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Experimental Study of Energy-Saving Low Friction Oil on Small Turbocharged Engine

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Abstract:

In order to analyze the fuel economy improvement of energy-saving low friction oil on a small turbocharged engine under WLTC, eight oil candidates were blended by adding different kinds and doses of friction modifiers to the existing 5W30 oil formula from the perspective of commercialization, and the piston ring cylinder friction test was carried out to screen the formula, and then followed by the friction torque test and vehicle chassis dyno test on WLTC condition. The test results showed that the energy-saving low friction oil formula with fatty acid ester and MoDTC friction modifier can reduce the engine friction torque by 6.58% to 8.6%, and improve the fuel economy by 0.88% to 2.01%. According to the General Motors engine parts research and development system, GED test and GETC test were carried out to verify the influence of low friction oil formula on engine reliability. The test results showed that the energy-saving low friction formula with composite friction modifier can ensure engine reliability and has practical value.

Keywords*: Low friction oil, WLTC, Friction torque, Fuel economy improvement, Reliability.*

I. INTRODUCTION

With the rapid development of the automobile industry, the number of automobile has increased sharply. Governments, car manufacturers and users are paying more and more attention to fuel consumption. Energy conservation and emission reduction have become the driving force for the development of the automobile industry. Various national passenger car fuel consumption standards have been introduced, such as the United States average fuel economy standards “Corporate Average Fuel Economy”, Japanese fuel consumption limits for passenger cars "Energy use rationalization law", Chinese standard "Fuel consumption limits for passenger cars"(GB 19578), etc. At present, China's fuel consumption reduction requirements in the fifth phase of fuel consumption standard from 2020 to 2025 are the most stringent in the world [1, 2]. In order to meet the fuel consumption laws and regulations, small turbocharged engines are the mainstream trend, and among the energy-saving and emission reduction

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technologies, energy-saving low friction oil is the most cost-effective way [3-5]. Therefore, it is of great significance to study the friction reduction and consumption reduction of energy-saving low friction oil in small turbocharged engines for automobile enterprises.

Energy-saving low friction oil contributes mainly from two aspects: one is low viscosity to reduce internal friction when oil flows; the other is to add efficient friction improvers to reduce friction coefficient so as to achieve friction reducing and energy saving [6]. But due to relatively limited level of engine manufacturing technology, the reduced oil viscosity exacerbate the wear problem of the piston-cylinder liner friction pair caused by micro convex rough peak contact at top and bottom dead center. So the OEM (Original Equipment Manufacturer) can achieve the goal of anti-friction and energy saving by efficient friction improver, while maintaining the oil viscosity to ensure durability.

At present, the friction improver commonly used in the formulation of energy-saving low friction engine oil mainly includes organic friction improver and oil-soluble organic molybdenum friction improver. Organic friction improver includes amides, phosphorous and phosphoric acid derivatives as well as organic polymers, etc., which can avoid metal contact by forming molecular layers on metal surfaces through adsorption of polar groups[7,8], while oil-soluble molybdenum friction improver is mainly MoDTC (Molybdenum Dithiocarbamate). Its mechanism is to form a MoS2 thin layer on the metal surface under high temperature conditions to reduce friction [9]. Foreign scholars' researches on energy-saving low friction oil formula mainly focused on the influence of low viscosity or ultra-low viscosity oil formula on fuel saving performance and other performance of diesel engines. General Motors' Salvatore studied the energy-saving potential of low viscosity oils and lightweight crankshafts on diesel oils [10]. Bernardo et al. studied the fuel consumption performance of low viscosity oil under actual road conditions in heavy-duty CNG and diesel buses [11]. Takumaru et al. studied the influence of ultra-low viscosity 0W16 oil on timing chain wear while saving energy [12]. However, domestic scholars' researches on the formulation of energy saving engine oil started late, mainly focusing on the friction reduction research of MoDTC and its compatibility with other additives. Hao Lichun et al. studied the friction performance of MoDTC in ultra-low viscosity 0W16 engine oil formulation by SRV and spectrometer [13]. Wang Wen et al. investigated the influence of different detergents on the detersive and anti-wear performance of base oil containing MoDTC through the simulated friction test of crankcase and the anti-wear test of four-ball machine [14].

In conclusion, the domestic and foreign scholars have a lot of research on the mechanism of friction improver and the friction performance of friction improver, but lack in the research on the influence of friction and reliable performance of small turbocharged engine body

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Nevertheless, the research on friction reduction and energy saving based on WLTC (Worldwide harmonized Light vehicles Test Cycle) condition in the China's fifth stage fuel consumption standard is even less. In this paper, the energy-saving low friction oil formula was prepared by adding different friction modifiers into the factory fill oil of a small turbocharged engine. With the self-built piston ring-cylinder liner friction test and the engine friction torque test based on the WLTC condition, the fuel saving effect of low friction oil was studied in combination with the vehicle fuel consumption test, and GED (Global Engine Durability Test) and GETC (Global Engine Thermal Cycle) tests were carried out according to the GMW (GM Worldwide Engineering Standard) standard to verify its influence on the reliable performance of the engine.

II. ENERGY-SAVING LOW FRICTION ENGINE OIL

The production friction modifiers were added in the oil formula include phosphate, glycerol monolete, fatty acid amide and fatty acid ester four kinds of organic friction improver and MoDTC. Taking the factory fill oil as the baseline oil (oil sample ①), eight low friction oil samples were prepared by adding 0.8% of different types of organic friction reducers and different doses of MoDTC friction modifiers on the basis of the existing oil formula from the perspective of commercial and cost performance, as shown in TABLE I.

The physical and chemical properties of the energy-saving low friction oil prepared above were shown in TABLE II. The results showed that the addition of different doses and types of friction modifiers on the basis of oil sample① had no significant influence on the physical and chemical properties of the oil, such as viscosity parameters, sulfate ash content and total alkali value.

TABLE I. Basic information of samples

NO.	**NAME & QUANTITY**
SAMPLE①	Baseline oil (Existing oil formulation)
SAMPLE②	Baseline oil + 0.8% phosphite
SAMPLE③	Baseline oil + 0.8% glycerol monooleate
SAMPLE④	Baseline oil + 0.8% fatty acid amide
SAMPLE⑤	Baseline oil + 0.8% fatty acid ester
SAMPLE⑥	Baseline oil + 700ppm MoDTC

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SAMPLE⑦	Baseline oil + 700ppm MoDTC + 0.8% fatty acid ester
SAMPLE⑧	Baseline oil + 1000ppm MoDTC + 0.8% fatty acid ester

It is worth noting that the anti-wear performance under extreme pressure of oil samples was improved in four ball tests, which indicated that the protective film formed by the friction improver on the metal surface had the effect of friction reduction and some anti-wear protection.

