12th International Research/Expert Conference
”Trends in the Development of Machinery and Associated Technology”
TMT 2008, Istanbul, Turkey, 26-30 August, 2008

MODELING OF COUPLING TURBOCHARGER AND DIESEL ENGINE

Dr. sc. Naser Lajqi
Dr. sc. Bashkim Baxhaku
Dip. ing. Shpetim Lajqi
University of Prishtina, Mechanical Engineering Faculty
Prishtina, Republic of Kosova

ABSTRACT
Development of IC engines is limited with reduction of pollutant emission and CO_2. Passenger car with diesel IC engine approximately emit 25 % less CO_2 compare with Otto engines. Efficient way which currently uses to reduce the fuel consumption is based in “downsizing” concept, i.e. reduction cylinder volume of IC engines and power to be same or to be higher. Key component suchlike concepts are supply fuel system and turbocharged diesel IC engine.
This paper deals with process modeling turbocharged diesel engine. During the modeling process, the more sensitive problem is how to couple the diesel IC engine, as a typical cyclic machine, with a turbocharger, as a representative of flow machines. Beside the description of turbocharger model in this paper also are represented results from calculation of pressure, temperature and turbocharger rotor speed for one turbocharged diesel engine. All of these will be shown in this paper.
Keywords: IC engine, turbo charging, rotor speed, simulation, modeling, manifold, etc.

1. INTRODUCTION
Nowadays technical level of development internal combustion engines has difficulty for still reduction of fuel consumption. To achieve objective ACEA (European Automobile Manufacturers Association) for decreases emission of CO_2 in 140 g/km, must used different ways. One of popular method is decreasing displacement engine but power to be same or to increase. Key factor of this method is turbocharged system at diesel engine. Engine brake power (P_e) with four strokes engine depends:
- Crankshaft rotational speed (n_e),
- break mean effective pressure (p_e) and
- displaced or swept volume (V_h).

Therefore brake power can be estimated with equation:

$$P_e = \frac{p_e \cdot V_h \cdot n_e}{4} \qquad \text{... (1)}$$

Beyond increasing crankshaft rotational speed is limited due inertial forces and friction losses (friction power) are proportionally with mean piston speed at square (v^2) according to Millington and Hartles. Therefore is needed to increase only break mean effective pressure.

2. MODELING OF COUPLING TURBOCHARGER AND DIESEL ENGINE
In literature which analyze problems of turbocharged and his coupling with internal combustion engine are given a lot of methods for calculation of engine working points and turbocharger. Used

methods can be graph analytic and other methods which cover simulation process with support of computers respectively numerically methods [1, 2, 3].

In figure 1 is presented model of turbocharged diesel engine through the models are defined adapted model for numerically select of coupling engine – turbocharger.

Based on the laws for ideal gas, conservation of mass and first law of thermodynamics for open system are:

$$p = R \cdot \rho \cdot T\,, \qquad \dot{m} = \sum_j \dot{m}_j \quad \text{and}$$

$$\dot{E} = \sum_j m_j h_j - Q_w - \dot{W} \qquad \text{... (2)}$$

Equations for pressure and temperature variability in the cylinder:

$$\dot{p} = \frac{\rho}{\partial\rho/\partial p}\cdot\left(-\frac{\dot{V}}{V} - \frac{1}{\rho}\frac{\partial\rho}{\partial T}\cdot - \frac{1}{\rho}\cdot\frac{\partial\rho}{\partial\Phi}\cdot\dot{\Phi} + \frac{\dot{m}}{m}\right) \ldots(3)$$

