m_g – exhaust gas mass flow in the exhaust manifold; m_T - exhaust gas mass flow entering in the turbine; m^*_{g0} - exhaust gas flow coming out in the environment.
On the basis of the differential equations (on [2], [3], [4], [5], [6], [7]) which simulate the dynamical behaving of constitutive parts of engine, it's possible to achieve the differential equation for the dynamical behavior of whole supercharged engine, which would represent a mathematical model for simulating real engine operation.
According with [2], [3], [4], [5], [6], [7] can write:
-- for Diesel engine own:

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

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 fuel 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:

$$\left(T_{c.ev}\, s + K_{c.ev}\right)\left(\Delta P^{*}_{T}(s)\right) = \\ = \Delta\Omega^{*}(s) + \theta_{c.ev\,\Delta p_k}\Delta P_k^{*}(s) + \\ \theta_{c.ev\,\Delta c_{cycle}}\Delta C^{*}_{cycle}(s) - \\ - \theta_{c.ev\,\Delta h_T}\Delta H^{*}_{T}(s) \tag{5'}$$

Where:

$$\omega = \omega_0 + \Delta\omega; \Delta\omega^{*} = \frac{\Delta\omega}{\omega_0} \tag{6}$$

$$\Delta c^{*}_{cycle} = \frac{\Delta c_{cycle}}{c_{cycle0}}; \Delta p^{*}_{K} = \frac{\Delta p_K}{p_{K0}} \tag{7}$$

$$\Delta h^{*}_{S} = \frac{\Delta h_S}{h_{S0}} = \Delta h^{*}_{R} \tag{8}$$

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

$$T_{eng} = \frac{J_{eng}\cdot\omega_0}{\left(\frac{\partial M_{te}}{\partial c_{cycle}}\right)_{\omega_0, p_{k0}}\cdot c_{cycle\,0}} \tag{9}$$

$$\theta_{eng\,\Delta p_k} = \frac{\left(\frac{\partial M_{te}}{\partial p_k}\right)\cdot p_{k0}}{\left(\frac{\partial M_{te}}{\partial c_{cycle}}\right)_{\omega_0, p_{k0}}\cdot c_{cycle\,0}} \tag{10}$$

$$K_{eng} = \frac{F_{st\_eng}\cdot\omega_0}{\left(\frac{\partial M_{te}}{\partial c_{cycle}}\right)_{\omega_0, p_{k0}}\cdot c_{cycle\,0}} \tag{11}$$

$$\theta_{eng\,\Delta h_s} = \frac{\left(\frac{\partial M_{teR}}{\partial h_S}\right)_{\omega_0}\cdot h_{s0}}{\left(\frac{\partial M_{te}}{\partial c_{cycle}}\right)_{\omega_0, p_{k0}}\cdot c_{cycle\,0}} \tag{12}$$

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

$$K_{TK} = \frac{F_{st\,TK}\,\omega_{TK\,0}}{\left(\frac{\partial M_{tTG}}{\partial p_T}\right) p_{T0}} \tag{14}$$

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

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

$$\theta_{TK\,\Delta h_K} = \frac{\left(\frac{\partial M_{tK}}{\partial h_K}\right) h_{K0}}{\left(\frac{\partial M_{tTG}}{\partial p_T}\right) p_{T0}} \tag{17}$$

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

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

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

$$\theta_{ap\_inj\,\Delta\omega} = \frac{\left(\frac{c_{cycle}}{\partial\omega}\right)\omega_0}{\left(\frac{\partial c_{cycle}}{\partial h}\right) h_0} \tag{21}$$

$$\theta_{TK\,\Delta C_{cycle}} = \frac{\left(\frac{\partial M_{tTG}}{\partial c_{cycle}}\right) c_{cycle\,0}}{\left(\frac{\partial M_{tTG}}{\partial p_T}\right) p_{T0}} \tag{22}$$

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

$$\theta_{c.ad\,\Delta\omega} = \frac{\left(\dfrac{\partial \dot{m}_{k,cil}}{\partial \omega}\right)\omega_0}{\left(\dfrac{\partial \dot{m}_k}{\partial \omega_{TK}}\right)\omega_{TK\,0}} \qquad (24)$$

