Sport Utility Vehicle Refinement by Engine Upgrading and Gear Ratio Optimization

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Abstract

To improve the drivability and reduce fuel consumption and emissions of an existing sport utility vehicle (SUV), simulation-based vehicle refinement was conducted in this study. A simulation model of the SUV was first established and validated. Both simulation and test results confirm that due to low reserve power both at low and high speed bands, the drivability, fuel economy and emissions of the SUV are unsatisfactory. To refine the vehicle, complex precautions were implemented to upgrade the diesel engine, and the powertrain gear ratios were optimized with a design of experiments (DOE) approach to match the upgraded engine. Updated simulation results demonstrate that the refined vehicle is powerful enough at a sufficient speed range, with moderate fuel consumption at Euro Ⅲ standard emissions. An interim vehicle prototype, with an upgraded engine but an unmatched transmission, was produced; field test results of the interim vehicle proved the effectiveness of the simulation-based SUV refinement. The final, fully refined SUV product is currently in development.

Keywords: *Drivability, fuel economy, emissions, engine upgrading, gear ratio optimization*

1. Introduction

A sport utility vehicle (SUV) refinement project was proposed in Jiangling Motors of China to tackle the problem of unsatisfactory torque supply both at low and high speed bands and to further optimize fuel economy and emissions. The strategy was to put a new SUV on the market; this new SUV should be powerful enough at a wide speed range to meet torque demands, and it should experience moderate fuel consumption with Euro Ⅲ standard tail-gas emissions. To reach this goal with minimal cost and time, powerful computer-aided modeling and simulation were used. The project ultimately focuses on engine upgrading and powertrain gear ratio optimal matching.

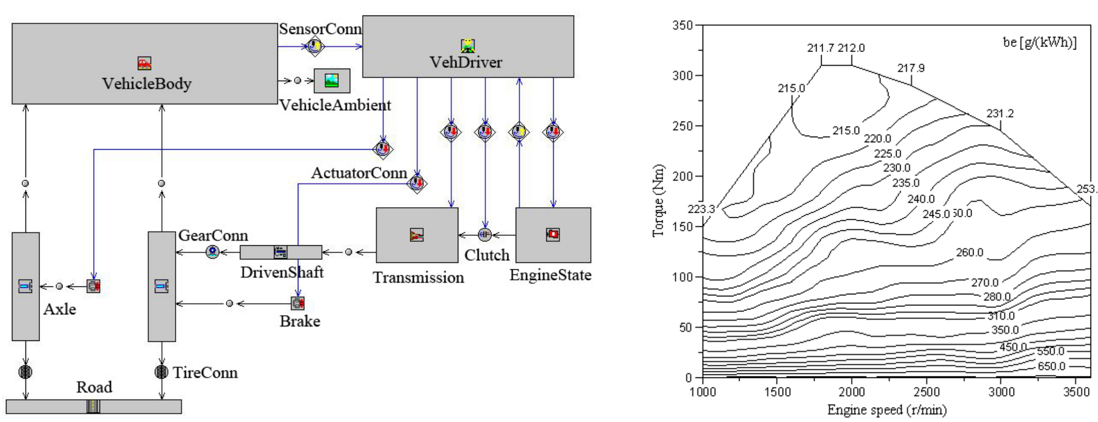
Powertrain optimization has always been the main subject of automotive design, for which a variety of cost-effective modeling and simulation methods are used. Kolmanovsky, *et al.*, [1] and Zhou, *et al.*, [2], optimized powertrain parameters to improve drivability and reduce fuel consumption and emissions, using multi-objective mathematical modeling; Wang [3] applied SIMULINK-based modeling and simulation results to engine diagnostics, powertrain design and automatic transmission shift control; Sandberg [4] developed a computer package named STARS to predict the fuel consumption of heavy-duty vehicles; Giannelli, *et al.*, [5], improved the US EPA simulator model for heavy-duty vehicle fuel consumption and CO_2 emissions; Moser, *et al.*, [6-7], employed a Hardware-In-the-Loop simulation approach in the development of an optimal powertrain; Lyu [8] and Guzzella, *et al.*, [9], optimized fuel economy based on commercial software analysis and gear ratio optimal design, respectively; Fröberg, *et al.*, [10], put forward an implicit driver model for efficient drive-cycle simulation in powertrain optimization. In addition, many works have studied the contributing factors in engine emissions [11-13] and to investigate other methods to improve fuel economy [14-16].

In this study, a simulation model of an existing SUV was first established with the commercial software GT-Drive [17], the model was validated with field test data. Both simulation and test results showed that drivability of the SUV was unsatisfactory, while fuel consumption was acceptable, despite emissions that failed to meet the Euro Ⅲ standard requirements. Complex improvements were implemented to upgrade the diesel engine, and the powertrain gear ratios were optimized with a design of experiments (DOE) approach to match the new engine. GT-Drive results demonstrated that the refined SUV was powerful enough at a wide speed range with moderate fuel consumption that met the Euro Ⅲ standard emission limits. As a first step, an interim vehicle with an upgraded engine but an unmatched transmission was produced. Field test results of the interim vehicle proved the effectiveness and high efficiency of the simulation-based SUV refinements. Consequently, these refinements are now being implemented on a final SUV prototype.

2. Vehicle Performance Modeling and Validation

The front-engine, rear-drive SUV has overall dimensions of 4740 (long)x1895 (width)x1825 (height) mm, a wheel base of 2750 mm, a curb weight of 1855 kg, a laden weight of 2510 kg and a wheel diameter of 378 mm. The vehicle has a five-gear manual transmission with gear ratios of 3.889, 2.475, 1.536, 1 and 0.807. Additionally, it has a final drive with a gear ratio of 3.818. The employed four-cylinder, four-stroke in-line diesel engine has an overall displacement of 2.4 L, a compression ratio of 17.5, a maximum power of 85 kW @ 3500 rpm, a maximum torque of 310 Nm @ 1800-2000 rpm and a minimum fuel consumption rate of 215 g/kWh.