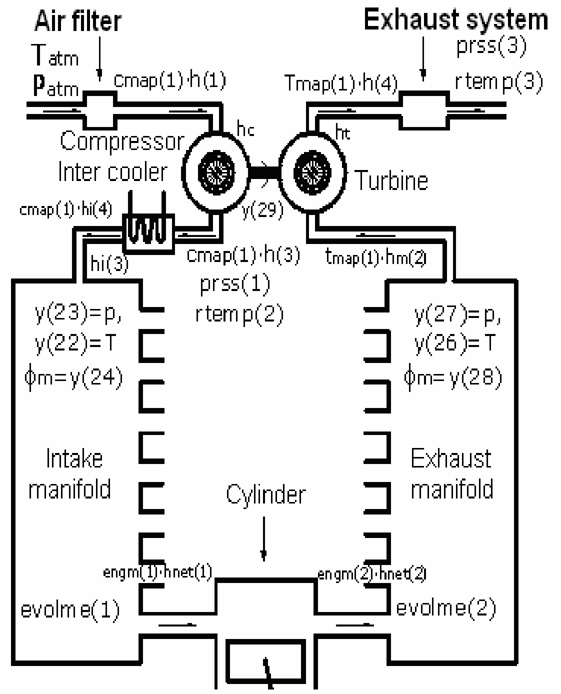

Figure 1. Model of turbocharged diesel engine

$$\dot{T} = \frac{B}{A}\cdot\left(\frac{\dot{m}}{m}(1-\frac{h}{B}) - \frac{\dot{V}}{V} - \frac{C}{B}\cdot\dot{\Phi} + \frac{1}{B\cdot m}(\sum_j \dot{m}_j \cdot h_j - \dot{Q}_W)\right) \qquad \text{... (4)}$$

Gas exchange process (intake and exhaust) is modeled with one dimensional quasi – steady flow equation:

$$\dot{m} = c_d \cdot A_{gj} \cdot \frac{p_o}{R\cdot T_o}\cdot\sqrt{\gamma\cdot R\cdot T_o}\left\{\frac{2}{\gamma-1}\left[(\frac{p_s}{p_o})^{\frac{2}{\gamma}} - (\frac{p_s}{p_o})^{\frac{(\gamma+1)}{\gamma}}\right]\right\}^{\frac{1}{2}} \qquad \text{... (5)}$$

Because intake manifold are assumed with great volume, in which are connected some cylinder, mass flow through compressor can be take as constant.

Above equation which takes overall turbocharged diesel engine model, can be extended through dynamic equation of turbocharger.

The rate of change of the mechanical energy of the turbocharger rotor, ($\dot{E}_{t/c}$) depends on the difference the power required to drive the compressor (negative) and the power delivered by the turbocharger turbine (positive):

$$\dot{E}_{t/c} = \dot{W}_{turbin} - \dot{W}_{kompresor} \qquad \text{... (6)}$$

The change in mechanical energy relates to the change in the rotor speed according to the turbocharger dynamics, i.e.,

$$\dot{E}_{t/c} = J\cdot\dot{\omega}\cdot\omega + B\cdot\omega^2 \qquad \text{... (7)}$$

Where:

J, kgm^2	- rotational inertia of turbocharger,
B, $kg\cdot m^2 s^{-1}$	- rotational damping of turbocharger and
ω, s^{-1}	- angular velocity.

The compressor and the turbine power can be expressed as:

$$\dot{W}_{kompresor} = \dot{m}_c (h_3 - h_1), \qquad \dot{W}_{turbin} = \dot{m}_t (h_4 - h_{m2}) \qquad \text{... (8)}$$

Solving for the change in the speed $\dot{\omega}$ gives:

$$\dot{\omega} = \left[\dot{m}_c \cdot (h_3 - h_1) + \dot{m}_t \cdot (h_4 - h_{m2}) - B \cdot \omega^2 \right] / J \cdot \omega^2 \qquad \text{... (9)}$$

According to above equation for all processes of turbocharged diesel engine can be defined according to the change of working mass m, pressure, temperature (compressor, intercooler, intake manifold, cylinder, exhaust manifold, turbine), rotor speed of turbocharger and another important parameters which categorized this process. Below are presented simulation results of process in turbocharger diesel engine given from presented model, respectively used program in software MATLAB [3].

