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Supercharging Study for Diesel Engine by Using AVL_Boost Software

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Abstract

Increasing power and improving exhaust emissions are challenges for the development of new technologies for internal combustion engine. Turbocharger 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. However, in Vietnam the non-turbocharged diesel engine is very popular. Therefore, supercharging study for diesel engine is necessary. This paper presents simulation study on the thermodynamic process of four cylinders diesel engine without and with turbocharger system by using AVL_Boost software. Since then analyze and evaluate the possibility of turbocharger for diesel engines are circulating in Vietnam. The results showed that after turbocharged the engine power was increased while the CO, HC and PM emissions ware reduced. However, NO_X emission and the stress effects on details ware significantly increased. These results will be valuable bases to calculate and design the turbocharger system of the diesel engine in Vietnam.

Keywords

Diesel engine with turbocharger and intercooler, Simulation, Turbocharger system.

1. INTRODUCE

The Boost of internal combustion engine using turbocharger is an efficient method of boosting power and reducing exhaust emission, fuel consumption. However, In Viet Nam, Almost diesel engines are not retrofitted turbocharger systems [1].

In this research, base on the existing diesel engine with displacement volume is 4 liter, non-turbocharged, maximum power output is 80 hp/2300 rpm, maximum torque is 280 Nm/1200 rpm, we modeled it by AVL_Boost software and compared with the characteristic of real engine in test bed to estimate the reliability of the model. This model is base to build the turbocharged diesel engine to serve for some other researching.

2. SIMULATION OF DIESEL 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]

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

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

AVL_Boost software uses the Mixing Controlled Combustion (MCC) model for the prediction of 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

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

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]

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

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]

$$\alpha_W = 130.D^{-0,2}.p_c^{0,8}.T_c^{-0,53}.[C_1.c_m + C_2.\frac{V_D.T_{c1}}{p_{c,1}.V_{c,1}}.(p_c - p_{c,0})]^{0,8} \quad (4)$$

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, $c_u = \pi.D.nd/60$, 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.

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

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 AVL_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. Model of 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	Displacement volume (V_h)	4,75	dm^3
3	Bore/Stroke (D/S)	110/125	mm/mm
4	Pressure ratio (ε)	16,4	-
5	Max power (N_{e-max})	80	HP
6	Max torque (M_{e-max})	280	N.m
7	Timing injection (φ_s)	25÷27	Deg CA

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. Tables 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 of model

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 software is shown in Figures 1 and 2.

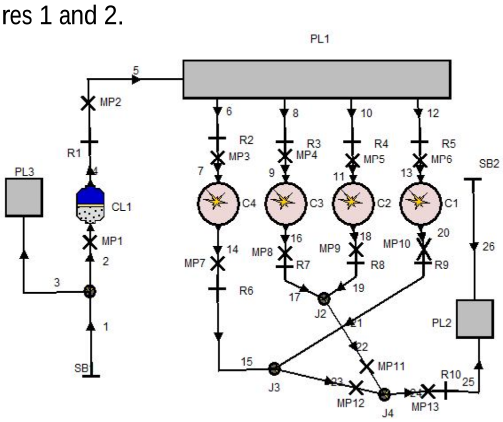

Fig.1 Model of non-turbocharged engine

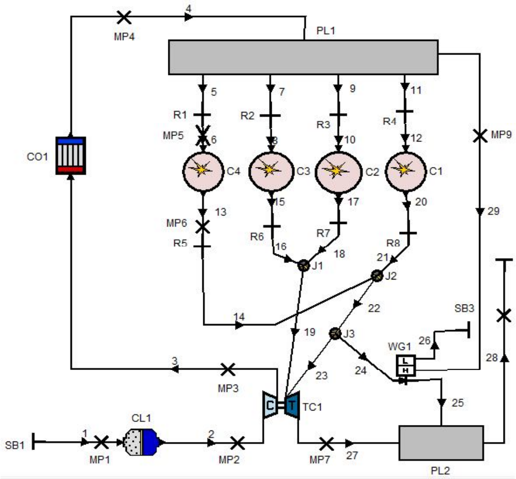

Fig.2 Model of turbocharged engine

3. SIMULATION RESULTS AND DISCUSSION

3.1. Validation of model

Figure 3 shows the results of simulation and experiment. They are quite similar, the maximum difference about the fuel consumption is 15.2% at the n = 1600 rpm, about the power is 5.2% at the n = 2200 rpm. But the difference isn't too much, the results are enough reliability to serve for the continuous researching.