III. ENERGY-SAVING FRICTION TEST

Once the energy-saving low friction oil samples were prepared, in order to accurately evaluate the fuel-saving effect of the oil samples, the simulated friction test, engine fuel economy test and vehicle fuel economy test were be carried out step by step in three stages, according to the standard oil development process.

In the simulated friction test stage, HFRR (High Frequency Reciprocating Rig) or MTM (Mini Traction Machine) and other test methods were usually used to simulate the piston-cylinder friction system to conduct friction performance research and preliminary screening of oil samples [15]. However, such testing methods are all carried out on standard samples and in working conditions, so it is difficult to truly reflect the influence of energy-saving low friction oil on the frictional performance of the piston-liner friction pair of the small turbocharged engine. Therefore, the simulated friction test in the paper was carried out by cutting the piston ring of the engine and a part of the cylinder liner entity to independently build the piston ring-cylinder liner test bench.

After preliminary screening of the formula, automobile companies currently choose the backward towing engine method to measure friction torque loss to evaluate engine fuel economy [16]. However, the test is not instructive, as its test condition differs greatly from the current fuel consumption standard. Therefore, in this paper, based on the WLTC condition, the test samples and conditions were redesigned to accurately evaluate the influence of engine friction loss of the low friction oils.

After the engine fuel economy evaluation test, the vehicle fuel consumption test was carried out, which is the most intuitive and important test method for automobile enterprises to evaluate the fuel saving performance of oils.

TABLE II. Physical and chemical parameters of oil samples

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Parameter	Sample①	Sample ②	Sample ③	Sample ④	Sample ⑤	Sample ⑥	Sample ⑦	Sample ⑧
KV40℃ (cSt)	61.40	61.52	61.75	60.82	61.59	61.65	61.78	61.89
KV100℃ (cSt)	10.27	10.34	10.27	10.17	10.30	10.32	10.33	10.35
HTHS, 150℃ (cp)	3.14	3.15	3.11	3.13	3.13	3.15	3.15	3.16
Viscosity index	156.00	157.00	155.00	155.00	156.00	156.00	156.00	156.00
Cold start viscosity, -30℃ (m·Pas)	5714.45	5802.27	5861.18	5762.68	5724.12	5938.09	5945.36	6043.99
Pour point (℃)	-39	-40	-41	-40	-39	-41	-40	-40
Flash point (℃)	243	247	245	241	239	241	243	246
Sulfate ash (%)	0.82	0.81	0.82	0.83	0.81	0.82	0.81	0.83
TBN (mg/g)	7.09	7.10	7.12	7.11	7.08	7.09	7.10	7.10
Deposit (mg)	34.80	35.20	34.90	35.00	34.70	34.80	34.80	34.90
Gel index	5	5	5	5	5	5	5	5
Noack (%)	9.7	9.6	9.7	9.8	9.7	9.6	9.8	9.7
Four ball test (N)								
P_B Load	736	785	883	883	834	883	932	981
P_D Load	1962	1962	1962	2453	2453	1962	1962	2453

3.1 Piston Ring-Cylinder Liner Test

3.1.1 Test preparation

The lubrication performance of piston ring-cylinder liner friction pair has a significant influence on the reliable performance and service life of the engine.

The engine piston ring was cut into section (about 50 mm) and the cylinder liner section (about 30 mm x 20 mm). Then the piston ring section was installed on the mobile arm bracket, and cylinder liner section in a shallow tray. The heat resistant tray contained oil and thermocouple for temperature control. The tray was filled with approximately 15ml of the test oil sample to immerse the cylinder liner, and the verticality of the load and the horizontal reciprocating arm were checked to ensure proper alignment of the sample, gaskets were used if necessary to achieve proper alignment. Through the reciprocating movement and surface contact and load and oil temperature control, to fully simulate actual piston ring-liner friction

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pair near the dead center of the working environment and lubrication state, so as to study the friction performance of energy-saving low friction oil samples at the critical conditions. The bench layout diagram was shown in figure 1.

As mentioned above, the test bench included servo motor power drive system, oil supply system, loading system, temperature control system and friction acquisition system. The equipment information is shown in TABLE III. The load was manually loaded by the loading screw, the test temperature was controlled within ±1℃ by the temperature control meter, and the friction was collected by the piezoelectric sensor.

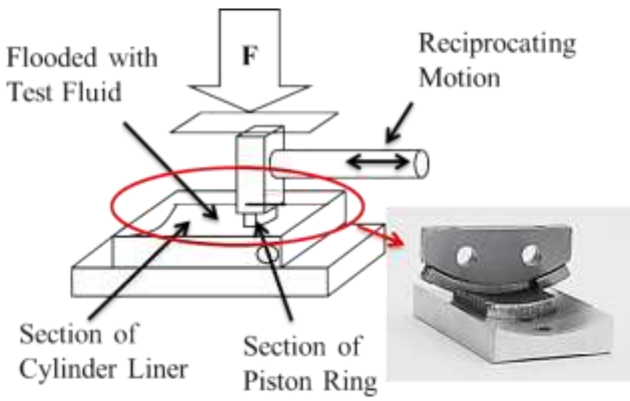

Fig 1: Bench layout diagram of piston ring-cylinder liner test

TABLE III. Basic information of test equipment

EQUIPMENT	TYPE	SPECIFICATION	PRECISION
MOTOR	YVP160N-4	5～3000rpm	±3rpm
OIL SUPPLY	BT100-1f Peristaltic pump	0.01～0.2 ml/min	±0.001 ml/min
LOADING	Screw	5N～10kN	±2N
TEMPERATURE CONTROL	Heating rods	200W	-
	WRN-440	25～300℃	±1℃
DATA COLLECTION	DL-YB-2/1tB	≤300N	±0.5%FS
	LABVIEW	-	-

According to the key operating condition (2000rpm, 2bar) of the small turbocharged engine, the working environment and lubrication state of the piston ring-cylinder liner, the parameters such as load, frequency and oil temperature were selected for the test shown in TABLE IV, and

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the test site was shown in Figure 2.