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

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

$$T_{c.ad} = \frac{V_{c.ad} \cdot p_{k0}}{n_{TK} \cdot p_k \left(\dfrac{\partial \dot{m}_k}{\partial \omega_{TK}}\right)\omega_{TK0}} \qquad (27)$$

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

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

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

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

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

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

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

J_{TK} (kgm^2) - mechanical moment of inertia for parts of turbocharger, reduced to its revolution axis; ω_{TK} (s^{-1}) - rotation speed of the compressor and turbine of turbocharger;
ω (s^{-1}) - rotation engine speed;
J_{eng} (kgm^2) = mechanical moment of inertia of the engine, reduced to its revolution axis; M_{te} (Nm) = torque of engine; $M_{te,R}$(Nm) – the resisting torque that must be overcome by the engine torque; M_{tK} (Nm)-- resisting 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 adjusting device for injection pump; h_T – position of actuator turbine control; h_K – position of actuator compressor control;
$c_{cycle,0}$(kg fuel/cycle) – fuel consumption on cycle for stationary running;
p_T(N/m^2) – exhaust gas pressure in the turbine entry; p_K(N/m^2) - fluid pressure at the compressor exit;
T_k(K) - fluid compressed temperature at the exit of the turbocharger compressor; T_T(K) – exhaust gas temperature at the entrance of turbocharger gas turbine;

c_{cycle}(kg fuel/cycle) – fuel consumption on cycle for variable speed regimes (non-stationary regimes);
'0' index indicates stationary operating regimes;
h_{K0} = position of actuator compressor control at steady working conditions;
h_{T0} = position of actuator turbine control at steady working conditions;
ω_{TK0} (s^{-1}) - rotation speed of the compressor and turbine of turbocharger at stationary working conditions;
M_{teTG0} (Nm) - turbine shaft torque at stationary running; M_{teK0} (Nm) - compressor shaft torque at stationary running;
p_{T0}(N/m^2) – exhaust 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 hasn't adequate devices (actuators) for adjustment, 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 result equations:

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

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

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

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

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

To obtain the differential equations of the supercharger internal combustion engines without the adequate adjustment devices for turbine and compressor, it is necessary to solve the particular equations system resulting from the general equations 35, 36, 37, 38, 39.

In Figures 2, 3, 4, 5, 6 are presented the structural diagrams for the supercharger engine without systems and for 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 $\Delta\omega^*, \Delta c^*_{cycle}, \Delta p^*_K, \Delta p^*_T, \Delta\omega^*_{TK}$.

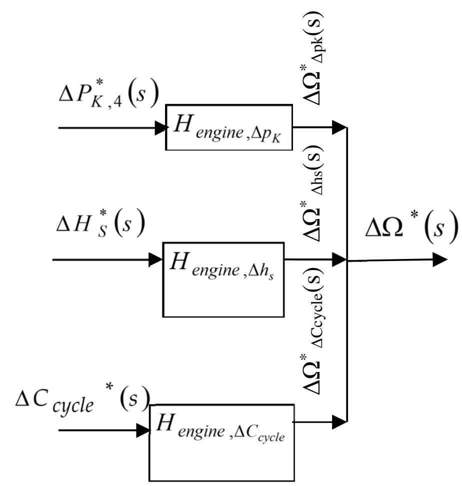

Fig. 2. Structural diagram for the turbocharger engine itself

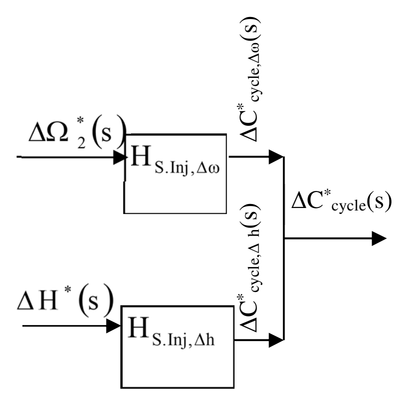

Fig. 3. Structural diagram for the fuel injection system

For the internal combustion engine as the subject of automatic adjustment abiding by the revolution, we consider $\Delta\omega^{*}$ 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:

$$\Delta=T^{2}_{eng\_s_2}\cdot s^{2}+T_{eng\_s_1}\cdot s+K_{eng\_s}$$
$$\Delta_{\Delta\Omega^{*}}=\left(T_{act\_inj\_pump}\cdot s+\theta_{act\_inj\_pump}\right)\cdot \qquad (41)$$
$$\cdot\Delta H^{*}(s)-\left(T_{s}\cdot s+\theta_{s}\right)\cdot\Delta H^{*}_{s}(s)$$

