International Journal of Advances in Engineering and Management (IJAEM)
Volume 5, Issue 7 July 2023, pp: 733-738 www.ijaem.net ISSN: 2395-5252

Modelling an Electrically Turbocharged Engine by Using Commercial Software

Hoang Phuc Trinh
Department of Automotive Engineering Technology, EAST ASIA University of Technology

Date of Submission: 20-07-2023 Date of Acceptance: 31-07-2023

ABSTRACT
Increasing power and improving exhaust emissions are challenges for the development of new technologies for internal combustion engine. Supercharger of internal combustion engine can be considered one of the most effective methods to solve the above problem, especially for diesel engine. Currently, most diesel engines in the world are equipped with turbocharger system. This study aims to build a full model of the systems of gasoline engines using electric turbochargers, calculate and give the power and emission characteristics of the engine. From there evaluate the ability to increase pressure and optimize the technical parameters.The results showed that after supercharging the engine power was increased dramatically. These results will be valuable bases to calculate and design the turbocharger system of the gasoline engine in Vietnam.

Keywords: Gasoline engine;Turbocharger; Supercharger; Temperature distribution; Engine efficiency.

I. INTRODUCE

Currently, emissions and environmental pollution are serious problems for Vietnam and around the world. Scientists around the world have also come up with many solutions to solve this problem, such as using alternative fuels, finding new energy sources such as fuel cells or the sun. However, the internal combustion engine is still an important source of driving force and is unlikely to be replaced soon due to the advantages of the internal combustion engine in terms of cost and specific power [1-5].

Currently, a large number of diesel engines are equipped with turbochargers.However, turbocharging for gasoline engines is still not popular due to limitations in durability as well as the possibility of detonation.Many commercial vehicles using gasoline engines have been equipped with turbochargers and have improved many indicators in terms of economy, capacity and emissions.In 2015, Lexus officially introduced its first turbocharged gasoline engine with the European version of the NX 200t luxury SUV.The all-new Lexus 2.0L turbocharged I4 petrol engine has a capacity of up to 238 horsepower at the range of 4,800 - 5,600 rpm.Maximum torque reaches 350Nm between 1,650 - 4,000 rpm.The NX 200t version equipped with this engine is capable of accelerating from 0-100km / h in 7.1 seconds and reaching a maximum speed of 200km / h, CO2 emission coefficient is only about 178 g / km.The Honda Accord model uses a turbocharged 4-cylinder gasoline engine with a capacity of 192 horsepower and a CVT continuously variable transmission.The model has a fuel consumption of about 7.13 liters / 100 km on mixed roads, 7.85 liters / 100 km on urban roads and 6.19 liters / 100 km on highways.Chang Sik Lee and colleagues studied mechanical turbochargers for gasoline engines in a study published in the KSME International Journal.Experimental results have also shown that the power and combustion characteristics are significantly improved compared to the pre-turbocharged engine [7].The characteristics of gasoline engines using exhaust gas turbines were also studied by T. Sadoi and colleagues published in the Journal of Engineering for Gas Turbines and Power [8].This study has calculated the theory, then selected the turbine and installed it on the actual engine to evaluate the efficiency.The results also show that the engine after turbocharging has significantly improved working characteristics.In a similar study, Qingning Zhang and colleagues proposed turbocharging that uses electrical energy to supercharge an internal combustion engine.This paper simulates a turbine using electric energy for a gasoline engine with a capacity of 2.0.Simulation results show that the turbocharger reduces CO2 emissions and fuel consumption by nearly 1% in different drive cycles compared to the exhaust turbocharger [9]. Common types of turbochargers for commercial vehicles today include turbochargers using exhaust turbines (Turbocharger) and