TABLE IV. Piston ring-cylinder liner friction test parameters

LOAD F (N)	FREQUENCY (Hz)	TEMPERATURE (℃)	AMPLITUDE (mm)	TIME (min)
100	13	115	10.8	20

Fig 2: Piston ring - cylinder liner friction test

3.1.2 Analysis of piston ring-cylinder liner test results

The piston ring-cylinder liner friction test results of 8 energy-saving low friction oil samples were shown in Figure 3. The low friction oil with friction improver had different degrees of friction reducing effect on the piston ring-cylinder liner friction pair, with a decrease range of 8.92% to 67.52%. The single agent of friction improver was MoDTC > fatty acid ester > phosphite > fatty acid amide > glycerol monooleate in the order of friction reducing effect. The combined effect of fatty acid ester and MoDTC friction improver was stronger than that of single agent. In order to save the test resources, four oil samples (⑤, ⑥, ⑦ and ⑧) were selected to carry out the engine friction torque test and the vehicle fuel consumption test.

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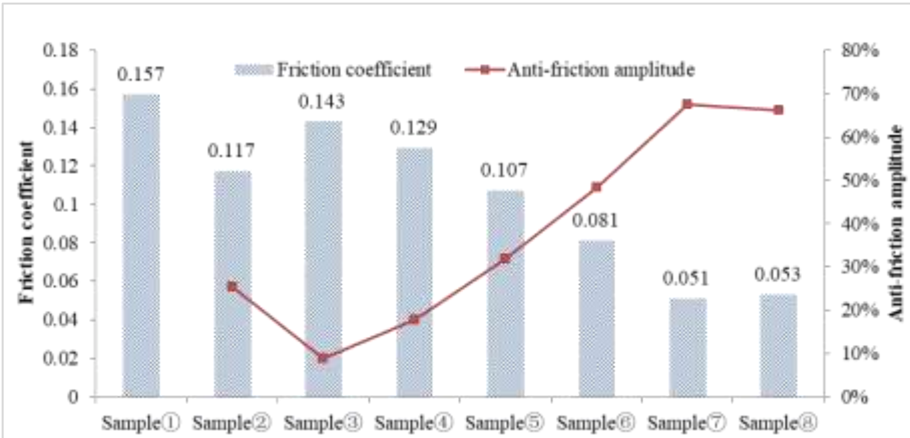

Fig 3: Piston ring-cylinder liner test results of oil samples

3.2 Engine Friction Torque Test Based on WLTC

3.2.1 Test preparation

WLTC was obtained based on the analysis of working conditions data in Europe, India, Japan, Korea and the United States [17]. WLTC can be divided into three grades and four speed curves according to the vehicle PMR (Power Mass Ratio) and the maximum speed. The PMR of the vehicle equipped with the turbocharged engine in this paper was greater than 34W/kg, and the maximum speed over 120km/h. The test cycle was WLTC Class 3B cycle It included 589s low speed segment, 433s medium speed segment, 455s high speed segment and 323s extra high speed segment, and the whole test cycle was 1800s.

In this paper, the engine speed and oil temperature data were collected in the WLTC fuel consumption test, the acquisition frequency was 0.5s, totaling 3600 points, combined with the WLTC vehicle speed curve, the results were shown in Figure 4.

The blue curve represented the change of engine speed. During the WLTC, the engine speed at most operating conditions was within the range of low and medium speed between 750rpm and 2500rpm. The engine speed was about 760rpm at the idle operating conditions. Under most operating conditions in the low, medium and high speed segments of WLTC, the engine speed was fluctuated at 1500rpm, and at some peak of vehicle speed, the engine speed could reach more than 2000rpm. However, under most working conditions in the extra high speed segment of WLTC, the engine speed exceeded 2000rpm, and the maximum speed could reach about 2700rpm.

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The red curve represented the oil temperature changes. In the first half of low speed segment, the oil temperature was at normal temperature 25℃ or so, during the second half slowly up to 40℃, and then from 40℃ quickly up to about 90℃ in the medium speed segment, from 90℃ slowly up to 100℃ in the first half of the high speed segment, maintained at about 100℃ in the second half of high speed segment and extra high speed segment.

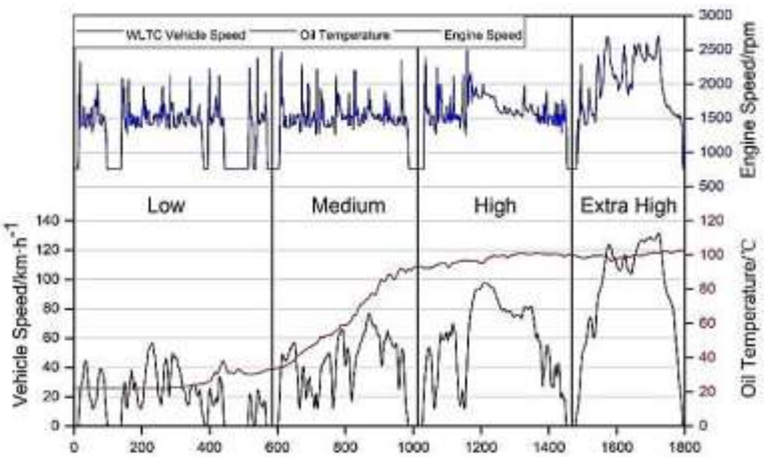

Fig 4: Engine speed and oil temperature of WLTC

According to the quantitative decomposition of 3600 data points, the engine speed was in an interval of 250 rpm, and the oil temperature was in an interval of 10℃. The intermediate point of the interval was recorded as shown in the TABLE V. The percentage of operating conditions higher than 1% was marked by green. The higher the percentage, the darker the green. The operating conditions with the percentage less than 1% were marked by yellow, and the operating conditions with the percentage of 0% were marked by red. Both the engine speed and the oil temperature were in a two-stage differentiation state. Although the engine speed of the WLTC cycle was between 750rpm and 2500rpm, but 48.4% of the operating conditions were concentrated at 1500rpm, 16.7% were concentrated at 1750rpm, and 14.3% were concentrated at 750rpm.