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

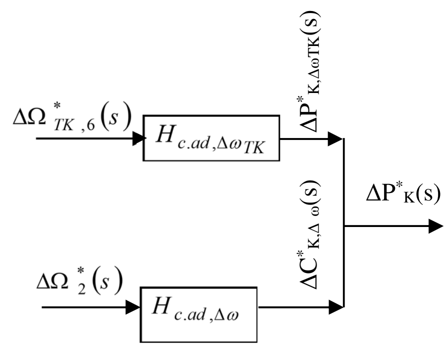

Fig. 4. Structural diagram for the intake manifold

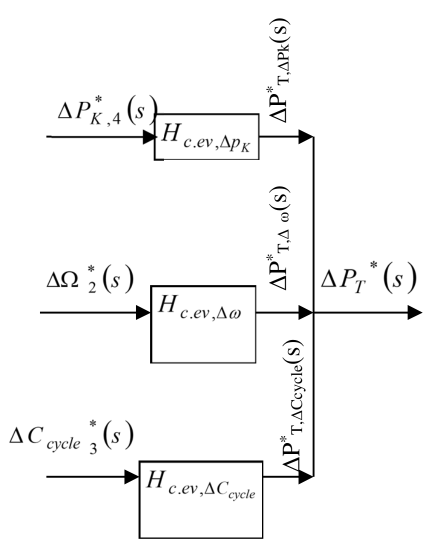

Fig. 5. Structural diagram for the exhaust manifold

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

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

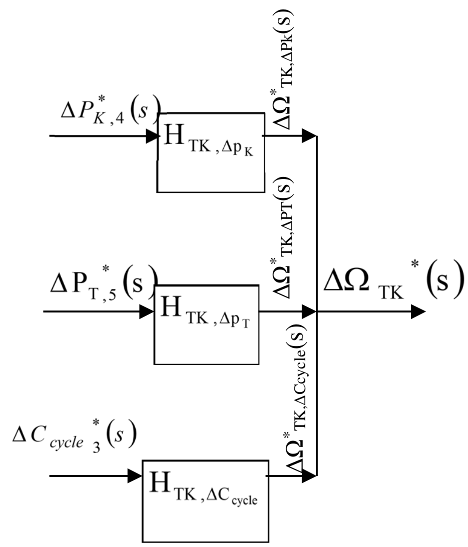

Fig. 6. Structural diagram for the turbocharger

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

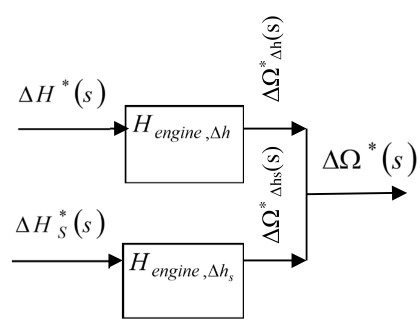

Fig. 7. Structural diagram for the supercharged internal combustion engine

Relation (44) leads to:

$$\Delta\Omega^*(s)=\frac{\left(T_{act\_inj\_pump}\cdot s+\theta_{act\_inj\_pump}\right)\cdot\Delta H^*(s)}{\left(T^2_{eng\_s_2}\cdot s^2+T_{eng\_s_1}\cdot s+K_{eng\_s}\right)}-$$
$$-\frac{\left(T_s\cdot s+\theta_s\right)}{\left(T^2_{eng\_s_2}\cdot s^2+T_{eng\_s_1}\cdot s+K_{eng\_s}\right)}\cdot\Delta H^*_s(s)= \quad (45)$$
$$=H_{engines\Delta h}(s)\cdot\Delta H^*(s)-H_{engines\Delta h_s}(s)\cdot\Delta H^*_s(s);$$

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

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

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

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

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

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

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

Figure 7 shows the structural diagram of the diesel supercharged engine [8], considerate the rotation as the size for automatic adjustment.
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 behaviour of supercharged engines can be obtained.

