



Simulation of the effect of using a turbocharger on XU7 engine performance with mechanical and electronic wastegate control mechanisms

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ABSTRACT

Given the importance of improving the efficiency of internal combustion engines, reducing fuel consumption, and mitigating emissions, boosting technology particularly the turbocharger has been introduced as an effective solution. Turbocharging enables the realization of the engine downsizing concept. In this study, the XU7JP4/L3 engine was first simulated in GT-Power software, and subsequently, a boosting circuit was integrated into the engine model. The performance of the boosting system was evaluated under two wastegate control strategies. In the first case, an internal wastegate with a mechanical control mechanism was employed, while in the second case, an electronic control mechanism was utilized. The results revealed that brake power under boosted conditions with mechanical wastegate control increased by an average of 33%, and with electronic wastegate control by 37%, compared to the naturally aspirated mode. The brake torque in the boosted mode with mechanical wastegate control also increased by 25%, 37%, and 41% at 1500 rpm (simulation start), 2500 rpm (maximum torque), and 6000 rpm (maximum power), respectively, compared to the naturally aspirated mode. The implementation of electronic wastegate control shifted the maximum torque to 3500 rpm and further increased it by 5% compared to the maximum torque achieved under mechanical control. Furthermore, the brake-specific fuel consumption (BSFC) decreased by an average of 2% at low-to-medium speeds and by 3% at medium-to-high speeds in both boosted modes compared to the naturally aspirated mode. In contrast, the instantaneous fuel consumption increased by an average of 32% across the entire engine operating range. Overall, the results of this study confirm the superiority of the electronic wastegate control mechanism in enhancing the impact of the boosting system on the performance of the XU7 engine.



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1- Introduction

In line with the expansion of global policies aimed at reducing emissions, automotive manufacturers have been compelled to develop innovative solutions to decrease the level of pollutant emissions [1]. In urban areas, vehicles are regarded as one of the primary sources of air pollution [2]. An ideal engine is defined as one that combines low emissions, high fuel efficiency, and adequate specific power output [3]. Accordingly, the application of advanced technologies such as turbocharging, variable valve timing (VVT) systems, and exhaust gas recirculation (EGR) has been widely developed and implemented in recent years [4]. Among these technologies, the use of turbochargers is recognized as one of the most effective methods for recovering exhaust gas energy [5], improving engine efficiency and performance [6-7], and reducing emissions [8] in internal combustion engines.

The use of computer-aided simulation software enables the accurate prediction of potential engine modifications, thereby significantly reducing the costs associated with experimental testing [9]. A study conducted by Mousavi *et al.* [10] on the matching of turbochargers with internal combustion engines demonstrated that the simulation results exhibited good agreement with experimental data, showing less than a 1.5% error in engine modeling and a maximum of 16% deviation in overall simulation results. This indicates that such simulation tools can be reliably utilized for engine analysis. Mahmoudi *et al.* [11], after modeling a 1994 Nissan Maxima engine using GT-Power software, compared their results with the manufacturer's reported data and observed a strong correlation between their simulation results and the manufacturer's data. Subsequently, they added the selected turbocharger to the model and analyzed its effects on the baseline engine. The investigation revealed that the maximum brake power and peak torque in the turbocharged condition increased by 95% and 97%, respectively, compared to the naturally aspirated engine. However, the emission levels showed a considerable increase, which was attributed to higher fuel consumption and elevated combustion temperatures. Overall, the results indicate that the application of optimized and compact turbochargers in naturally aspirated engines makes it possible to achieve comparable power output and efficiency levels with lower emissions and improved compliance with environmental standards. Silva *et al.* [12] simulated the turbocharging of a 1.6 L Fiat X500 engine using GT-Power software. In this study, the simulation results were evaluated for the engine with the original displacement under both naturally aspirated and turbocharged conditions. After the addition of the turbocharger, the maximum brake power and maximum torque increased by 76% and 70%, respectively. Moreover, with the use of the turbocharger, the peak torque became available over a wider range of engine speeds. The brake-specific fuel consumption (BSFC) showed a slight reduction at low to medium engine speeds, followed by a considerable increase at higher speeds. Kim *et al.* [13] investigated the effect of turbocharging on a small gasoline engine and reported that the engine torque increased by 12.78%, while NO_x emissions were reduced by approximately 54.31%. According to the findings of Ebel *et al.* [14], the selection of a suitable turbocharger for an internal combustion engine does not depend solely on the inherent efficiency of the compressor and turbine; rather, its full compatibility with the engine characteristics plays a crucial role. The results also indicated that the addition of a turbocharger can lead to a significant increase in engine torque, while the BSFC rises by only about 1.5%. Al-Rawashdeh *et al.* [15] investigated the effect of employing a variable-pressure-ratio turbocharger on a six-cylinder inline engine. The results showed that this system could enhance the output power, brake mean effective pressure (BMEP), and engine torque from 280 kW, 11.7 bar, and 1779 N.m to 1042 kW, 43.7 bar, and 6638 N.m, respectively. Alexander *et al.* [16] examined the performance improvement in a two-cylinder spark-ignition engine by implementing turbocharging along with a reduced compression ratio. The results revealed that these modifications increased brake power by 19.3% and reduced fuel consumption by 10.1%. Jain *et al.* [17] studied the influence of exhaust gas flow on turbocharger performance in three-cylinder and four-cylinder engines

with identical displacements. Their findings showed that using a smaller turbine in the turbocharger reduced turbo-lag and improved volumetric efficiency and engine power by up to 5%. Hamor et al. [18] investigated turbocharger matching with a three-cylinder engine. The results demonstrated that with turbocharging, the thermal efficiency at full-load increased by 5.41%, and the BSFC decreased by 11 g/kWh, indicating a notable enhancement in engine performance and efficiency. In another study, Hamor et al. [19] selected an optimized turbocharger and matched it with the engine to improve efficiency and reduce emissions. The results showed that BSFC improved by 6.65%, while NO_x, HC, and CO emissions decreased by 1.73%, 6.23%, and 2.61%, respectively.