TABLE V. Percentage statistics of engine speed and oil temperature

	Engine Speed/rpm									
Total	750	1000	1250	1500	1750	2000	2250	2500		
21.3%	3.2%	0.3%	1.5%	13.1%	2.3%	0.8%	0.1%	0.0%	20	Oil

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12.2%	6.9%	0.3%	0.3%	2.8%	1.2%	0.3%	0.3%	0.1%	30	Tem p/℃
4.4%	0.0%	0.0%	0.9%	3.0%	0.4%	0.0%	0.1%	0.0%	40	
4.6%	0.0%	0.0%	0.6%	3.3%	0.4%	0.3%	0.0%	0.0%	50	
3.5%	0.0%	0.0%	0.2%	2.3%	0.9%	0.1%	0.0%	0.0%	60	
2.1%	0.0%	0.0%	0.0%	1.7%	0.3%	0.0%	0.1%	0.0%	70	
2.7%	0.0%	0.0%	0.0%	2.0%	0.6%	0.1%	0.0%	0.0%	80	
11.0%	2.6%	0.1%	0.4%	6.2%	1.2%	0.3%	0.2%	0.0%	90	
38.2%	1.6%	0.1%	0.5%	14.0%	9.4%	5.1%	2.6%	4.9%	100	
	14.3%	0.8%	4.4%	48.4%	16.7%	7.0%	3.4%	5.0%	Total	

The engine oil temperature mainly scattered under 40℃ and above 90℃, while 21.3% operating temperature at 20 ℃, and 12.2% at 30℃. In line with the WLTC low speed segment, while 38.2% and 11.0% operating temperature nearby 100℃ and 90℃ respectively, mainly related to the WLTC high speed and extra high speed segment.

Friction torque test bench was mainly composed of a motor and control device, torque meter, engine and oil temperature control system, as shown in TABLE VI and Figure 5.

TABLE VI. Basic information of test equipment

PARAMETER	EQUIPMENT	SPECIFICATION	ACCURACY
SPEED	AVL Dyno Train 220	220kW/3000rpm	±3rpm
TORQUE	T40B Torque Flange	0-100Nm	±0.01Nm
COOLANT TEMPERATURE	AVL 553 - C200	70-120℃	±1℃
OIL TEMPERATURE	ToCeil - LQY	10-130℃	±1℃

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Fig 5: Friction torque test bench

Before the engine torque friction test, dedicated preparation was required for the engine to keep same test initial condition, such as thorough cleaning and inspection, removal of the exhaust manifold and drilling a hole through the piston in order to oil temperature variation caused by the pumping and compression of the stroke, installation of the temperature control system in the oil pan to keep oil temperature stable. The basic information of the engine was shown in TABLE VII. In order to reduce the "carrying over effect" caused by residual oil from previous cycle, flushing by subsequent candidate oil was applied 3 times before each oil change. Then, the friction torque was measured from high temperature, and then gradually cooled to low temperature. The data were measured from high speed to low speed at the same temperature, in order to improve the efficiency of the test. The Sample① test was conducted at the beginning and the end of test sequence to check data deviation.

TABLE VII. Main technical parameters of engine

PARAMETER	**UNIT**	**SPECIFICATION**
ENGINE TYPE	——	L4
AIR INTAKE MODE	——	Turbocharging
OIL SUPPLY MODE	——	Multi-point Fuel Injection
TOTAL DISPLACEMENT	ml	1485
COMPRESSION RATIO	——	10.2:1
CYLINDER DIAMETER × PISTON STROKE	mm×mm	74.7×84.7
MAXIMUM TORQUE	Nm	230
MAXIMUM POWER	kW	110

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MAXIMUM POWER SPEED	rpm	5600
IDLE SPEED	rpm	750±50
OIL CAPACITY	L	4

In case of limited test resources, it was important to select the appropriate test operating conditions and give a reasonable WF (weighting factor) as far as possible. The calculation of WF was based on the premise that the operating conditions marked in dark green were taken as the test operating conditions as far as possible and combined with the adjacent light-colored operating conditions. The test parameters and each WF were shown in TABLE VIII.

TABLE VIII. FTT parameters and WF

POINT	TEMPERATURE/℃	SPEED/rpm	WF/%
1	20	750	3.5
2		1500	16.9
3		2250	0.9
4	30	750	7.2
5		1500	4.3
6		2250	0.8
7	50	1250	1.7
8		1500	8.6
9		1750	2.1
10	70	1500	3.7
11		1750	1.1
12	90	750	2.7
13		1500	7.8
14		2250	0.5
15	100	750	1.7
16		1500	23.9
17		2250	12.6

3.2.2 Analysis of engine friction torque test results

In this paper, the assessment of friction torque test results was the friction torque reduction rate of each candidate oil relative to the reference oil calculated as follows:

$$\text{FT Reduction Ratio}(\%) = \left[\frac{REO\ FT - Candidate\ FT}{REO\ FT}\right] \times 100 \qquad (1)$$

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The test results showed that the data deviation of reference oil was not more than 0.2%, which basically remains below 0.1%, indicating that the test state of the whole process was relatively stable and the data reliability was high. In figure 6(a), at the 20℃ oil temperature, the friction reduction ratio of different samples at 750rpm, 1500rpm and 2250rpm were 10.98% to 11.4%, 11.09% to 11.89% and 9.59% to 9.88% respectively, considering the test accuracy, five samples could be considered to have consistent friction reduction at a certain speed, while the same situation at 30℃ oil temperature, indicating both adsorptive fatty ester and reactive MoDTC could improve the lubrication state of the engine friction pair in figure 6(b).