3. CONCLUSION

The purpose of the paper was to present the mathematical model and establishing the differential equation expressing the dynamic behavior of turbocharged internal combustion engines which use turbocompressor with free rotation, aim which was achieved by establishing the differential equation 46.

On the basis of the mathematical model of the dynamic behaviour of the supercharged engine and using the transfer functions established, the following can be done:
-- optimal tuning the automatic speed regulator located on the engine.
-- establish performance of the whole system and its subsystems, during transitory regimes of operation.
-- obtained the quality indices for the dynamic behaviour of the whole system.
-- performance indication of system in stationary regimes of operation, establish speed and acceleration deviations.

4. REFERENCES

[1] Miedema, S.A., & Lu, Z., *The Dynamic Behavior of a Diesel Engine*, Proc. WEDA XXII Technical Conference & 34th Texas A&M Dredging Seminar, June 12-15 2002 Denver, Colorado, USA.

[2] Burciu, M., *Calculation of the unsteady mechanical and gasothermodynamic processes in the free rotation supercharger units of the supercharged internal combustion engines*, Bulletin of the Transilvania University of Brasov, vol 2 (51) series I, 2009.

[3] Dumitru, Gh., Burciu Burciu, S.,M., *Dynamical Characteristics of Supercharged Internal Combustion Engine Itself*, The scientific session in University " Dunarea de Jos " , Galati 1992 .

[4] Dumitru, Gh., *Dynamical Characteristics of Admission Pipe for Internal Combustion Engine* , The scientific session in University " Dunarea de Jos ", Galati 1993.

[5] Dumitru, Gh., *Dynamical Characteristics of Evacuation Pipe for Internal Combustion Engine*, The scientific session in University " Dunarea de Jos ", Galati 1993.
[6] Dumitru Gh., Burciu, S.,M. , *Dynamical Characteristics of Turbocharger for Internal Combustion Engine*, The scientific session in University " Dunarea de Jos " , Galati 1994.
[7] Dumitru Gh., Burciu, S.,M., *Dynamical Characteristics of Injection System for Internal Combustion Engine*, The scientific session in University "Dunarea de Jos", Galati 1995.
[8] Burciu M.S, Thesis Ph.D., *Contributions to the study of nonstationary processes in the internal combustion engine overloading turbo-compressors*, University of Galati, Faculty of Mechanical Engineering, Department of Internal Combustion Engines, 2000.
[9] Burciu, M., *Internal Combustion Piston Engines – Thermodynamic Processes, Turbocharger, Operating Characteristics and Engine Installations*, Europlus Publishing, Galati 2006.

Comportamentul dinamic al motoarelor diesel cu ardere internă supraalimentate utilizand funcțiile de transfer

Abstract: În lucrare este prezentat un model matematic bazat pe funcțiile de transfer, utilizat pentru stabilirea ecuației diferențiale ce exprimă comportamentul dinamic în funcționare al motoarelor cu ardere internă supraalimentate. Modelul matematic se bazează pe cunoașterea caracteristicilor de funcționare pentru regimurile staționare și nestaționare din vecinătatea lor, pentru subsistemele motorului, cum ar fi motorul fără sisteme auxiliare, turbocompresorul motorului, galeria de evacuare și de admisie, precum și sistemul de injecție al combustibilului. Sunt prezentate funcțiile de transfer pentru fiecare subsistem al motorului, precum și pentru întregul motor supraalimentat, funcții utilizate pentru reglarea sistemelor automate de comandă și control ale motoarelor.

Salvadore-Mugurel BURCIU, Associate Professor Phd., Faculty of Engineering, Thermal Systems and Automotives Department, Dunarea de Jos University, Galati, Romania; Mugurel.Burciu@ugal.ro.