Recent studies have highlighted the importance of wastegate control strategies in optimizing turbocharged engine performance. Shoman et al. [20] demonstrated that wastegate settings significantly influence key parameters such as BSFC, power, and torque. Their steady-state simulation results indicate that proper wastegate adjustment substantially enhances engine performance, increasing brake power and volumetric efficiency while reducing both BSFC and equivalent brake-specific fuel consumption (EBSFC). Notably, EBSFC exhibits nuanced variations depending on wastegate configuration and engine speed; for instance, at a turbocharger speed of 140,000 rpm, EBSFC decreases by 2.8% with a 40% wastegate opening at 10,000 rpm engine speed, while a 2.5% reduction is observed with a 20% opening at 12,000 rpm.

Conventionally, the wastegate valve is controlled by a pneumatic actuator, which opens the valve according to a predefined and constant maximum boost pressure downstream of the compressor across the entire engine operating range. Consequently, the boost pressure at high engine speeds—where knocking is less critical and volumetric efficiency is lower—is limited by the same threshold value defined for medium-low speeds. In other words, the pneumatic wastegate inherently restricts the maximum achievable engine power. In contrast, Romani et al. [21] implemented an electronic control system for the wastegate valve with a dedicated control strategy aimed at providing the desired boost pressure at full load for each engine speed, in order to achieve maximum power without knocking. The results demonstrated that the electronic wastegate provided higher power output and a more extended torque curve compared to the conventional pneumatic system.

These findings underscore the potential of electronic wastegate control to enhance engine performance, yet its application on production engines such as the XU7 series remains underexplored.

In the present study, in addition to investigating the effect of the boosting system under the default configuration (a mechanically controlled wastegate) on the engine performance, the electronic wastegate control mechanism was also analyzed. The implementation of this electronic mechanism enables more precise control of boost pressure, thereby optimizing engine performance. A review of previous studies indicates that the electronic control capability of the wastegate system in boosting configurations has received limited attention and detailed analysis. Therefore, this paper aims to fill this research gap through comparative simulation and analysis of these two control mechanisms.

2- Engine Simulation Method and Equations

In this study, GT-Power software, part of the GT-Suite, was used for the simulations. This software is recognized as a standard tool for engine performance simulation and is employed by leading engine manufacturers to predict parameters such as power, torque, fuel consumption, and emissions, as well as to evaluate turbocharger matching and performance. GT-Power is capable of analyzing and optimizing various internal combustion engines, including gasoline engines, and its numerical calculations are based on one-dimensional solutions of fluid dynamics equations, covering phenomena such as gas flow, combustion, and heat transfer in different engine components. In this research, the four-cylinder XU7JP4/L3 engine was used for simulation. The specifications are presented in Table 1 [22]:

Table 1 XU7JP4/L3 engine specifications

Number	Characteristics	Value (Unit)
1	Bore	83 (mm)
2	Piston Stroke	81.4 (mm)
3	Engine Displacement	1761 (cc)
4	Number of Valves	8
5	Compression Ratio	9.3:1
6	Firing Order	1-3-4-2
7	Maximum Power	91.19 (hp) @ 6000 (rpm)
8	Maximum Torque	144 (N.m) @ 2500 (rpm)

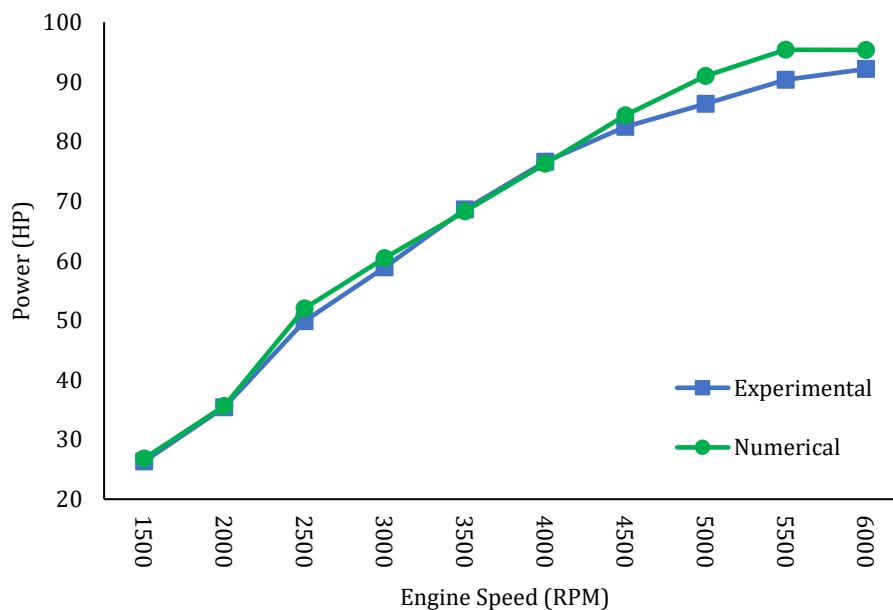
In the first step, all engine components, including intake and exhaust paths, injectors, valves, and cylinder geometric specifications, were modeled in the GT-Power environment.