International Journal of Advances in Engineering and Management (IJAEM)
Volume 5, Issue 7 July 2023, pp: 733-738 www.ijaem.net ISSN: 2395-5252

mechanical turbochargers (Supercharger).In particular, the turbocharger system uses energy from the exhaust system to compress the air as the basis for increasing the vehicle's capacity.In contrast, a mechanical turbocharger system uses the energy taken from the crankshaft to compress the air flow.As a result, the mechanical turbocharger system can eliminate the disadvantage of exhaust gas energy flow delay and reduce the working temperature for the turbine-compressor assembly.However, this type of turbocharger uses energy from the crankshaft, so it takes mechanical work.Therefore, this system is only installed on vehicles with large cylinder capacity to ensure the best performance.In addition, the efficiency is less than the turbocharger system because the car has to consume more fuel to pull both engines to operate at the same time.In addition, by putting extra pressure on the engine, details such as pistons, armrests, and valves must operate at higher speeds and pressures than normal, so the life of these important parts will certainly be reduced.over a long period of use.Turbochargers using electric turbines will partially solve the disadvantages of exhaust gas turbine turbochargers and mechanical turbochargers, since electricity is used as the power source, this system is completely independent of exhaust gas or power from the exhaust gas.engine to operate.Besides, electric turbocharger also solves the problem of engine temperature, so this solution helps to make the engine compartment space more spacious and the problem of heat dissipation will be better. The topic "Study on characteristics of gasoline engines using electric turbochargers" aims to fully model the systems of gasoline engines using electric turbochargers, calculate and give output characteristics as well as develop power generation.engine exhaust.From there evaluate the ability to increase pressure, and optimize the technical parameters as the basis for conducting experiments.

II. SIMULATION D243 ENGINE BY AVL_BOOST SOFTWARE

2.1. Theoretical basic

Theoretical background including the basic equations for all available elements is summarized in this paper to give a better understanding of the AVL BOOST program

2.1.1. Basic conservation equations

The calculation of the thermodynamic state of the cylinder is based on the first law of thermodynamics. The first law of thermodynamics for high pressure cycle states that the change of the internal energy in the cylinder is equal to the sum of piston work, fuel heat input, wall heat losses and the enthalpy flow due to blow-by, equation 1 [2]

$$\frac{d(m_c.u)}{d\alpha} = -p_c.\frac{dV}{d\alpha} + \frac{dQ_F}{d\alpha} - \sum\frac{dQ_W}{d\alpha} - h_{BB}.\frac{dm_{BB}}{d\alpha} \quad (1)$$

Where m_c - mass in the cylinder, u - specific internal energy, p_c cylinder pressure, V - cylinder volume, Q_F - fuel energy, Q_W – wall heat loss, α - crank angle, h_{BB} - enthalpy of blow-by, m_{BB} - blow-by mass flow

2.1.2. Combustion model

BOOST_SOFTWARE uses the Mixing Controlled Combustion (MCC) model for the predictionof the combustion characteristics in direct injection compression ignition engines.The heat release is a function of the fuel quantity available (f_1) and the turbulent kinetic energy density (f_2), equation 2

$$\frac{dQ}{d\phi} = C_{Comb}\cdot f_1(M_F,Q)\cdot f_2(k,V) \quad (2)$$

Where $f_1(M_F,Q) = M_F - \frac{Q}{LVC}$,

$f_2(k,V) = \exp(C_{rate}\cdot\frac{\sqrt{k}}{\sqrt[3]{V}})$, C_{Comb} - combustion constant (kJ/kg.deg CA), C_{rate}- mixing rate constant (s), k - local density of turbulent kinetic energy (m^2/s^2), M_F- vapourized fuel mass (kg), LCV - lower heating value (kJ/kg), Q - cumulative heat release for the mixture controlled combustion (kJ), V - cylinder volume (m^3), α - crank angle (deg CA).