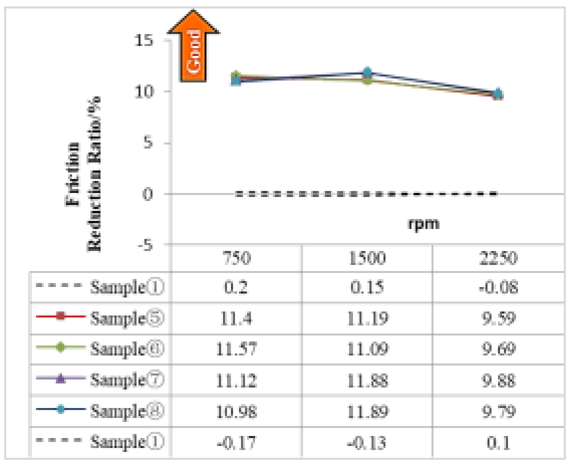

Fig 6(a): Friction reduction ratio at 20℃

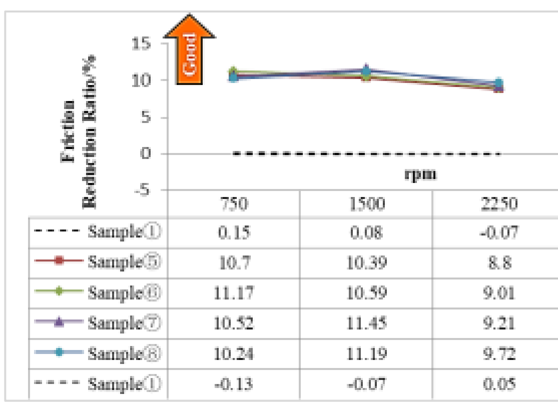

Fig 6(b): Friction reduction ratio at 30℃

With the gradual increase of oil temperature, the friction reduction ratio of the five samples at 1500rpm at 50℃ decreased by 8.59 to 9.02% in figure 6(c), and at 1500rpm at 70℃ by 5.12% to 6.15% in figure 6(d), indicating that the gradual decrease of viscosity aggravated engine

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wear, and the effect of anti-friction effect began to weaken.

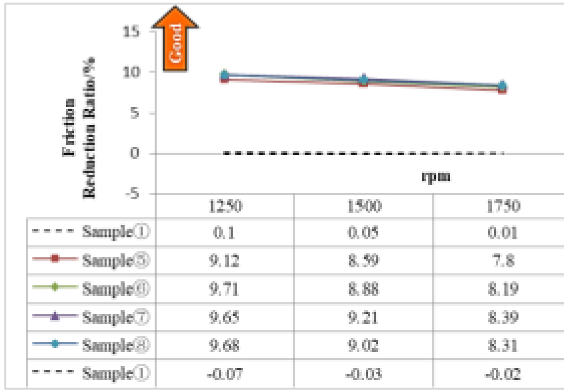

Fig 6(c): Friction reduction ratio at 50℃

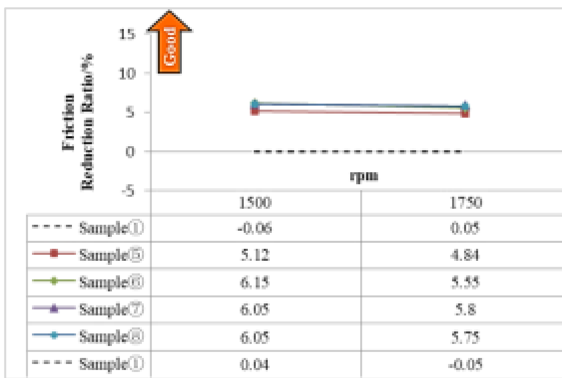

Fig 6(d): Friction reduction ratio at 70℃

At the same time, the differences between the samples began distinctive gradually, when the oil temperature at 90 ℃ in figure 6(e), the sample⑤ with fatty ester had showed the gap with other samples added up with MoDTC, and the gap became more obvious when the temperature reaches 100 ℃ in figure 6(f), the sample⑤ with fatty ester could not protect the engine effectively protect the engine friction, on the contrary, the sample⑥ with MoDTC can provide better protection, the combination of fatty ester and MoDTC could reduce the engine wear and tear more effectively.

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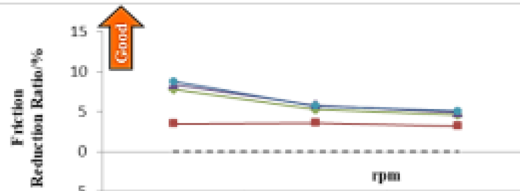
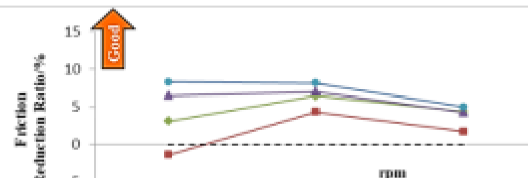


	750	1500	2250
Sample①	0.03	-0.02	0.01
Sample⑤	3.45	3.59	3.19
Sample⑥	7.75	5.22	4.54
Sample⑦	8.4	5.79	4.84
Sample⑧	8.8	5.69	5.04
Sample①	-0.02	0.01	-0.02

Fig 6(e): Friction reduction ratio at 90°C



	750	1500	2250
Sample①	0.03	-0.02	0.015
Sample⑤	-1.41	4.34	1.7
Sample⑥	3.11	6.42	4.37
Sample⑦	6.49	7.02	4.29
Sample⑧	8.3	8.13	5.02
Sample①	-0.01	0.02	-0.015

Fig 6(f): Friction reduction ratio at 100°C

Throughout all the tests, MoDTC demonstrated excellent anti-friction performance. Obviously, MoDTC began to play a friction reduction role at high temperature from 120°C [18], but it was obviously impossible to operate the WLTC test to reach that high oil temperature to achieve the friction reduction effect from MoDTC. This abnormal phenomenon can be explained that in the mixed lubrication state, the local contact point under low speed taxing would produce a very high contact temperature, facilitating the friction reduction effect of the MoDTC. Those were consistent with some scholars` low speed sliding temperature and friction research conclusions [19, 20].

3.3 Vehicle Fuel Consumption Test Based on WLTC

3.3.1 Test preparation

The vehicle fuel consumption test was conducted on five samples in accordance with the WLTC operating conditions in “Limits and Measurement Methods for Emissions from

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Light-Duty Vehicles (CHINA 6)" as shown in Figure 3 [21]. The information of the vehicle equipped with the turbocharged engine was shown in TABLE IX and the test was shown in Figure 7.