For simulating the combustion process in this study, the EngCylCombProfile model was used. This sub-model is a non-predictive model in which the combustion rate is determined based on crank angle and the amount of energy released from combustion. Additionally, the WoschniGT model was employed to simulate the heat transfer rate in GT-Power. This model is a classical Woschni model developed by Gamma Technologies and is used in cases where detailed information about fluid motion is not available. The Chen-Flynn model was utilized to simulate engine friction.

After simulating the required engine parameters and components in the GT-Power software, the results were compared with existing experimental data to ensure the accuracy of the simulation.

2-1- Validation

Given the inherent uncertainties associated with numerical methods, validation of the simulated model through comparison with experimental data is essential. Accordingly, the outputs of the simulation model for the main engine performance indicators, including power, torque, and brake specific fuel consumption (BSFC), were compared with reliable experimental data. This evaluation was conducted over an engine speed range of 1500 to 6000 rpm. Figures 1 to 3 respectively illustrate the agreement between the simulated and experimental values of power, torque, and BSFC.

**Figure 1** Power chart according to engine speed

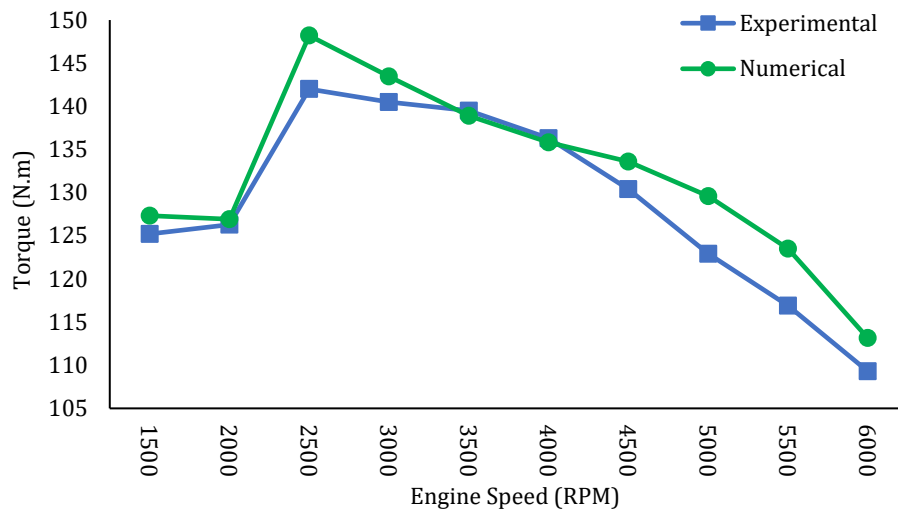


Figure 2 Torque chart according to engine speed

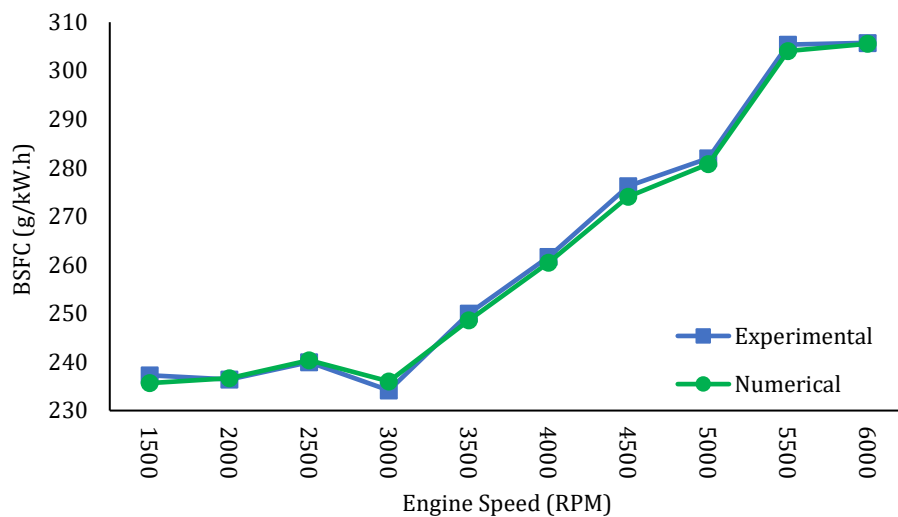


Figure 3 BSFC chart according to engine speed

As observed in Figures 1 to 3, the simulation results for power and BSFC show excellent agreement with the experimental data. Although the simulated brake torque is slightly higher than the experimental values, the overall trend of its variation is accurately captured.

Overall, the comparison of key engine performance parameters, including brake power, brake torque, and BSFC, between the simulated model and experimental data indicates the high accuracy and reliability of the model for further research and analysis.

2-2- Turbocharger Selection and Matching

After ensuring the accuracy of the baseline model, a turbocharger was simulated and added to the engine model. Selecting a suitable turbocharger for a specific engine plays a decisive role in overall efficiency and directly affects brake-specific fuel consumption, emission levels, and drivability [23]. Based on the specifications of the simulated XU7JP4/L3 engine, the Garrett GT2560R turbocharger was selected.

The suitability of the GT2560R turbocharger for the XU7JP4/L3 engine was further evaluated through both manufacturer recommendations and quantitative performance analysis. According to the manufacturer's catalog, this turbocharger is designed for engines with displacements ranging from 1.4 to 2.2 L, which places the 1.8 L XU7 engine well within its intended application range.