2.1.3. Heat transfer model

The heat transfer to the walls of the combustion chamber, i.e. the cylinder head, the piston, and the cylinder liner, is calculated from equation (3) [3]

$$Q_{wi} = A_i\cdot\alpha_i\cdot(T_c - T_{wi}) \quad (3)$$

Where Q_{wi} - wall heat flow, A_i - surface area, α_i- heat transfer coefficient, T_c gas temperature in the cylinder,T_{wi} - wall temperature.

Heat transfer coefficient (α_i) is usually calculated by WOSCHNI Model, The Woschni model published in 1978 for the high pressure cycle is summarized as follows: [4]

$$\alpha_w = 130.D^{-0,2}.p_c^{0,8}.T_c^{-0,53}.[C_1.c_m + C_2.\frac{v_D.T_{c1}}{p_{c,1}.V_{c,1}}.(p_c - p_{c,0})]^{0,8}$$

(4)

Where C_1 = 2,28 + 0,308.c_u/c_m, C_2 = 0,00324 for DI engines, D - cylinder bore, c_m -mean piston speed, c_u - circumferential velocity,c_u= π.D.nd/60, V_D- displacement per cylinder, $p_{c,o}$ - cylinder pressure of the motored engine (bar), $T_{c,1}$ - temperature in the cylinder at intake valve closing (IVC) , $p_{c,1}$ - pressure in the cylinder at IVC (bar).

2.1.4. Turbocharger simulation

a. Turbine

For the simulation of a turbine, the performance characteristics along a line of constant turbine are required. The power provided by the turbine is determined by the turbine mass flow rate and the enthalpy difference over the turbine.

$$P_T = \dot{m}.\eta_m.(h_3 - h_4) \quad (5)$$

Where P_T - turbin power, $\dot{m}$ - turbin mass flow, η_m - mechanical efficiency of the turbocharger, h_3 - enthalpy at the turbine inlet, h_4 - enthalpy at the turbine outlet.

$$h_3 - h_4 = \eta_{s,T}.c_p.T_3.\left[1 - \left(\frac{p_4}{p_3}\right)^{\frac{K-1}{K}}\right] \quad (6)$$

Where $\eta_{s,T}$- isentropic turbine efficiency, c_p - mean specific heat at constant pressure between turbine inlet and outlet, T_3 - turbine inlet temperature, p_4/p_3- turbine expansion ratio, η_{tot}- total efficiency of turbine.

b. Compressor

The power consumption of the turbo compressor depends on the mass flow rates in the compressor and the enthalpy difference over the compressor. The latter is influenced by the pressure ratio, the inlet air temperature, and the isentropic efficiency of the compressor.

$$P_c = \dot{m}.(h_2 - h_1) \quad (7)$$

Where P_c-compressor power consumption, $\dot{m}_c$ - mass flow rate in the compressor, h_2 -enthalpy at the outlet of the compressor, h_1 - enthalpy at the inlet to the compressor.

$$h_2 - h_1 = \frac{1}{\eta_{s,c}}.c_p.T_1.\left[\left(\frac{p_2}{p_1}\right)^{\frac{K-1}{K}} - 1\right] \quad (8)$$

$\eta_{s,c}$

Where $\eta_{s,c}$- isentropic efficiency of the compressor, c_p - mean value of the specific heat at constant pressure between compressor inlet and outlet, T_1- compressor inlet temperature, p_2/p_1 - compressor pressure ratio.

c. Turbocharger

For steady state engine operation the performance of the turbocharger is determined by the energy balance or the first law of thermodynamics. The mean power consumption of the compressor must be equal to the mean power provided by the turbine:

$$P_c = P_T \quad (9)$$

The overall turbocharger efficiency (η_{TC}) is defined as follows

$$\eta_{TC} = \eta_{m,TC}.\eta_{s,T}.\eta_{s,c} \quad (10)$$

In BOOST software, three calculation modes for the supercharged engine are available:

1. In the turbine layout calculation, the desired pressure ratio at the turbo compressor is specified as input to the calculation. The program adjusts the flow resistance of the turbine automatically, until the energy balance over the turbocharger is satisfied.
2. For the boost pressure calculation, the actual turbine size is specified in the input. By solving the energy balance over the turbocharger, the actual boost pressure is calculated.
3. For the waste gate calculation, both the turbine size as well as the desired pressure ratio at the compressor are specified in the input. The program bypasses a certain percentage of the exhaust gases in order to achieve the energy balance over the turbocharger. If the desired compressor pressure ratio cannot be achieved with the specified turbine size, the program switches over to the boost pressure calculation mode.