A simulation model of the SUV was established in GT-Drive environment. Figure 1a shows the model diagram; Figure 1b gives the universal engine characteristics built into the model.



(a) **(b)**

Figure 1. Simulation Model of the SUV: (a) Model Diagram in GT-Drive; (b) Universal Characteristics of the Diesel Engine in Service

Table 1. Vehicle Performance given by both Simulation and Field Test Data

	Vehicle Performance	Simulation	Test	Error (%)
Drivability	Maximum vehicle speed (kM/h)	155.7	150	3.80%
	0-100 kM/h continuous-shift speed-up time (s)	20.5	21	2.38%
	50-100 kM/h direct-gear speed-up time (s)	21.2	22.1	4.07%
	0-400 m continuous-shift speed-up time (s)	22.9	–	–
	Maximum climbing grade (%)	45.8	45	1.78%
Fuel Economy	Fuel Consumption (FC) @ NEDC (L/100 kM)	8.34	8.41	0.80%
	FC @ UDDS (L/100 kM)	8.68	–	–
	FC @ 60 kM/h (the fifth gear) (L/100 kM)	5.20	5.31	2.07%
	FC @ 90 kM/h (the fifth gear) (L/100 kM)	7.01	7.16	2.09%
	FC @ 120 kM/h (the fifth gear) (L/100 kM)	10.26	10.38	1.20%

Table 1 demonstrates performance indices concerning the SUV's drivability and fuel economy given by both simulation and test data. The drivability results show that the SUV has an unsatisfactory maximum climbing grade about 45% at lower gears and a lower maximum vehicle speed about 150 kM/h at higher gears. A 21 seconds of 0-100 kM/h speed-up time is also too long for an SUV. The right reason leads to the above drawbacks can be illustrated by Figure 1b, which demonstrates that the diesel engine employed has only a relatively high torque level between the narrow bands of 1500-2500 rpm, outside of the narrow bands, however, the torque descends sharply on both sides, and that is to say, the engine has low reserve power.

The above low reserve power engine performance would also lead to poor fuel economy, especially those at low and high speeds. Table 1 shows that the constant-speed fuel consumption of the vehicle at 60 kM/h and 120 kM/h reaches about 5.3 L/100 kM and 10.3 L/100 kM, respectively. These figures are higher than expected levels. The reason can also be explained by Figure 1b: due to the low torque reserve at the low and high speed bands, the engine has to operate at high fuel consumption rate points, which leads to high fuel consumption. The fuel consumption at medium speeds, however, *e.g.*, 7.01 L/100 kM @ 90 kM/h, is acceptable; the driving-cycle fuel consumption of the vehicle was 8.34 L/100 kM @ NEDC [18] or 8.68 L/100 kM @ UDDS [19], which is also acceptable.

A comparison of the simulated data with the field test data in Table 1 demonstrates that the maximum relative error of drivability items was 4.07%, while that of fuel economy items was 2.09%, which are both less than 5%. Therefore, it was concluded that the simulation model was accurate in predicting performance of the vehicle. Because there are no universal engine emission characteristics as model inputs, emissions indices cannot be simulated. Pertinent test data confirm, however, that the engine emissions are better than the Euro Ⅱ standards but worse than the Euro Ⅲ standards.

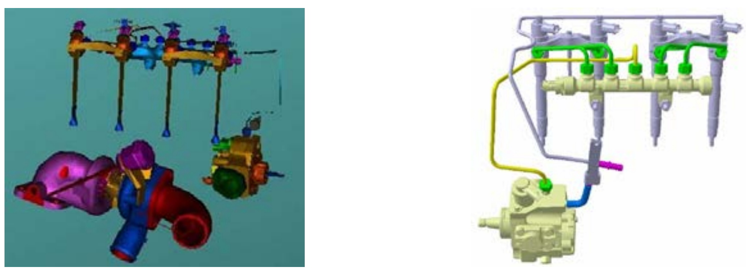
In conclusion, because the diesel engine has low reserve power at both low and high speed bands, the drivability of the SUV is insufficient. Although the fuel economy is acceptable in a general sense, it is also unsatisfactory at those speed bands. In addition, Figure 1b indicates that the working zone with a low fuel consumption rate is too small, so the room for fuel economy improvement is limited. Therefore, comprehensive upgrades to the engine should be made.

The introduced simulation model established was accurate, and thus, it can be used as an effective analysis tool in the following steps.

3. Engine Upgrading

Complex upgrades were implemented to promote more efficient combustion, less emissions and better torque performance at a wider speed range. The main improvements include upgrading the common-rail fuel injection system, matching the inter-cooled turbocharger, employing a rate-variable exhaust gas recirculation (EGR) system and using lighter materials to cut engine weight.

As shown in Figure 2, a second-generation BOSCH common-rail fuel injection system CRI2.2 was used to replace the former DENSO system in an attempt to further improve combustion and reduce emissions. The BOSCH CRI2.2 is capable of injecting five times in a single combustion cycle; of the five injections, two are pre-injections, one is the main injection and two are post-injections. The pre-injections serve to reduce combustion noise and NO_x emissions, while the post-injections aim to reduce particulate matter (PM) emissions.



(a) **(b)**

Figure 2. Common-rail Injection Systems: (a) DENSO; (b) BOSCH CRI2.2

(a) **(b)**

Figure 3. Turbochargers: (a) CARRETT; (b) MHI TD03 (with a Small Cross-Section and an Intercooler)

Matching the engine with a proper turbocharger was another centerpiece of the project. Investigations showed that the former HONEYWELL CARRETT turbocharger with a charge pressure of 100 kPa @ 3500 rpm was too high for the engine; hence, this is one of the main reasons why the old engine responded slowly and supplied inadequate torques, especially at low engine speeds. The old turbocharger was replaced by a more advanced MITSUBISHI MHI TD03 turbocharger, as shown in Figure 3.