One of the most important performance characteristics of a turbocharger is the compressor pressure ratio, which is directly related to the intake air pressure supplied to the engine. This parameter is calculated as shown in Equation 1 [16]:

$$\pi_c = \frac{p_{2c}}{p_{1c}} \quad (1)$$

In this equation, π_c is the compressor pressure ratio, p_{2c} is the compressor outlet pressure, and p_{1c} is the compressor inlet pressure.

The compressor efficiency, as another key performance indicator of the turbocharger, is calculated according to Equation 2 [25]:

$$\eta_c = \frac{\pi_c^{\left(\frac{k-1}{k}\right)} - 1}{\left(\frac{T_{2c}}{T_{1c}}\right) - 1} \quad (2)$$

In this equation, η_c is the compressor efficiency, π_c is the compressor pressure ratio, k is the specific heat ratio, T_{2c} is the compressor outlet temperature, and T_{1c} is the compressor inlet temperature.

The turbine mass flow coefficient, which plays a role in turbine performance analysis, is calculated according to Equation 3 [26]:

$$\Phi_t = \frac{\dot{m}_t \sqrt{T_{T3}}}{p_{T3}} \quad (3)$$

In this equation, Φ_t is the turbine mass flow coefficient, \dot{m}_t is the mass flow rate of gas passing through the turbine, T_{T3} is the turbine inlet temperature, and p_{T3} is the turbine inlet pressure.

To evaluate the suitability of the selected turbocharger, the engine's air demand was calculated based on the following equation for a four-stroke engine:

$$\dot{m}_{air} = \frac{\eta_v V_d N \rho_{air}}{2 \times 60} \quad (4)$$

where η_v is the volumetric efficiency, V_d is the engine displacement (m^3), N is the engine speed (rpm), and ρ_{air} is the intake air density (kg/m^3). Using the calculated mass flow rates and the required pressure ratios obtained from Equation 1, the engine operating points were plotted on the compressor map of the GT2560R, confirming that they lie within the high-efficiency islands of the compressor. This combined approach—considering both manufacturer guidelines and thermodynamic matching—validates the appropriate selection of the GT2560R for this application.

The technical specifications of the selected turbocharger are presented in Table 2. Additionally, the image of the turbocharger and the compressor performance map are shown in Figures 4 and 5, respectively [24].

Table 2 GT2560R turbocharger specifications

Number	Characteristics	Values (Unit)
1	Compressor Inducer	42 (mm)
2	Compressor Exducer	54 (mm)
3	Compressor Trim	60
4	Compressor Pressure Ratio	1.8
5	A/R	0.64
6	Turbine Inducer	53 (mm)
7	Turbine Exducer	42 (mm)
8	Turbine Trim	62

Opening the wastegate by diverting a portion of the exhaust gas flow from the turbine path significantly reduces the turbine's generated power, thereby controlling the rotational speed of the turbocharger assembly [27]. Consequently, the performance of this valve has a direct and substantial impact on turbocharger efficiency and its effects on engine performance.



Figure 4 GT2560R turbocharger

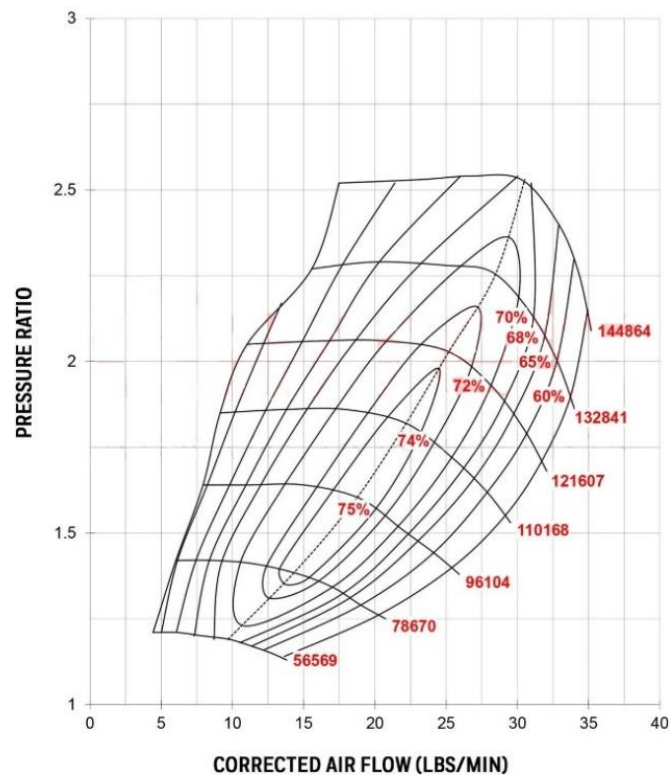


Figure 5 GT2560R compressor map

Wastegates are available in two types: internal and external. In an internal wastegate, the turbocharger boost is controlled in a fixed manner via a spring mechanism and the compressor-side inlet pressure. In an external wastegate, a separate path is provided for the valve, and the spring stiffness can be adjusted.

With mechanical wastegates, whether internal or external, it is not possible to adjust the turbocharger boost according to real-time engine demand; in the internal type, the spring stiffness is fixed, while in the external type, any adjustment to the spring stiffness applies uniformly across all engine operating speeds.