2.2. Modeling engine

2.2.1. The characteristic of engine

This engine is four-stroke diesel engine, max output 80 hp/2200 rpm. In Vietnam, these models engine are popular. The characteristic of engine is shown in Table 1

Table 1. The characteristic of engine

No	Parameter	Value	Unit
1	Firing order	1-3-4-2	-
2	Displacementvolume (V_h)	4,75	dm^3
3	Bore/Stroke (D/S)	110/125	mm/mm
4	Pressure ratio (ε)	16,4	-
5	Power (N_{e-dm})	80	HP
6	Maximum torque (M_{e-max})	280	N.m
7	Timing injection (φ_s)	25÷27	BTDC

International Journal of Advances in Engineering and Management (IJAEM)
Volume 5, Issue 7 July 2023, pp: 733-738 www.ijaem.net ISSN: 2395-5252

The parameter which is imported into the model is determined from some measurements in laboratory of internal combustion engine. The result about the power and fuel consumption is shown in Table 2. This result is base to estimate the exactitude of the model.

Table 2. The parameter is measured in Lab.

n (rpm)	N_e (kW)	G_{fuel} (g/cycle)
1000	31.38	0.066
1400	45.87	0.074
1600	51.89	0.073
1800	57.03	0.070
2000	56.18	0.061
2200	56.09	0.055

2.2.2.Engine model in AVL_BOOST software

The model is built base on the structure of the existing diesel engine and the relative document. Table 3 and 4 show some elements and parameters of model.

Table 3. Some elements of model

No	Element Name	Symbol
1	Intake, Exhaust pipe	-
2	Boundary elements	SB
3	Plenum	PL
4	Cylinder	C
5	Restriction	R
6	Measuring point	MP
7	Air cleaner	CL
8	Turbocharger	TC
9	Wastegate	WG
10	Aircooler	CL
11	Aircleaner	CO

Table 4. The main parameters ofmodel

No	Parameter	Value
1	RPM	1200÷2200
2	Air pressure (bar)	1
3	Air temperature (^{0}C)	25
4	Number cycle	50
5	Fuel per cycle (g/cycle)	0,055÷0,074
7	Low heat value (kJ/kg)	42800
8	Ratio A/F	14,7
9	Combustion Model	AVL MCC
10	Type engine	4 kỳ
11	Firing order	1-3-4-2

The engine models in AVL_BOOST is shown in Figure 1 and 2:

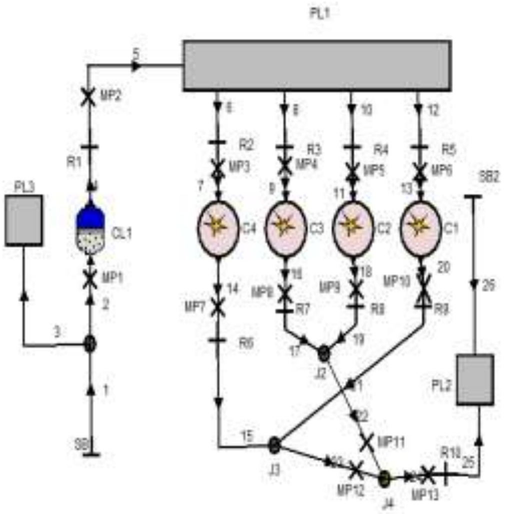

Fig.1 Model of non-turbocharged engine

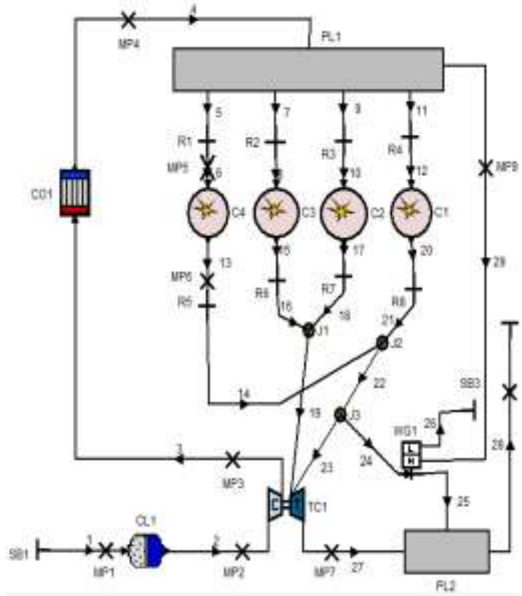

Fig.2 Model of turbocharged engine

III. SIMULATION RESULTS AND DISCUSSION

3.1. Validation of model

Figure 2 shows the results of simulation and experiment. They are quite similar, The maximum difference about the fuel consumption is 15.2% in n = 1600 rpm, about the power is 5.2% in n = 2200 rpm.But the difference isn't too much, the resultsare enough reliability to serve for the continuous researching.

International Journal of Advances in Engineering and Management (IJAEM)
Volume 5, Issue 7 July 2023, pp: 733-738 www.ijaem.net ISSN: 2395-5252

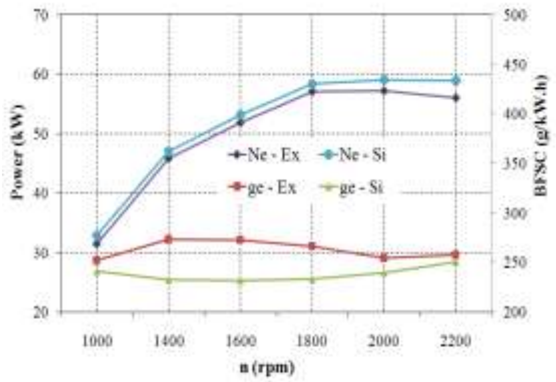

Figure3. Engine performance of simulation and experiment

3.2. The ability supercharging of engine

3.2.1. Engine performance

The engine performance of non-turbocharged engine and turbocharged engine are shown in Figure 4

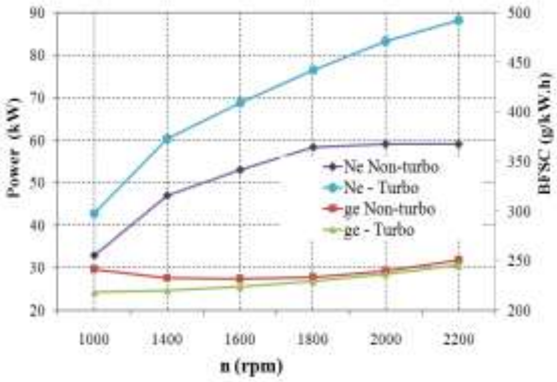

Fig.4The characteristic of power and fuel consumption

The result shows that the power of turbocharged engine increases and fuel consumption significantly reduces. At the n = 1000 rpm, the power increases 30.4% and fuel consumption decreases 9.4%. At the n = 1600 rpm, the power increases 29.5% and fuel consumption decreases 3.5%. At the n = 2200 rpm the power increases 49.5% and fuel consumption reduces 2.1%.

3.2.2. Exhaust emission

The exhaust emission of non-turbocharged engine and turbocharged engine are shown in Table 4,5 and 6.