3. SIMULATION RESULTS

For verification of model described above we have used data for turbocharger diesel engine: **Cummins** type **N14-M**, which are shown in table 1 and table 2.

Table 1. Engine parameters

Describe of engine	Turbocharger diesel engine, intercooler, water cooling system
Bore	5.5 inch = 139 mm
Stroke	6 inch = 152 mm
Connecting road length	12 inch = 304.8 mm
Compression ratio	17 : 1
Number of cylinders	6
Engine swept volume	14.0 l
Intake manifold volume	5.5 l
Exhaust manifold volume	7.8 l
Injection timing	17 deg BTC
Intake valve opens	11 deg BTC
Intake valve closes	32 deg ABC
Exhaust valve opens	35 deg BBC
Exhaust valve closes	16 deg ATC

Table 2. Effective performance

rpm	**1600**	**1400**	**1200**
Value	Heavy duty		
rpm	1600	1400	1200
kW	190	164	122
g/kW-h	209	212	217
l/h	48.0	42	32

In Fig. 2, Fig. 3, Fig. 4, Fig. 5, Fig. 6 and Fig. 7 are shown results achieved by simulation of processes turbocharger engine according to model describe above.

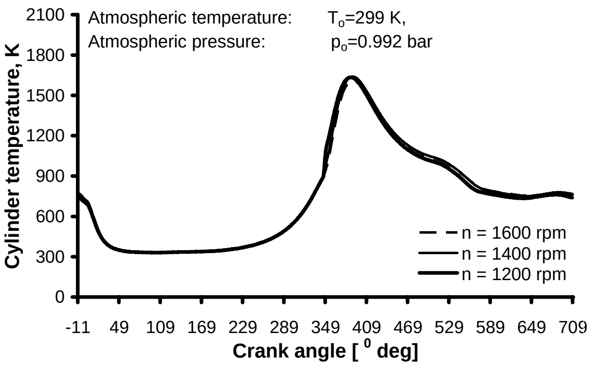

Figure 2. Cylinder pressure predicted by the simulation as a function of crank angle

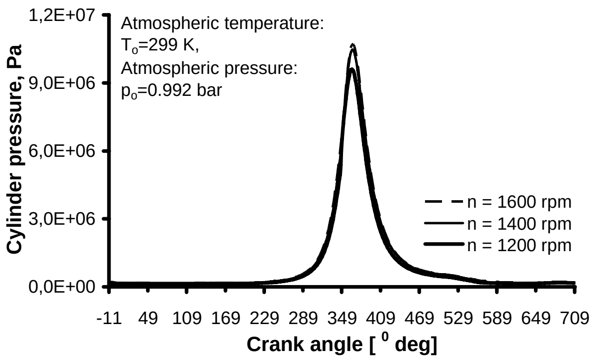

Figure 3. Cylinder temperature predicted by the simulation as a function of crank angle

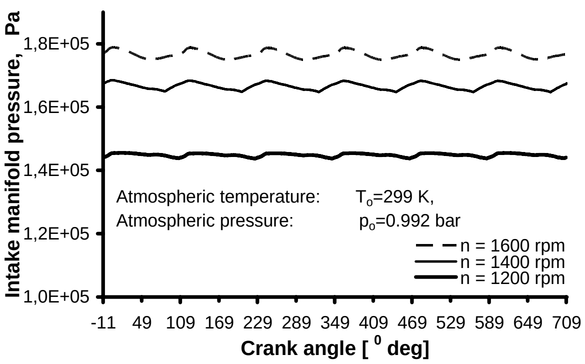

Figure 4. Intake manifold pressure predicted by the simulation as a function of crank angle