Internally controlled electronic wastegates represent a new generation of wastegate valves, providing precise control of the wastegate opening to regulate the required boost across a wide range of engine operating conditions. Electronically controlled wastegates

reduce the inertia of the turbocharger shaft and compressor and improve turbocharger efficiency and response [28]. Overall, electronic wastegates outperform mechanical ones in terms of functionality; however, due to lower cost and simpler construction, most automotive manufacturers still prefer mechanical wastegate systems. In this study, the effect of the turbocharger on the XU7 engine performance was investigated and compared under two distinct wastegate control strategies. In the first scenario, the turbocharger boost pressure was controlled by the internal mechanical wastegate of the system through inlet air pressure and the spring mechanism. In the second scenario, the turbocharger wastegate was adjusted via an electronic control mechanism according to the specifications of the simulated engine.

An intercooler with dimensions of 180×450 mm was installed in the intake air path to the throttle valve in order to cool the compressed air from the compressor. The use of an intercooler is crucial for reducing the temperature of the compressed intake air and preventing knock.

To configure the electronic wastegate control mechanism in the software, the brake mean effective pressure (BMEP) was used to determine the wastegate opening. This parameter was extracted by defining a sensor in the software map, and the data were used to select the desired wastegate positions. The values were chosen for different engine speeds to ensure optimal performance and efficiency while controlling the turbocharger shaft speed to prevent damage. Initially, various wastegate openings were tested in the software using trial and error. Based on the resulting BMEP, power, torque, and BSFC, engine performance was analyzed under these variations. The ideal BMEP values were then determined and applied to the wastegate controller in the software. The BMEP values and corresponding wastegate openings are shown in Figure 6. As illustrated in Figure 6, based on engine performance and the evaluation of key parameters at the desired operating speeds, the appropriate values were selected for the wastegate controller. The capability of precisely adjusting the wastegate according to real-time engine demand in the electronic control mode allows achieving optimal efficiency without imposing excessive stress or damage on the engine and turbocharger assembly; this key advantage is not achievable with mechanical wastegate control.

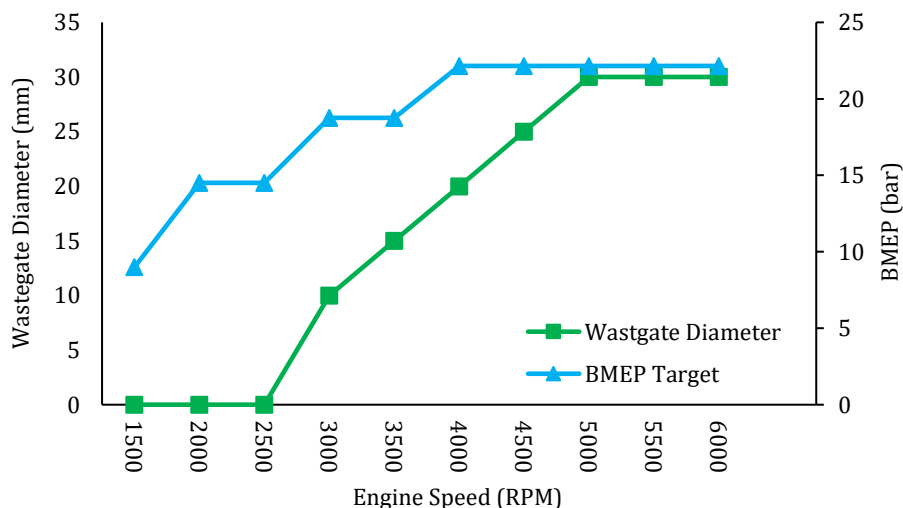


Figure 6 Target brake mean effective pressure and wastegate opening at different engine operating speeds

The maximum allowable turbocharger shaft speed, as specified by the manufacturer, is 144,864 rpm. This limit is entered into the internal wastegate controller within the software, and the wastegate opening is adjusted to achieve the target brake mean effective pressure while preventing the turbocharger shaft speed from reaching its maximum limit.

3- Results and Discussion

In this section, the simulation results of engine performance are analyzed and compared under three conditions: naturally aspirated, turbocharged with mechanical wastegate control, and turbocharged with electronic wastegate control. The key parameters examined include brake power, brake torque, BSFC, and instantaneous fuel consumption.

Figure 7 illustrates the brake power under the three conditions. As shown, in the low-speed range of 1500 to 3000 rpm, the brake power in the turbocharged mode increased by an average of 34% compared to the naturally aspirated condition.

Since the impact of turbocharging is generally more pronounced at medium to high engine speeds, at 4000, 5000, and 6000 rpm, the brake power increased by 20%, 34%, and 41% with mechanical wastegate control, and by 39%, 35%, and 44% with electronic wastegate control, demonstrating the significant effect of the turbocharger on power enhancement.