The optimally matched MHI TD03 turbocharger has a smaller cross-section than that of the CARRETT turbocharger; the small cross-section can greatly improve the charge response, thus improving torque generation, especially at low speeds. Simultaneously, density of the charged air can be increased by the cooling effect of the intercooler of the new turbocharger; thus, the ECU-controlled injection would be increased automatically, which would lead to increased power and torque. In addition, due to a low-temperature air intake, combustion quality in the cylinders could be improved considerably, while emissions could be reduced by a significant amount.

To further cut NO_x and CO emissions, the former EGR system was also replaced by a more advanced rate-variable EGR system, as demonstrated by Figure 4. ECU-controlled EGR rates allow emissions to be reduced under various working conditions. In addition, its water cooler promotes the reduction of emissions through its cooling effect.

(a) **(b)**

Figure 4. EGR Systems: (a) Old; (b) New (EGR Rate Variable)

The BOSCH common-rail fuel injection system CRI2.2 works more efficiently with the assistance of a small-section inter-cooled turbo charging unit and with rate-variable EGR technologies. The combined action of these systems is expected to optimize the universal characteristics of the diesel engine, especially in regard to achieving the Euro Ⅲ emission standards.

In addition, light materials were broadly used to cut engine weight and to improve its efficiency. The cylinder head, cylinder block, cylinder and alternator bracket were redesigned using aluminum alloys; the connecting rod was optimized using C70S6 steel; the intake manifold, various covers, some of the pipes and the brackets were made of engineering plastics PA66. A number of other structural adjustments were also made in the project.

Multiple tests were performed to characterize [20] the performance of the upgraded diesel engine using the DYNA3-LI250 test cell, the results are demonstrated in Figure 5. Figure 5a illustrates that the full-load characteristics of the upgraded engine cover a wider speed region with high torque levels, despite a maximal torque drop relative to the old engine. The upgraded engine is capable of supplying satisfactory reserve power for acceleration at various speeds. Figure 5a also suggests that the upgraded engine can be more stable when working under high loads.

Comparing Figure 5b with Figure 1b demonstrates that fuel consumption rates of the upgraded engine are optimized, with the minimum rate declining as well as the economical region (between 1200-2800 rpm and above 150 Nm) enlarging considerably. Thus, the upgraded engine has increased potential to improve fuel economy.

Universal characteristics concerning NO_x, CO, HC and PM emissions of the upgraded engine were also obtained. As an example, Figure 5c shows the universal map of NO_x emission. Further simulation and test results verify that the emission levels of the upgraded engine meet the Euro Ⅲ standard requirements.

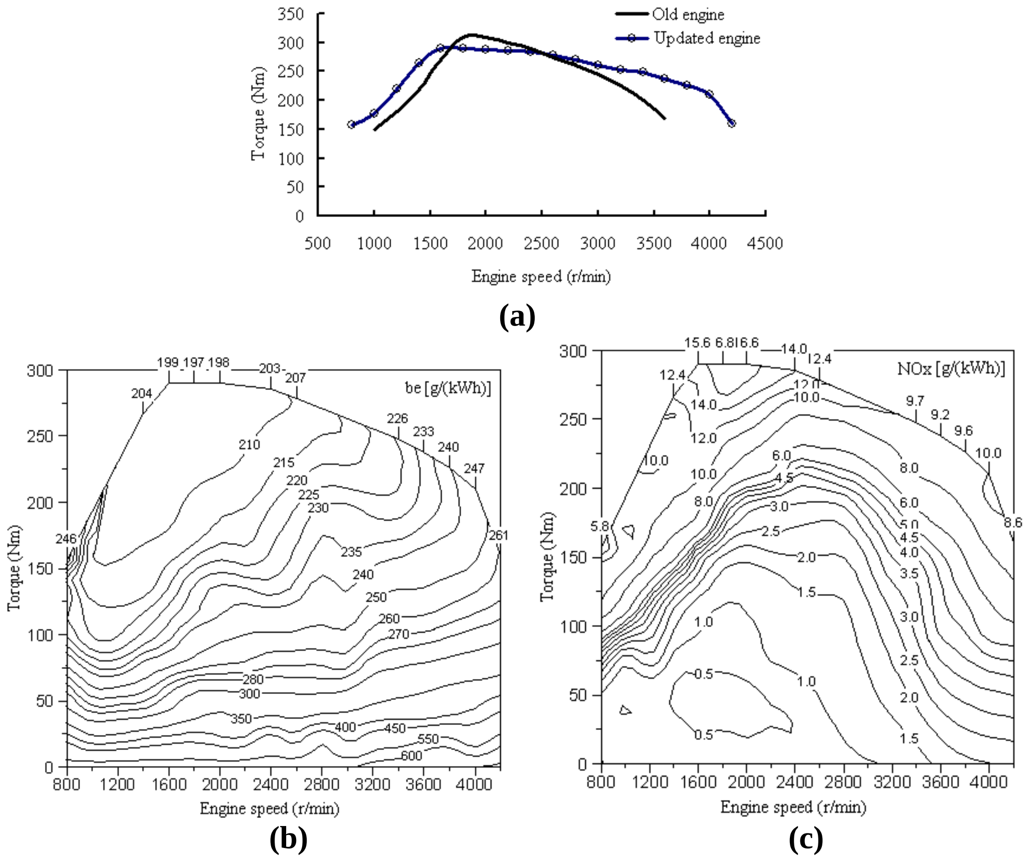


Figure 5. Performance of the Upgraded Diesel Engine: (a) Full-load Characteristics (Compared to that of the Old Engine); (b) Universal Characteristics; (c) Universal Characteristics of NO_x Emissions

In short, the upgraded diesel engine now has a maximum power of 90 kW @ 3800 rpm and a maximum torque of 290 Nm @ 1600-2200 rpm, which are sufficient values for an SUV. It has a minimum fuel consumption rate of 210 g/kWh at a full load and a fuel consumption rate below 245 g/kWh at rated working points. Additionally, the emission levels meet the Euro Ⅲ standards.