TABLE IX. Parameters of vehicle

PARAMETER	SPECIFICATION
CURB WEIGHT	1490kg
DRIVE	Front-wheel
WHEELBASE	2750mm
FRONT-WHEEL TRACK	1554mm
REAR-WHEEL TRACK	1549mm
TIRE	215/60 R17
MAXIMUM SPEED	170km/h

Fig 7: Vehicle fuel consumption test

The equipment for the vehicle fuel consumption test was relatively complex with uncertainty caused by human operation. The coupling error of the test results was qualified if it met the requirements of the national standard within 4%, which obviously failed to satisfy the requirements of the test in this paper. As a result, the consistency requirements on various controllable factors in the vehicle fuel consumption test process were carried out, based on a large number of previous test experience and data accumulated:

1) Before the test, the vehicle had a run in for more than 3000km to ensure that the state of

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the vehicle parts was stable. At the same time, the vehicle state was thoroughly checked to ensure that there was no fault or abnormality;

2) The immersion time was adjusted to 24±1 hours to ensure the consistency of the vehicle and the operator;

3) Before each test, the battery voltage was controlled within the range of 12.3±0.1V;

4) The initial oil temperature should be controlled within 24±1℃ before each test;

5) Before each test, the tire pressure was precisely controlled at 230kPa, and the change of rolling resistance coefficient caused by wear was avoided as far as possible;

6) The driver was required to be the same person throughout the test process;

7) The test equipment should be calibrated before each test;

8) Three effective test results (standard deviation < 0.05) should be guaranteed for each test oil sample, and a reasonable test process should be developed according to the stability of data;

9) When replacing the test oil sample, the flushing process should be consistent with the requirements of the friction torque test, and three times of full flushing should be carried out;

10) The overall analysis of the test data was carried out in combination with mean value processing and standard deviation, and the deviation of the reference oil was fully considered.

3.3.2 Analysis of vehicle fuel consumption test results

The evaluation of vehicle fuel consumption test results was to test the fuel economy improvement rate of candidate oil to reference oil calculated as follows:

$$\text{FEI Ratio}(\%) = \left[\frac{REO\ FC - Candidate\ FC}{REO\ FC}\right] \times 100 \tag{2}$$

The vehicle fuel consumption test results and FEI (Fuel Economy Improvement) of five samples were shown in Figure 8. The test data showed that the test accuracy was well controlled, and the test results of baseline sample① were 7.85L/100Km, 7.82 L/100Km and 7.84 L/100Km, the standard deviation was 0.012 indicating that the test state of the whole process was relatively stable and the data reliability was high as well. FEI of the other four samples were 0.88%,1.32%,1.89% and 2.01% respectively.

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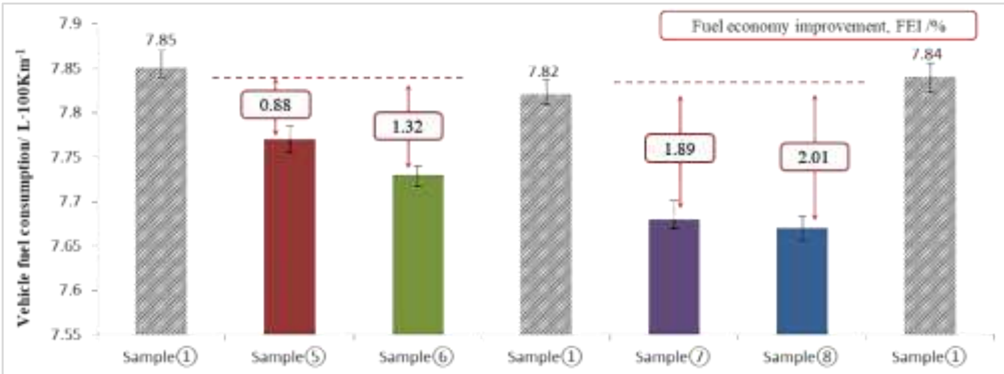

Fig 8: WLTC fuel consumption and FEI results of samples

IV.ENGINE RELIABILITY TEST

In order to ensure the applicability of energy-saving low friction oil, in addition to achieving the purpose of energy saving, it was also necessary to ensure that the oil does not affect the reliability performance of the engine. Through the above-mentioned friction energy saving tests, it had been fully proved that the oil sample⑧ with fatty acid ester and MoDTC compound friction improver had excellent energy saving effect.

On this basis, GED test according to GMW 3396TP test standard and GETC test according to GMW 3396TP test standard in the development process of General Motors engine parts were carried out to verify the influence of oil sample⑧ on engine reliability performance.

4.1 GED Durability Test

The total duration of durability test was 703 hours about 250,000 kilometers for the vehicle in this paper, and with 1 hour for each test cycle and 703 cycles in total. After every 100 hours test, oil change was made and the engine accessory status was checked. The engine coolant temperature was maintained in the range of 95℃±5℃, and the air temperature after intercooling in the range of 30℃~35℃. The fuel temperature and fuel pressure were controlled at 25±5℃ and 380±10kPa. After the test, the friction pairs were dismantled to evaluate the wear state.

4.2 Performance Comparison

The torque and power of the engine were measured before and after the GED test, as shown in Figure 9. After the test, the torque and power of the engine decreased by about 6% from

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3600rpm at the high speed stage, indicating that the engine was worn but still within the range of performance requirements.

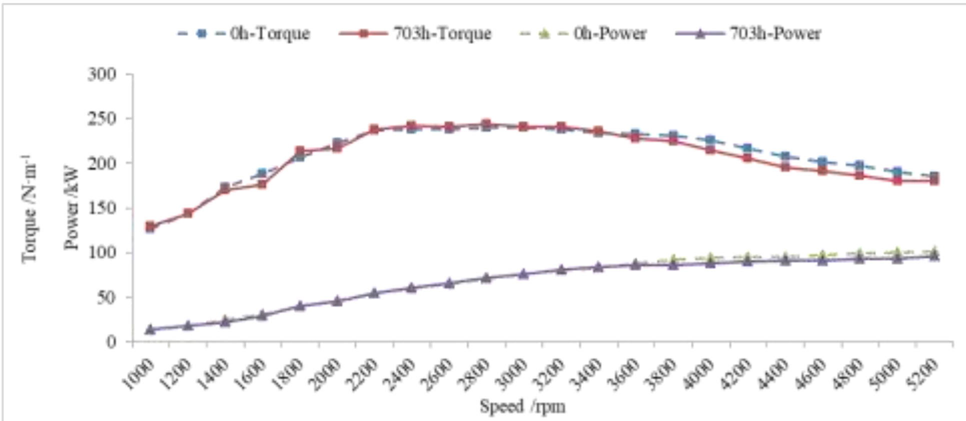

Fig 9: Performance comparison before and after GED test

4.3 Piston Leakage

During GED test, after every 100 cycles, piston leakage should be tested every 1000r/min at 100kPa air pressure and under full engine load condition to observe whether there is abnormal leakage. The test results were shown in Figure 10. During the whole testing process, there was no abnormal change in piston channeling leakage, which was within 40L/min as required by the performance design.