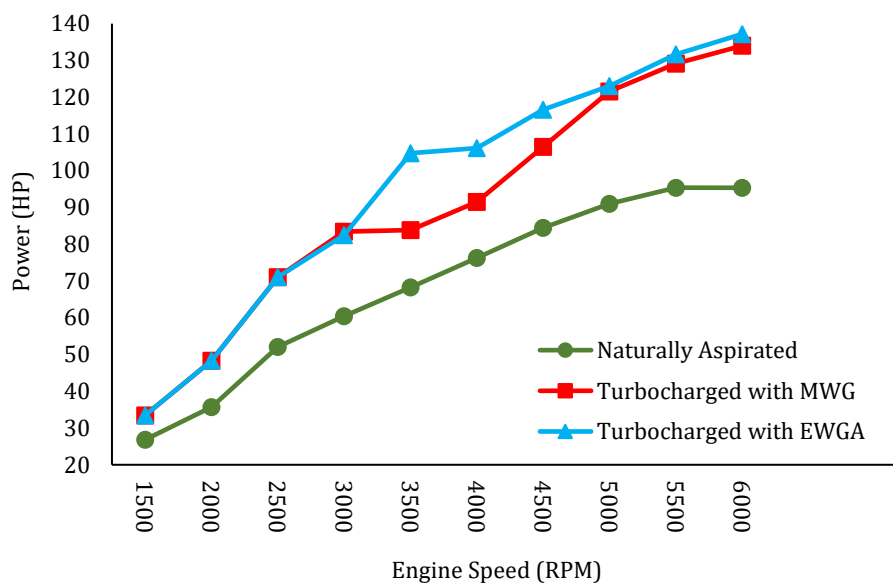


Figure 7 Brake power in turbocharged modes with mechanical and electronic wastegate control and naturally aspirated mode

Using electronic wastegate control and adjusting the turbocharger boost results in a greater impact on brake power in the mid-speed range. The brake power values for both mechanical and electronic wastegate control are identical at low engine speeds because the wastegate is fully closed to achieve maximum efficiency in this range. In the turbocharged mode with mechanical wastegate control, the wastegate opens instantaneously at mid-range speeds to prevent damage to the turbocharger assembly due to rotational speed limitations. In contrast, under electronic control, the wastegate opening is gradual and regulated, which not only prevents damage but also provides substantial brake power in the mid-speed range.

This behavior can be explained by the conservation of mass and energy in the turbine-compressor system. When the wastegate opens instantaneously in the mechanical system, a larger portion of the exhaust gas bypasses the turbine, causing a sudden drop in turbine expansion ratio. This reduces the work extracted by the turbine, which in turn decreases the compressor work and the boost pressure. The reduced boost pressure lowers the density of the intake air, thereby decreasing the mass of air trapped in the cylinders and limiting power output. In contrast, the gradual opening of the electronic wastegate maintains a more favorable pressure ratio across the turbine, sustaining compressor work and allowing for higher volumetric efficiency across a broader speed range.

From a thermodynamic perspective, the precise control of the electronic wastegate optimizes the energy transfer between the exhaust gases and the turbine. By maintaining the turbine operating points within their optimal efficiency islands, the system maximizes the extraction of exhaust gas energy, which is then converted into compression work. This results in a higher brake thermal efficiency (BTE) and explains the 3% power advantage observed at high engine speeds under electronic control, where slightly closing the wastegate allows achieving maximum power while protecting the turbocharger assembly. This precise adjustment and improved performance distinguish electronic wastegate control from mechanical control.

Figure 8 compares the simulated brake torque in both turbocharged modes with the brake torque in the naturally aspirated mode.

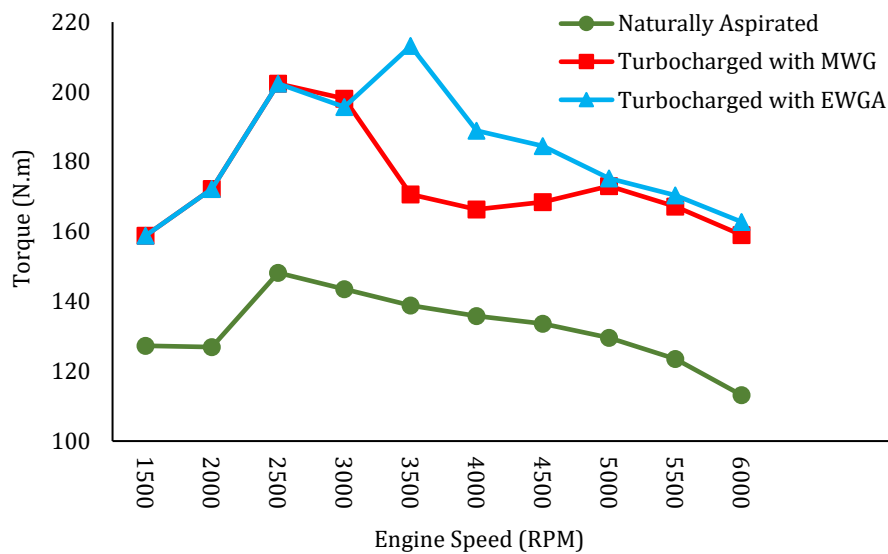


Figure 8 Brake torque in turbocharged modes with mechanical and electronic wastegate control and naturally aspirated mode

As shown in Figure 8, in the turbocharged mode with mechanical wastegate control, the brake torque at 1500 rpm (start of simulation), 2500 rpm (maximum torque), and 6000 rpm (maximum power) increased by 25%, 37%, and 41%, respectively, compared to the naturally aspirated condition. In the turbocharged mode with electronic wastegate control, the peak brake torque is shifted to 3500 rpm. This value represents an increase of 44% and 5% relative to the peak brake torque in the naturally aspirated mode and the turbocharged mode with mechanical wastegate control (which occurs at 2500 rpm), respectively.

The improved torque characteristics under electronic control can be attributed to the enhanced volumetric efficiency across the mid-speed range. By maintaining optimal boost pressure through precise wastegate control, the electronic system increases the indicated mean effective pressure (IMEP), resulting in significantly higher brake torque across the mid-speed range with the peak torque available over a wider span of engine speeds. Even at high engine speeds, the torque is 3% higher compared to the mechanical control mode. Additionally, the sudden drop in brake torque after the peak at 3500 rpm is mitigated under electronic control due to the precise and controlled wastegate opening, whereas in the mechanical control mode, the instantaneous valve operation diminishes the turbocharging effect as exhaust gas pressure rises with increasing engine speed. The wider torque curve also improves engine flexibility and drivability, as higher torque is available without the need for frequent gear changes.