4. Gear Ratio Optimization

Although the diesel engine has experienced substantial changes and its performance has been greatly improved, vehicle performance can continue to be improved. The upgraded engine can be matched with optimal powertrain parameters so that its full operating potential can be realized. Thus, determining a proper combination of driveline gear ratios is the central goal of the following section.

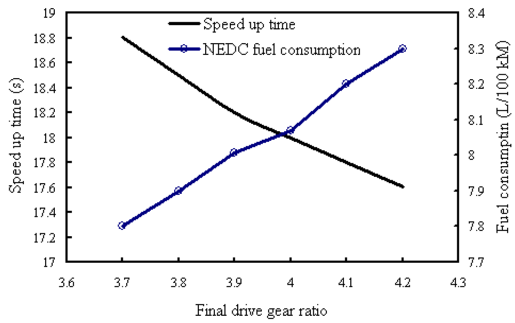
4.1. Parameter Sensitivity Analysis

Before optimizing the gear ratios, the nature of the different gear ratio effects on drivability and fuel economy of the vehicle should be understood. The established SUV model in Section 2 is used to perform this analysis, with the former engine universal characteristics being replaced by those of the upgraded engine.

Figure 6 highlights the influence of final drive gear ratio on vehicle/engine performance. Figure 6a demonstrates that an increase in final drive gear ratio leads to a reduction in speed-up time, *i.e.*, a better acceleration ability; however, an increases in this ratio also leads to an increase in fuel consumption.

The conflicting results above can be explained by Figures 6b and 6c, which show that the engine use points change under the NEDC driving cycle when the final drive gear ratio changes. Comparing Figures 6b with 6c shows that the engine-use points are more concentrated and closer to the regions with lower fuel consumption rate when the final drive gear ratio equals 3.7; however, when the ratio was increased to 4.2, the engine-use points spread to both sides, where the fuel consumption rate and the reserve power both tend to increase.

Thus, the bigger the final drive gear ratio, the better the drivability but the higher the fuel consumption of the vehicle. However, when the ratio is around 4, a tradeoff could be reached between the two conflicting performance indices to obtain acceptable performance.



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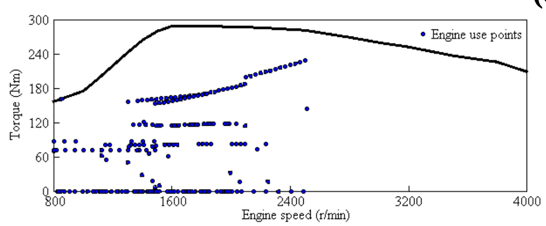

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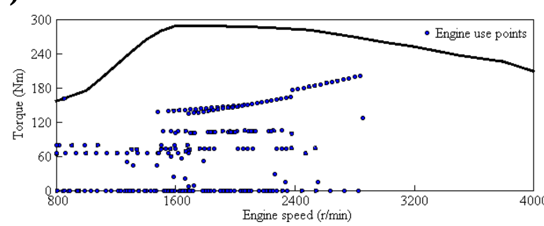


(c)

Figure 6. Influence of Final Drive Gear Ratio on Vehicle/Engine Performance: (a) Final Drive Gear Ratio versus 0-100 kM/h Speed-up Time and NEDC Fuel Consumption; (b) Engine-use Points @ NEDC (Final Drive Gear Ratio=3.7); (c) Engine-use Points @ NEDC (Final Drive Gear Ratio=4.2)

Similarly, Figure 7 illustrates the influence of the first gear ratio of transmission on vehicle/engine performance. Figure 7a shows that an increase in the first gear ratio leads to an increase in the maximum climbing grade, while not resulting in any change to the fuel consumption. Figures 7b and 7c combine to verify that with an increase in the first gear ratio, there are more engine-use points moving to the low-speed region, where the reserve power is high for climbing but the fuel consumption rate remains almost unchanged. An increase in the first gear ratio has almost no influence on the high-speed performance of the vehicle.

However, the first gear ratio of transmission should not be made too large in practical designs. With a larger first gear ratio, the total number of gears in the transmission must be increased or else the transmission becomes difficult to shift.

Additionally, the first gear ratio should be restrained [21] by driving adhesion coefficient and the minimum stable speed demand of the vehicle.