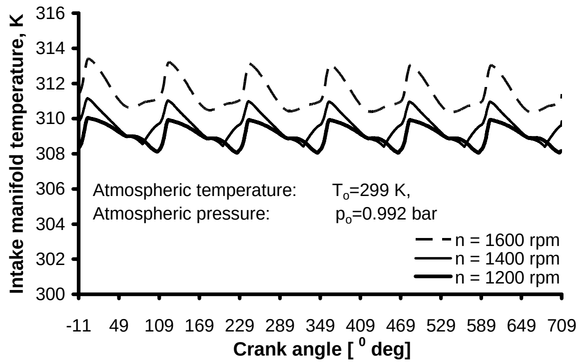

Figure 5. Intake manifold temperature predicted by the simulation as a function of crank angle

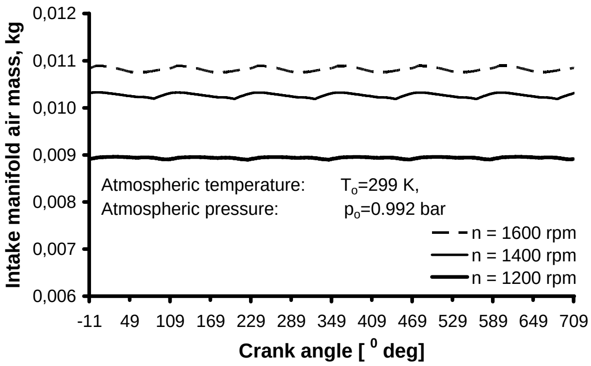

Figure 6. Intake manifold air mass predicted by the simulation as a function of crank angle

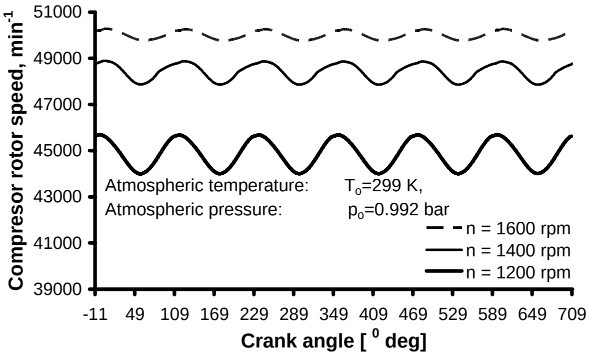

Figure 7. Compressor rotor speed predicted by the simulation as a function of crank angle

Results achieved by simulation of processes turbocharger engine according complex model and programs in Matlab software [3], is real for observed engine.

4. CONCLUSIONS

The curves in the graphs for pressure, temperature and air mass in intake manifold have the same behavior, i.e., a series of six identical repeating pulses, each produced by filling process of the individual cylinders of six cylinder engines. The curve in the graphs for rotor speed of turbokompresor has a same behavior, each produced by empting process of the individual cylinders of six cylinder engine. Compressor efficient is 83 % at 1600 rpm, respectively 81,74 % at 1200 rpm. Results achieved from mathematical model are based in fixed value of injection angle for full range of number of rotations, but in reality, injection angle depends on rotation numbers of engine shaft.
According the achieved results by simulation of processes turbocharger engine may notice that presented model for chosen turbocharger is a good base for system in *downsizing* concept.

5. REFERENCES

[1] Heywood, J.B., Internal Combustion Engine Fundamentals, McGraw-Hill Book Company, 1988,
[2] Filipović I.,Nadpunjenje klipnih motora sui. Masinski fakultet Sarajevo, Sarajevo 1998.
[3] Lajqi N., Modelimi i proceseve te motorët dizel shumë cilindrik me fryrje me përdorimin e paketëve bashkohore softuerike, punim i doktoraturës, Prishtinë 2006.,
[4] Dobovišek Ž., Černej A., Mirković M., Filipović I., "Simuliranje Procesa Klipnih Motora SUS", Sarajevo, 1983,
[5] Jankov Rade., Matematičko Modeliranje Strujno-Termodinamičkih Procesa i Pogonski Karakteristika Disel-Motora, Beograd 1984.