Figure 9 compares the brake specific fuel consumption of the simulated engine in the two turbocharged modes with that of the naturally aspirated engine.

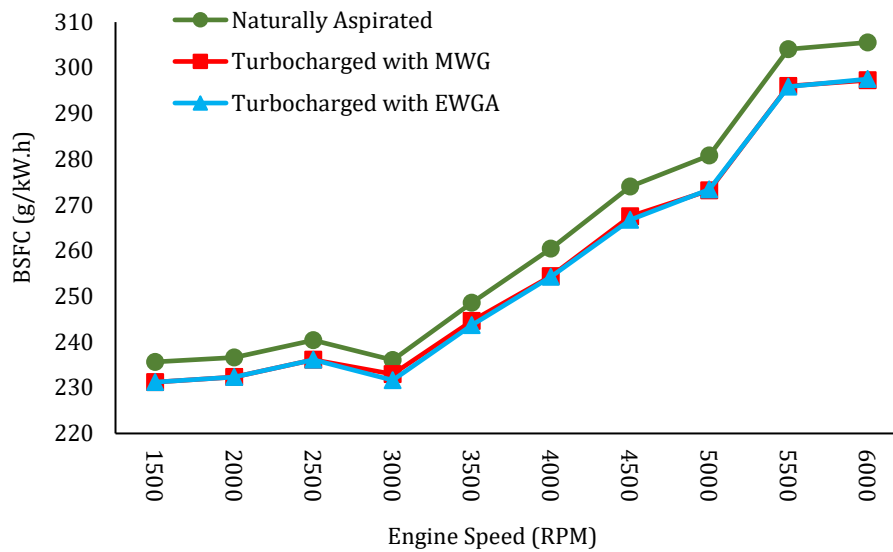


Figure 9 Brake specific fuel consumption in turbocharged modes with mechanical and electronic wastegate control and naturally aspirated mode

As shown in Figure 9, the BSFC in the turbocharged modes with mechanical and electronic wastegate control is identical between 1500 and 4000 rpm, with only a very minor difference observed at 3000 rpm. In this engine speed range, the BSFC in both turbocharged modes is on average 2% lower than in the naturally aspirated mode. From 4000 to 6000 rpm, this reduction slightly increases, reaching an average of 3%.

The reduction in BSFC under turbocharged conditions indicates an improvement in brake thermal efficiency. BSFC is inversely proportional to BTE; a 2-3% reduction in BSFC corresponds to a similar improvement in thermal efficiency. This efficiency gain arises from two main factors: first, the recovery of exhaust gas energy that would otherwise be wasted, and second, the improved combustion efficiency due to higher in-cylinder pressures and temperatures. The higher pressures promote faster flame propagation and more complete combustion, converting a greater fraction of the fuel's chemical energy into useful work.

Figure 10 shows the instantaneous fuel consumption of the simulated engine in the two turbocharged modes and the naturally aspirated mode.

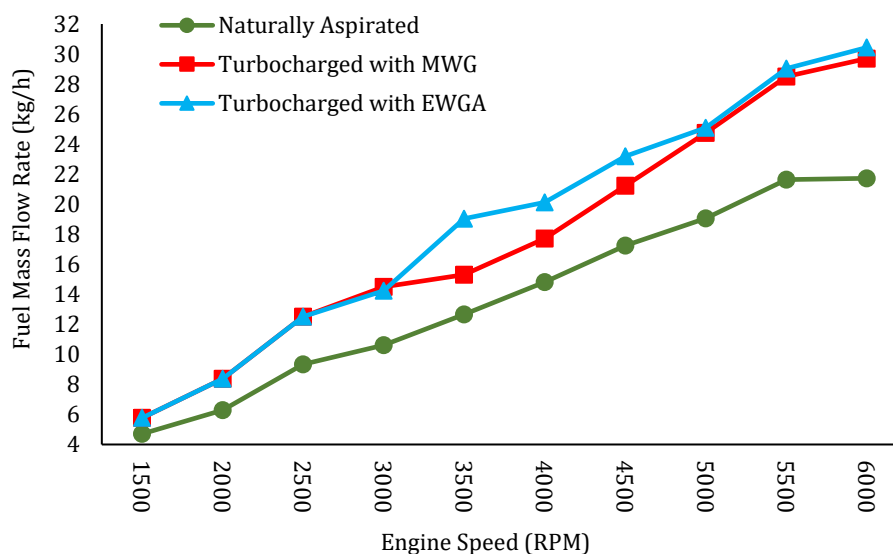


Figure 10 Instantaneous fuel consumption in turbocharged modes with mechanical and electronic wastegate control and naturally aspirated mode

As illustrated in Figure 10, at low to mid engine speeds (1500 to 3000 rpm), the instantaneous fuel consumption is identical in both turbocharged modes due to the same wastegate opening, showing an average increase of 30% compared to the naturally aspirated mode.

At 3500, 4000, and 4500 rpm, distinct behaviors emerge between the two control strategies. The mechanical wastegate control results in increases of 21%, 20%, and 23%, respectively. In contrast, the electronic wastegate control, with its precise boost regulation, achieves significantly higher power and torque in this range, accompanied by instantaneous fuel consumption increases of 50%, 36%, and 34%, respectively. This substantial increase in fuel flow is offset by proportional gains in power output, indicating that the overall energy conversion efficiency remains favorable.