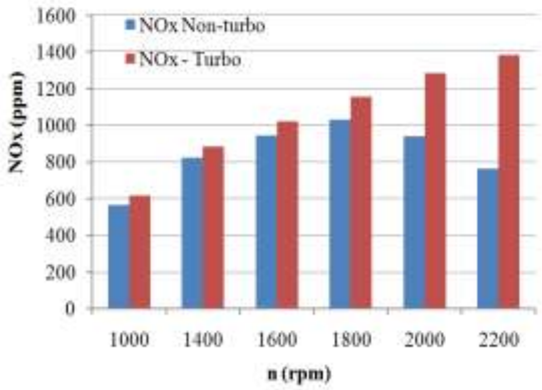

Fig 5. NO_Xemission of non-turbocharged engine and turbocharged engine

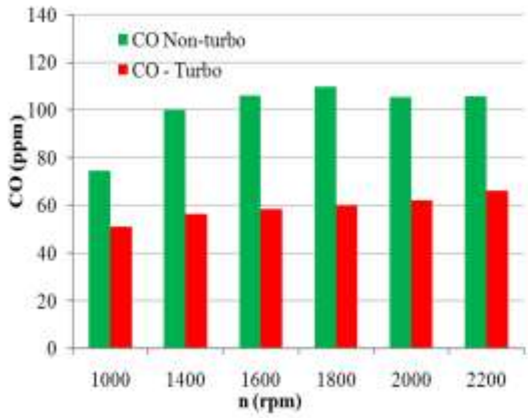

Fig 6. CO emission of non-turbocharged engine and turbocharged engine

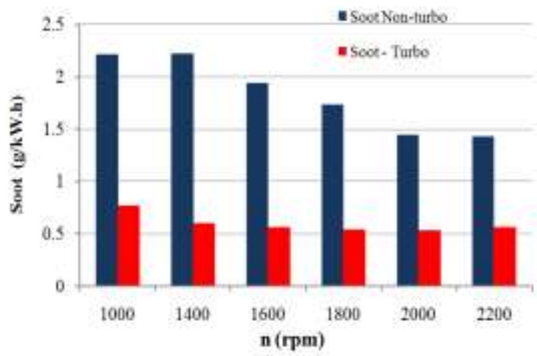

Fig 7. Soot emission of non-turbochaged engine and turbocharged engine

The result shows that with turbocharged engine, the exhaust emission significantly changes. In Figure 4, NO_X emission is higher than non-turbocharged engine. The maximum difference is 80.8% in n = 2000 rpm, this is disadvantage of turbocharged engine. Therefore, to satisfy the emission NO_X standard must use some methods to reduce NO_X.

In Figure 5, it shows that CO emission of turbocharged engine is lower. At the n = 2200 rpm, CO concentration lower 37.6% and in n = 1000 rpm, it is 31.7%.

In Figure 6, we can see that Soot emission reduces so much. At the n = 2200 rpm, it reduces 60.8% and 65.2% at the n = 1000 rpm.

IV. CONCLUSION

With the engine to be Retrofitted by turbocharger, the performance of enginesignificantly improve. CO and Soot emissions reduce but NO_X and the temperature combustion increase. It is suitable with experiment and many other researching.

The report of this simulation result is useful for design, prototype and test of engine in the future.

REFERENCE

[1]. The Report of simulation of turbocharged, six cylinders, diesel engine. VEAM Corporation, 2005.
[2]. AVL: Thermodynamic cycle simulation Boost, Primary, Version 3.2 1998.
[3]. AVL: Thermodynamic cycle simulation Boost, Boost user's guide, Version 3.2.
[4]. Khong Vu Quang "Simulation of the thermodynamic cycle and mass flow in scavenge process of ICE using AVL Boost". Master of science Thesis, University of Technology, Hanoi, 2002.
[5]. Pham Huu Tuyen "Research and Application of AVL Tycon to simulate valve and cam system of ICE". Master of science Thesis, University of Technology, Hanoi, 2002.