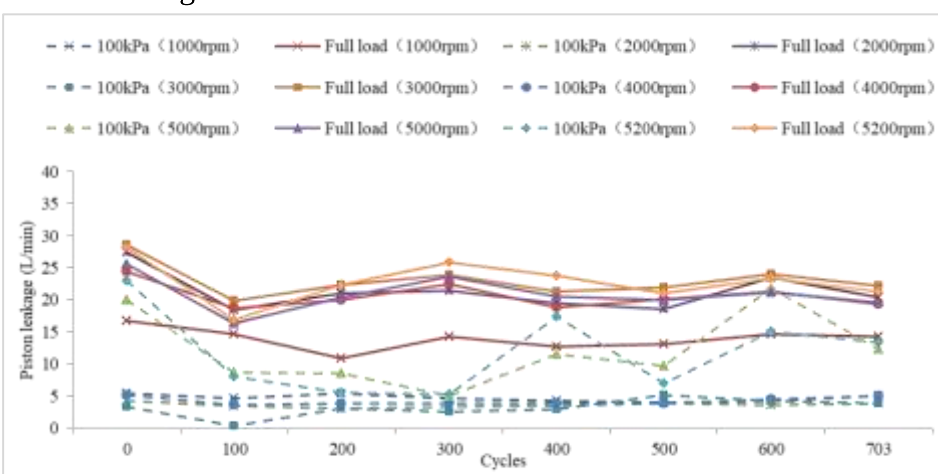

Fig 10: Piston leakage comparison during GED test

4.4 Oil Consumption

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The wear state between the engine piston ring and cylinder can be determined by the engine oil consumption rate. Every 100 hours of the durability test, the lubricating oil was replaced and the oil consumption rate was determined by the oil discharge weighing method. The results were shown in Figure 11.

There was no significant change in oil consumption during the whole test process, indicating that there was no significant wear between the piston ring and the cylinder wall, which met the performance requirements of less than 35g/h after durability test.

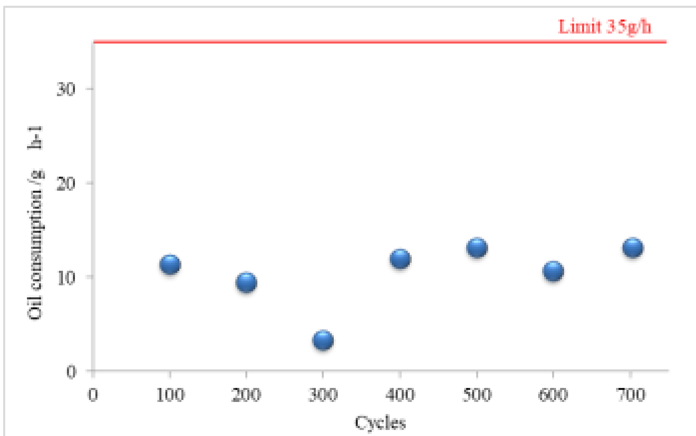

Fig 11: Oil consumption comparison during GED test

4.5 Disassembly of Engine Friction Pair

After the completion of durability test, the test engine was disassembled to observe the wear status of the main friction pairs and the state of the main vulnerable parts, such as spark plugs, thermostats and supercharger etc. The state of each component was rated to evaluate the durability test results.

Figure 12 showed the wear status of the three major friction pairs of the engine after endurance tests. There was an amount of carbon deposit on the piston skirt. The surface of the crankshaft connecting rod neck and the main journal was smooth and clean. No obvious scratch on camshaft surface, and no obvious wear on cylinder liner. Generally speaking, no obvious wear on the main friction pairs after durability test proved that Sample⑧ can meet the requirements of engine wear performance.

Fig 12(a): Piston skirt

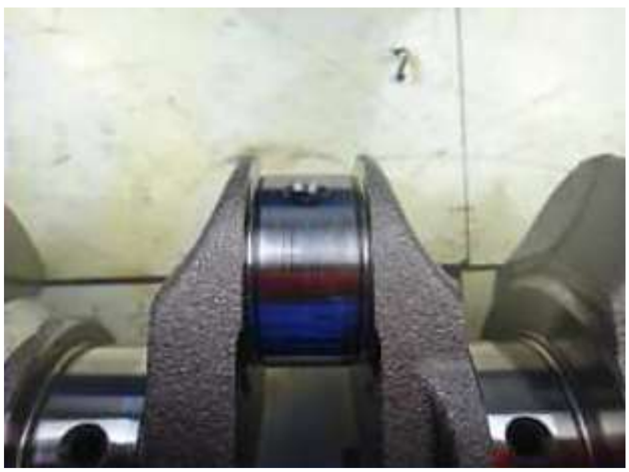

Fig 12(b): The main journal of crankshaft

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Fig 12(c): Camshaft surface

Fig 12(d): Cylinder liner

4.6 GETC Test

GETC test is used to evaluate the thermal characteristics of all engine parts, especially for evaluating the thermal characteristics of cylinder head, cylinder pad and exhaust manifold. The test consists of 8600 cycles, each consisting of a heating and cooling section, and the cycle time is determined by the time required to heat and cool to the specified temperature, which is generally 7-8 minutes. The fuel temperature was controlled within 25±5℃ and the fuel pressure was controlled within 380±10kPa during the whole test process. After the test, the friction pair should be disassembled to evaluate its performance state.

4.7 Performance Comparison

The torque and power of the engine were measured before and after the GETC test, as shown in Figure 13. The torque and power of the engine were basically consistent before and after the test, indicating that the engine performance was not significantly affected.