At high engine speeds (5000 to 6000 rpm), the mechanical wastegate control shows a 33% increase in instantaneous fuel consumption compared to the naturally aspirated mode. The electronic control, through slightly closing the wastegate to gain 3% more power, results in a corresponding 3% increase in fuel consumption, maintaining the favorable efficiency trend observed throughout the operating range.

The use of a turbocharger without altering engine displacement demonstrates the potential to achieve performance comparable to larger engines while consuming less fuel, which is a fundamental principle of engine downsizing.

4- Conclusions

In this study, the effect of using a turbocharger along with two control mechanisms mechanical and electronic for the wastegate on the performance of the XU7JP4/L3 engine was comprehensively investigated. The key findings are as follows:

- The use of the turbocharging system resulted in an average increase of 34% in brake power at low engine speeds compared to the naturally aspirated condition.
- In the electronic wastegate control mode, brake power increased by 39%, 35%, and 44% at 4000, 5000, and 6000 rpm, respectively, showing higher values compared to mechanical control. The gradual and precise opening of the electronic wastegate resulted in a 3% increase in power at high speeds and improved turbocharger assembly stability compared to the mechanical mode.
- In the turbocharged mode with electronic control, the peak torque shifted to 3500 rpm, showing an increase of 5% and 44% compared to the mechanical control and naturally aspirated modes, respectively.
- A more uniform torque distribution and the elimination of sudden torque drops at mid-range engine speeds were other advantages of the electronic wastegate control.
- Brake specific fuel consumption in the turbocharged mode decreased by an average of 2.5% compared to the naturally aspirated mode.
- Although instantaneous fuel consumption in the turbocharged mode increased by an average of 30%, this increase is justifiable in exchange for higher power and torque, particularly under electronic control.

Overall, the results of this study clearly demonstrate that using an electronic control system for the wastegate, compared to mechanical control, not only enables higher power and torque over a wider engine speed range but also improves the efficiency and stability of the assembly, allowing for more optimal utilization of the turbocharging system's potential. Furthermore, the results of this study indicate that the addition of a turbocharger can serve as an effective step toward realizing the engine downsizing strategy.

Future research could experimentally validate the simulation results presented in this study by implementing the electronic wastegate control on an actual XU7 engine test bench. Additionally, the combined effect of electronic wastegate control with other engine optimization strategies such as variable valve timing (VVT) and exhaust gas recirculation

(EGR) on performance and emissions could be investigated. The application of advanced control algorithms, such as model predictive control, for real-time wastegate management also presents a promising direction for further improving turbocharged engine performance and efficiency.

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شبیه‌سازی تأثیر استفاده از توربوشارژر بر عملکرد موتور XU7 با استفاده از سازوکار کنترل مکانیکی و الکترونیکی دریچه هدررو

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چکیده

با توجه به اهمیت بهبود بازده موتورهای احتراق داخلی، کاهش مصرف سوخت و آلاینده‌ها، فناوری پرخوران بویژه توربوشارژر به‌عنوان راهکاری مؤثر مطرح است. توربوشارژر قابلیت تحقق مفهوم کوچک‌سازی موتور را فراهم می‌نماید. در این پژوهش، ابتدا موتور XU7JP4/L3 در نرم‌افزار جی‌تی‌پاور شبیه‌سازی و سپس مدار پرخوران به مدل موتور افزوده شد. عملکرد سامانه پرخوران با دو نوع سازوکار کنترل برای دریچه هدررو بررسی شد. در حالت اول از دریچه هدررو داخلی با سازوکار مکانیکی و در حالت دوم از سازوکار الکترونیکی استفاده گردید. نتایج پژوهش حاکی از آن است که توان ترمزی در حالت پرخوران با کنترل مکانیکی دریچه هدررو به‌طور میانگین ۳۳ درصد و با کنترل الکترونیکی دریچه هدررو ۳۷ درصد افزایش یافت. گشتاور ترمزی نیز در حالت پرخوران با کنترل مکانیکی دریچه هدررو در دوره‌های ۱۵۰۰ (آغاز شبیه‌سازی)، ۲۵۰۰ (بیشینه گشتاور) و ۶۰۰۰ (بیشینه توان) دور بر دقیقه، بترتیب ۲۵، ۳۷ و ۴۱ درصد نسبت به حالت تنفس طبیعی بالاتر رفت. استفاده از سامانه کنترل الکترونیکی دریچه هدررو، نقطه بیشینه گشتاور را به دور موتور ۳۵۰۰ دور بر دقیقه منتقل کرد و مقدار آن را نسبت به حالت کنترل مکانیکی، ۵ درصد افزایش داد. همچنین مصرف سوخت ویژه ترمزی در هر دو حالت پرخوران، در دوره‌های پایین و میانی به‌طور میانگین ۲ درصد و در دوره‌های بالا، ۳ درصد کاهش نشان داد. در مقابل، مصرف سوخت لحظه‌ای در تمام بازه کاری موتور به‌طور میانگین ۳۲ درصد افزایش یافت. در مجموع، نتایج این پژوهش برتری سازوکار کنترل الکترونیکی دریچه هدررو را در افزایش تأثیرگذاری مجموعه پرخوران بر عملکرد موتور XU7 تأیید می‌کند.

اطلاعات مقاله

کلیدواژه‌ها:

موتورهای احتراق داخلی
سامانه پرخوران
دریچه هدررو الکترونیکی
جی‌تی‌پاور
کوچک‌سازی موتور



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