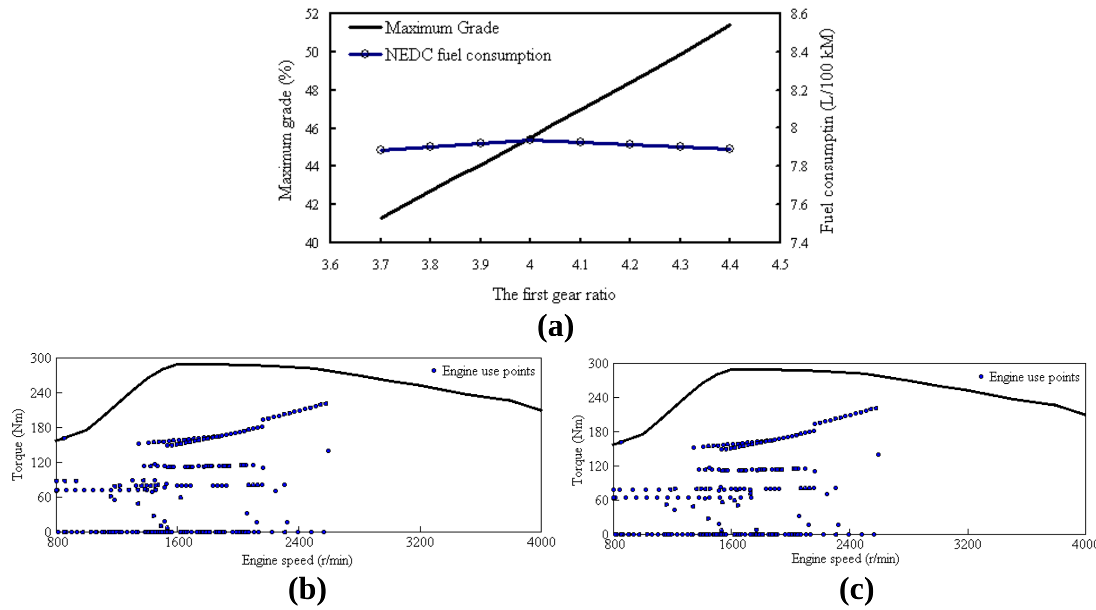


Figure 7. Influence of the First Gear Ratio of the Transmission on Vehicle/Engine Performance: (a) First Gear Ratio Versus the Maximum Climbing Grade and NEDC Fuel Consumption; (b) Engine-use Points @ NEDC (the First Gear Ratio=3.7); (c) Engine-use Points @ NEDC (the First Gear Ratio=4.4)

Figure 8 demonstrates that the fifth gear ratio of transmission has a remarkable impact on the maximum vehicle speed and on the fuel economy. Increasing the fifth gear ratio leads to both increases of the maximum vehicle speed and the fuel consumption. The cause can be explained by Figures 8b and 8c: when the fifth gear ratio is 0.6, most of the engine-use points gather in the economical region with lower reserve power; when the fifth gear ratio is increased to 0.9, however, most of the engine-use points move to the non-economical region with higher reserve power. Thus, when the fifth gear ratio is increased, both drivability and fuel consumption are increased.

Because the fifth gear is the most often used gear, the choice of its ratio is of crucial importance. Figure 8a shows that the fuel economy is best when the fifth gear ratio is 0.6, but the maximum speed is unsatisfactory at this ratio. However, when the fifth gear ratio is between 0.7 and 0.8, both of these performance indices are improved and acceptable.

In conclusion, the final drive gear ratio has a remarkable but conflicting influence on drivability and fuel economy of the vehicle, thus, a proper tradeoff should be made in choosing the ratio; the first gear ratio of transmission has a remarkable influence on the maximum climbing grade of the vehicle, so, due to the insensitivity of fuel economy on the first gear ratio, increasing the first gear ratio within constraints is more reasonable; the fifth gear ratio of transmission also has a remarkable influence on the drivability and fuel economy of the vehicle, therefore, optimization should be made to balance these two performance indices. Extensive studies show that the effects of the other gear ratios on vehicle performance are not significant and thus it is unnecessary to state more than is needed here.

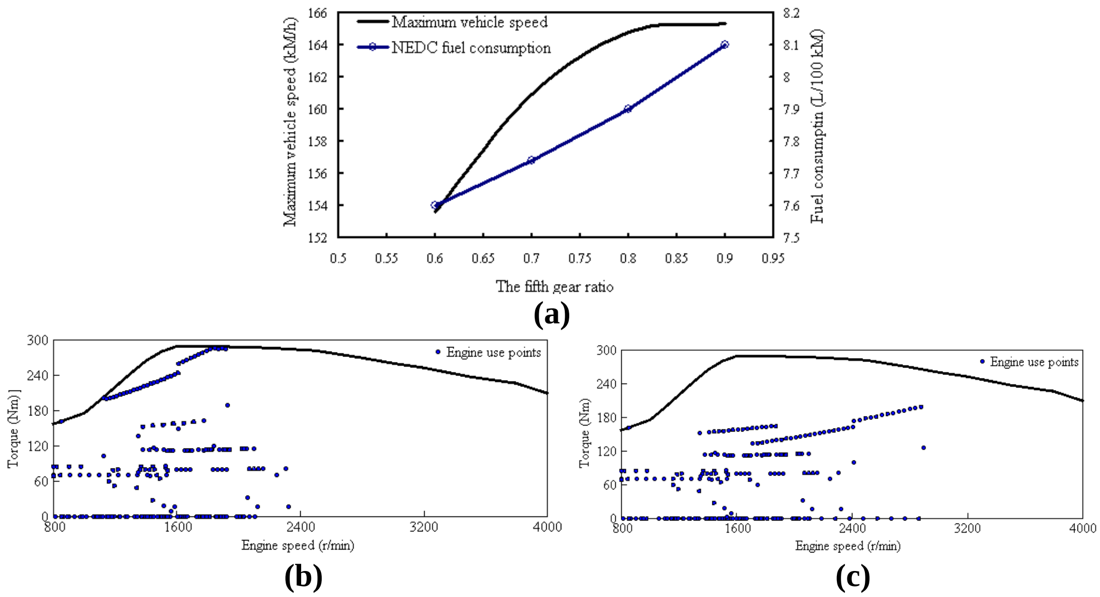


Figure 8. Influence of the Fifth Gear Ratio of Transmission on Vehicle/Engine Performance: (a) Fifth Gear Ratio Versus Maximum Vehicle Speed and NEDC Fuel Consumption; (b) Engine-use Points @ NEDC (the Fifth Gear Ratio=0.6); (c) Engine-use Points @ NEDC (the Fifth Gear Ratio=0.9)

4.2. Gear Ratio Optimization by DOE

The objective of the gear ratio optimization problem with DOE is to find the combination of gear ratios that causes the SUV to have the best drivability and moderate fuel consumption according to the Euro III standard emission requirements.