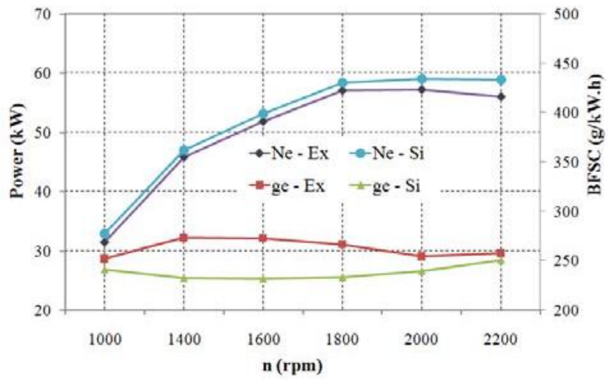

Figure 3. Engine performance of simulation and experiment

3.2. Turbocharger of engine

3.2.1. Engine performance

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

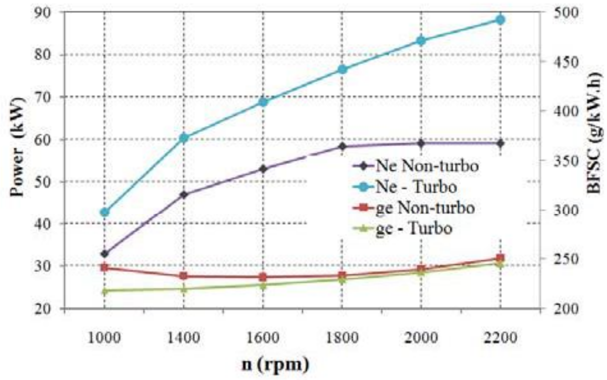

Fig.4 The characteristic of power and fuel consumption

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 Figures 5, 6 and 7.

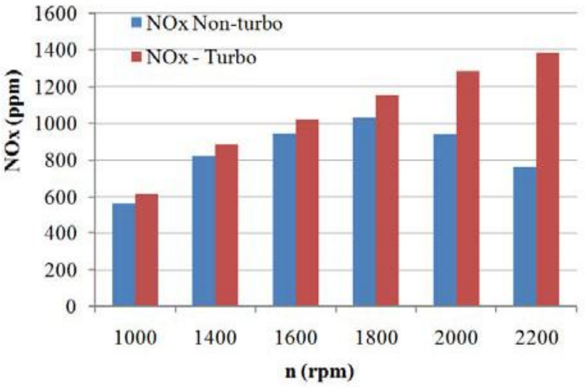

Fig 5. NO_X emission of non -turbocharged engine and turbocharged engine

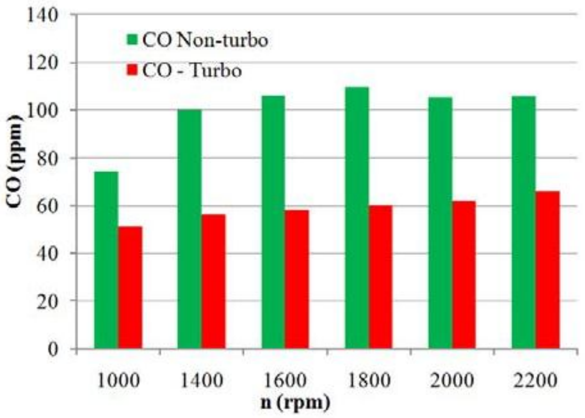

Fig 6. CO emission of non-turbocharged engine and turbocharged engine

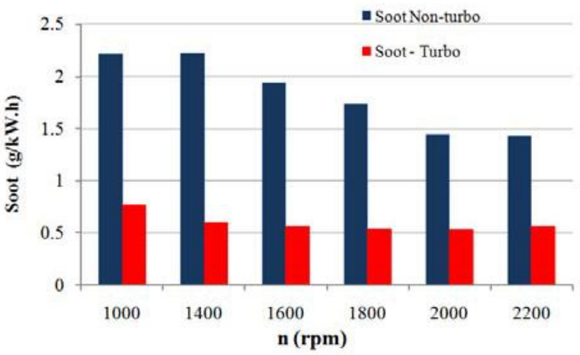

Fig 7. Soot emission of non-turbochaged engine and turbocharged engine

The result shows that with turbocharged engine, the exhaust emission significantly changes. In Figure 5, NO_X emission is higher than non-turbocharged engine. The maximum difference is 80.8% at the 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 6, it shows that CO emission of turbocharged engine is lower. At the n = 2200 rpm, CO concentration lower 37.6% and at the n = 1000 rpm, it is 31.7%.

In Figure 7, 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.

4. CONCLUSION

With the engine to be Retrofitted by turbocharger, the performance of engine significantly 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

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