The concrete constraints include the following:

(1) The NEDC fuel consumption is less than 8.1 L/100 kM;

(2) The NEDC emissions are as follows [22]: $CO \leq 0.95$ g/kM, $HC+NO_x \leq 0.86$ g/kM, $NO_x \leq 0.78$ g/kM and $PM \leq 0.1$ g/kM;

(3) The gear ratios should meet $(i_1/i_2) \geq (i_2/i_3) \geq (i_3/i_4) \geq (i_4/i_5)$, as well as $i_1/i_2 \leq 1.7$ ~ 1.8 and $1.2 \leq i_4/i_5 \leq 1.45$ [21], where i_1 ~ i_5 are the first to the fifth gear ratios of transmission;

(4) The maximum tractive force $F_{tmax} \leq F_z\varphi$, where F_z is the total normal reaction force of the road and φ is the driving adhesion coefficient; for an SUV, $\varphi = 0.5$;

(5) For an SUV, the maximum transmission ratio, i_0i_1, should meet the minimum stable vehicle speed requirement [21], i.e., $(0.377n_{min}r)/(i_0i_1) \geq v_{min}$, where n_{min} is the minimum stable engine speed, r is the tire diameter, and v_{min} (v_{min}= 5 kM/h) is the minimum stable vehicle speed.

A partial factorial DOE approach, the Latin Hypercube [23], was used to implement the gear ratio optimization. The approach was intended to determine the relationship between the dependent (response) and independent (factor) variables while running fewer experiments than would be required for a full factorial design. In this design, it was necessary to define the minimum and maximum values for each factor.

Based on extensive parameter sensitivity analysis introduced in Section 4.1, five factors and their ranges for DOE optimization were finally determined, as follows: $i_0 \in [3.7, 4.2]$, $i_1 \in [3.7, 4.4]$, $i_2 \in [2.3, 2.6]$, $i_3 \in [1.4, 1.6]$ and $i_5 \in [0.6, 0.9]$. Because i_4 is the direct-gear ratio 1, it was not assigned as a factor. After implementation, the target combination of optimal gear ratios was obtained from a total of 249 experiments; the results are offered in Table 2.

Table 2. The Powertrain Gear Ratios Before and After Optimization

	Transmission				Final drive
	i_1	i_2	i_3	i_5	i_0
Former	3.889	2.745	1.536	0.807	3.818
Optimized	4.330	2.530	1.570	0.800	4.000

4.3. Simulation Results and Implementation

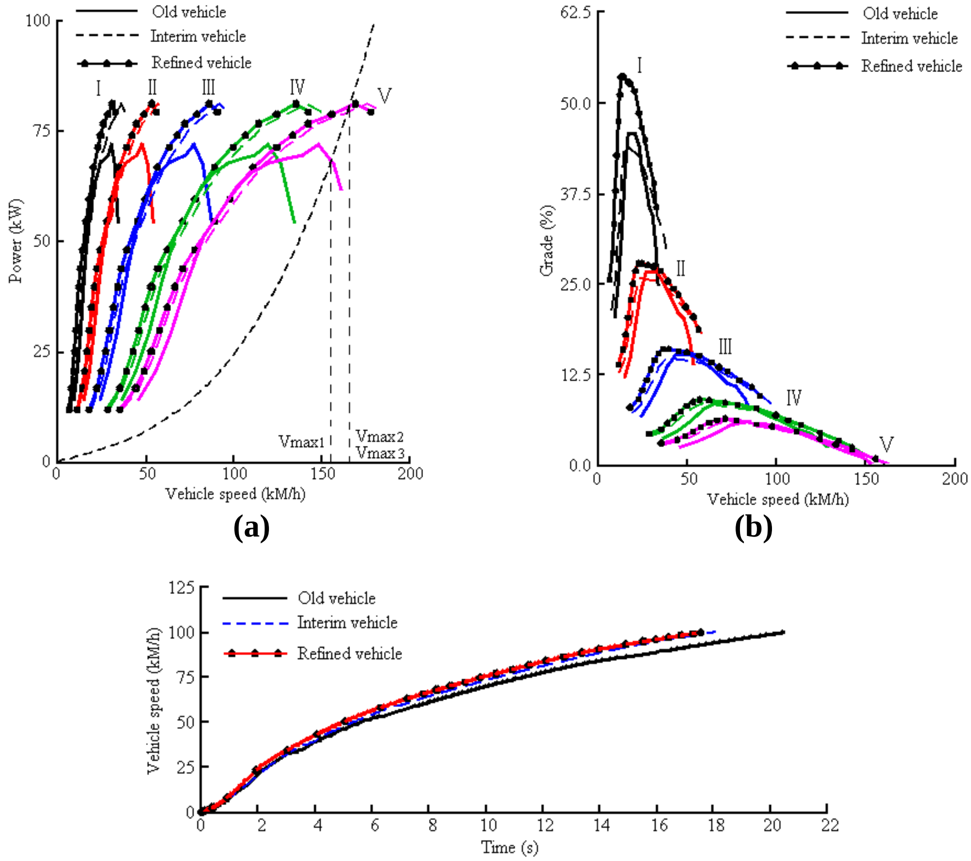
Based on the model established in Section 2, simulations were performed again to compare characteristics of three vehicles: the old vehicle, the interim vehicle with an upgraded engine but with unmatched powertrain gear ratios and the refined vehicle with an upgraded engine and optimally matched powertrain gear ratios.

Figure 9 illustrates driving performance of the three vehicles. Figure 9a shows that the tractive powers of both the interim vehicle and the refined vehicle have been enhanced considerably relative to that of the old vehicle. Consequently, their reserve power and maximum vehicle speed have also increased considerably. The tractive powers of the interim vehicle and the refined vehicle are very close.

Despite the curve of each gear of the interim vehicle covers a wider speed range than that of the old vehicle, Figure 9b shows that the maximum climbing grade of the interim vehicle has dropped. Thus, the slope climbing ability of the interim vehicle is still unsatisfactory. After gear ratio optimization, however, maximum climbing grades, especially at the first gear, have increased significantly. Thus, the general climbing ability of the refined vehicle has greatly improved.

Figure 9c shows that in spite of being similar, the speed-up performances of both the interim vehicle and the refined vehicle have improved considerably relative to that of the old vehicle.

Table 3 lists the main drivability indices of the three vehicles. Data about the old vehicle are from field test, while data about the two other vehicles are from simulation. Relative to that of the old vehicle, the maximum vehicle speed and the 0-100 kM/h speed-up ability of the interim vehicle have increased by 10.1% and 13.8%, respectively, but the maximum climbing grade has dropped by 2.2%, which is unsatisfactory. However, with the optimally matched powertrain gear ratios, not only have the maximum vehicle speed and the 0-100 kM/h speed-up ability of the refined vehicle increased by 10.8% and 16.2%, respectively, but its maximum climbing grade has also increased by 19.1%. Thus, the drivability of the refined vehicle is excellent for an SUV.



(c)

Figure 9. Drivability Performance of the Vehicles: (a) Power Balance Figure at Zero Grade between Tractive Power and Resistance Power; (b) Constant-speed Slope Climbing Ability; (c) 0-100 kM/h Continuous-shift Speed-up Performance

Table 3. Drivability Indices of the Three Vehicles

Drivability	Old vehicle (test data)	Interim vehicle (relative to that of old vehicle)	Refined vehicle (relative to that of old vehicle)
Maximum vehicle speed (kM/h)	150	165.1 (↑10.1%)	166.2 (↑10.8%)
Maximum climbing grade (%)	45	44 (↓2.2%)	53.6 (↑19.1%)
0-100 kM/h continuous-shift speed-up time (s)	21	18.1 (↓13.8%)	17.6 (↓16.2%)
50-100 kM/h direct-gear speed-up time (s)	22.1	18.92 (↓14.4%)	17.9 (↓19.0%)
0-400 m continuous-shift speed-up time (s)	–	20.68	20

Figure 10 illustrates the fuel economy performance of the three vehicles. It suggests that the constant-speed fuel consumption and the driving-cycle fuel consumption of the interim and the refined vehicle, although being similar, have been reduced considerably relative to that of the old vehicle. The fuel consumption of the refined vehicle is a little higher than that of the interim vehicle, for fuel economy of the refined vehicle was sacrificed moderately during gear ratio optimization.

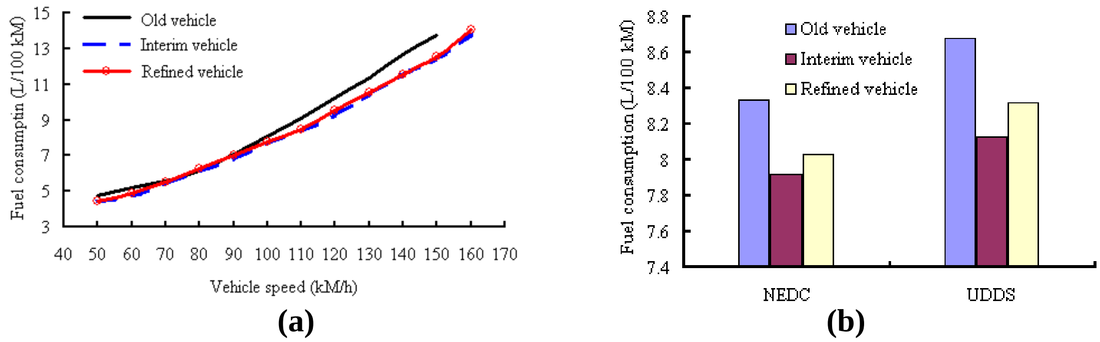


Figure 10. Fuel Economy Performance of the Three Vehicles: (a) Constant-speed Fuel Consumption (with the Fifth Gear); (b) Driving-cycle Fuel Consumption

Table 4. Fuel Economy Indices of the Three Vehicles

Fuel economy	Old vehicle (test data)	Interim vehicle (relative to that of old vehicle)	Refined vehicle (relative to that of old vehicle)
FC @ 60 kM/h (the fifth gear) (L/100 kM)	5.31	4.77 (↓10.2%)	4.9 (↓7.7%)
FC @ 90 kM/h (the fifth gear) (L/100 kM)	7.16	6.84 (↓4.5%)	7.03 (↓1.8%)
FC @ 120 kM/h (the fifth gear) (L/100 kM)	10.38	9.26 (↓10.8%)	9.52 (↓8.3%)
FC @ NEDC (L/100 kM)	8.41	7.92 (↓5.8%)	8.03 (↓4.5%)
FC @ UDDS (L/100 kM)	—	8.13	8.32

Table 4 lists the main fuel economy indices of the three vehicles. Relative to that of the old vehicle, the 60 kM/h and 120 kM/h constant-speed fuel consumptions of the interim vehicle have dropped by 10.2% and 10.8%, respectively. Thus, after the engine upgrade, both the low-speed and high-speed fuel economies of the interim vehicle have improved considerably. Additionally, the medium-speed fuel economy of the interim vehicle has also dropped moderately by 4.5%. The NEDC (or UDDS) fuel consumption is a more objective index in estimating everyday fuel use; Table 4 also shows that the NEDC fuel consumption of the interim vehicle has dropped by 5.8%.

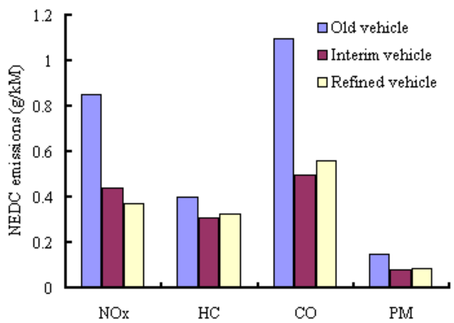
Table 4 demonstrates that the fuel consumption indices of the refined vehicle, relative to those of the old vehicle, have also dropped considerably. Despite being a little higher than that of the interim vehicle, the fuel consumption indices of the refined vehicle, *e.g.*, 8.03 L/100 kM @ NEDC, are also moderate and acceptable.

Figure 11 and Table 5 combine to illustrate emissions of the three vehicles. First of all, it can be verified from the above results that overall emission levels of the old vehicle were better than the Euro Ⅱ standards but worse than the Euro Ⅲ standards [22], while the two other vehicles were both better than the Euro Ⅲ standards.

Figure 11a and Table 5 demonstrate that various emissions of both the interim and the refined vehicle have dropped drastically when compared to those of the old vehicle, *e.g.*, NO_x, HC, CO and PM emissions of the refined vehicle under the NEDC driving cycle have dropped by 56.5%, 17.5%, 49.1% and 40.0%, respectively. That outcome is due to the inclusion of the BOSCH common-rail fuel injection system, the small-section inter-cooled turbo charging unit and the rate-variable EGR technologies in the refined system design.

Figure 11 and Table 5 also show that the emissions of the interim and the refined vehicle are on the same scale, despite there being moderate variations between the corresponding indices of the two vehicles.

Therefore, with the upgraded engine and the optimally matched powertrain gear ratios, the refined vehicle has improved drivability at a wider speed range and now boasts acceptable fuel economy at Euro Ⅲ standard emissions. Although the interim vehicle has obtained improved driving performance, its maximum climbing grade is still unsatisfactory.



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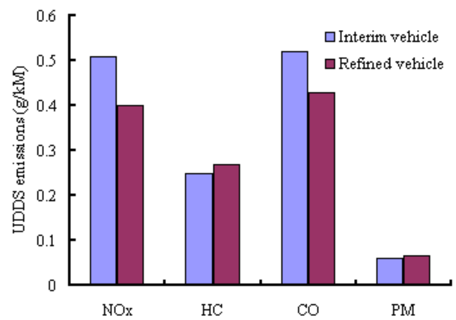

(b)

Figure 11. Emissions of the Three Vehicles: (a) NEDC Emissions; (b) UDDS Emissions

Table 5. Emissions Indices of the Three Vehicles

Emissions		Old vehicle (test data)	Interim vehicle (relative to that of old vehicle)	Refined vehicle (relative to that of old vehicle)
Emissions@ NEDC (g/kM)	NO_x	0.85	0.44 (↓48.2%)	0.37 (↓56.5%)
	HC	0.40	0.31 (↓22.5%)	0.33 (↓17.5%)
	CO	1.10	0.50 (↓54.5%)	0.56 (↓49.1%)
	PM	0.15	0.08 (↓46.7%)	0.09 (↓40.0%)
Emissions@ UDDS (g/kM)	NO_x	–	0.51	0.40
	HC	–	0.25	0.27
	CO	–	0.52	0.43
	PM	–	0.06	0.07

As a first step, an interim SUV JX6470T4, as shown by Figure 12, which had the upgraded engine, a new body style but with the old chassis and unmatched transmission, was produced. Field test results of the interim vehicle are very close to the simulation results offered by this study. Thus, the simulation-based SUV refinement introduced here is effective. As a second step, the development of a fully refined SUV, which has both the upgraded engine and optimally matched powertrain gear ratios, is currently underway.

Figure 12. The Interim Vehicle Prototype JX6470T4

5. Concluding Remarks

A validated simulation model of an existing SUV was established in GT-Drive environment to act as a benchmark during its refinement work. Both simulation and test results confirmed that because the diesel engine in-service has low reserve power at the low and high speed bands, the drivability and the fuel economy of the SUV is unsatisfactory. In addition, the working zone is too small for a low fuel consumption rate, while engine emissions are worse than those required by Euro Ⅲ standards.

To refine the SUV, the BOSCH common-rail fuel injection, a small-section inter-cooled turbo charging unit and rate-variable EGR technologies were combined to upgrade the diesel engine; Based on parameter sensitivity analysis, the powertrain gear ratios of the SUV were optimized by the DOE approach to match the upgraded engine.

The following simulation results verify that the refined SUV has reached the optimization goals. The maximum vehicle speed, the 0-100 kM/h speed-up ability and the maximum climbing grade have increased by 10.8%, 16.2% and 19.1%, respectively, relative to that of the old vehicle; the fuel consumption indices, e.g., 8.03 L/100 kM @ NEDC, are moderate and acceptable; the NO_x, the HC, the CO and the PM emissions, under the NEDC driving cycle, have also drastically dropped by 56.5%, 17.5%, 49.1% and 40.0%, respectively. Moreover, the emission levels now meet the Euro Ⅲ standard requirements.

That implementation of an interim SUV and subsequent field test results has proven the effectiveness and efficiency of the simulation-based refinements. Completion of the final, refined SUV, which includes both the upgraded engine and the optimally matched powertrain gear ratios, is forthcoming.

Therefore, by combining the engine upgrading with the powertrain gear ratio optimization, this study has made global performance improvements of the former SUV, especially in regard to drivability and the tail-gas emissions. The employed computer-aided modeling and simulation approach has reduced the cost and time of the refinement project.

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