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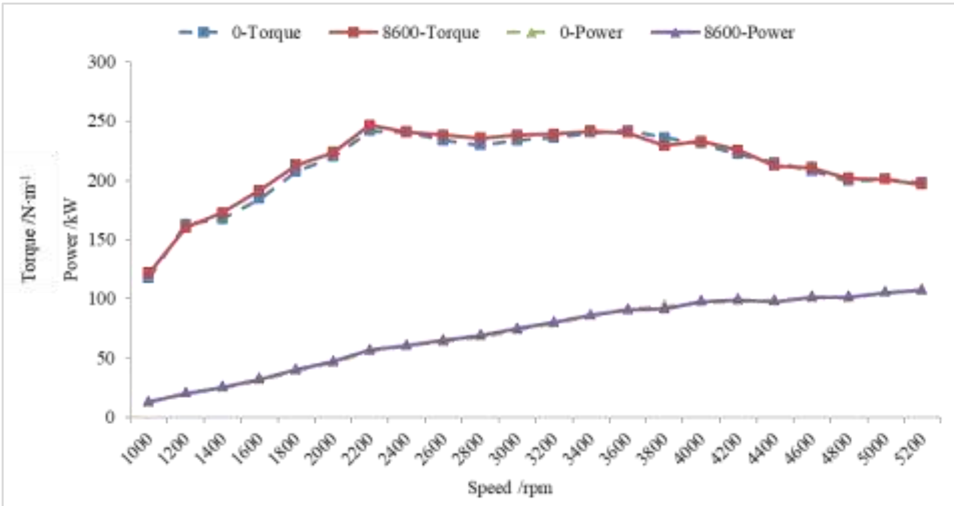

Fig 13: Performance comparison before and after GETC test

4.8 Piston Leakage

During GETC test, after every 1000 cycles, piston leakage should be tested every 1000r/min at 100kPa air pressure and under full engine load condition to observe whether there is abnormal leakage. The test results were shown in Figure 14. During the whole testing process, there was no abnormal change in piston channeling leakage to meet the performance requirements.

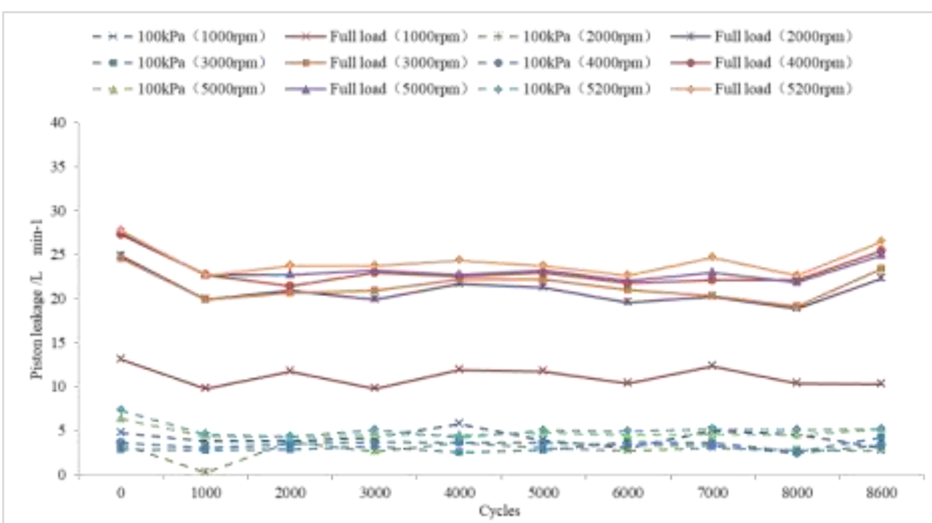

Fig 14: Piston leakage comparison during GETC test

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4.9 Oil Consumption

During the GETC test, the oil was changed every 500 cycles and the oil consumption rate was determined according to the oil dumping weighing method. The results were shown in Figure 15.

There was no abnormal change in the oil consumption during the whole test process, which met the performance design requirements.

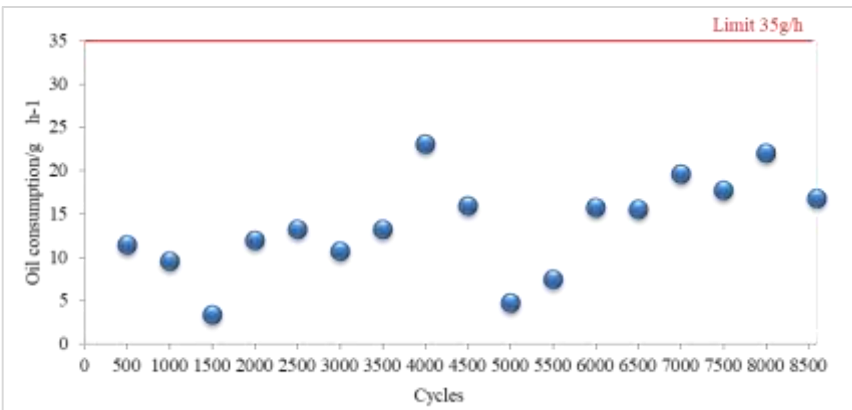

Fig 15: Oil consumption comparison during GETC test

4.10 Disassembly of Engine Friction Pair

After the GETC test was completed, the test engine was disassembled to observe whether the performance state of the engine parts was normal. Figure 16 shows the performance state of the engine's three main friction pairs after GETC test. Due to the harsh working conditions of GETC test, the carbon accumulation of piston top and camshaft and other parts is more serious than GED test, but the overall wear condition was good, and the performance state of all engine parts is in normal state. It is proved that the low friction oil sample⑧ can also meet the thermal cycle requirements of the engine parts.

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Fig 16(a): The top surface of piston

Fig 16(b): Exhaust camshaft

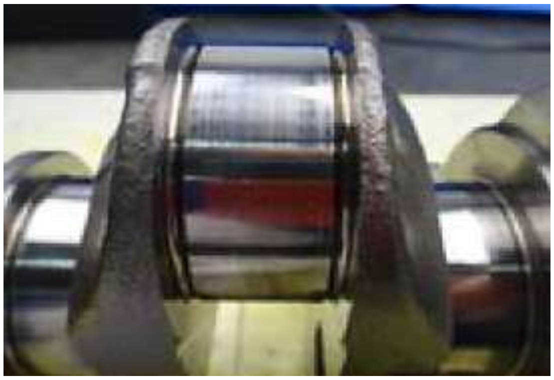

Fig 16(c): The main journal of crankshaft

Fig 16(d): Cylinder liner

To sum up, the energy saving low friction oil sample ⑧ has excellent fuel saving effect on the premise of ensuring the durability and reliability of the engine, indicating that the oil sample has engineering application value in this small turbocharged engine.

V. CONCLUSION

(1) On the basis of the existing oil formula of turbocharged engine, energy-saving low friction oil formula can be made with appropriate dosage and appropriate type of friction improvers, which has minor differences in the physical and chemical properties of the formula compared with the current oil, but can improve the oil friction reduction effect and a certain extreme pressure anti-wear effect.

(2) Piston ring-cylinder liner friction test and engine friction torque test results showed that adding friction improver energy-saving low-friction oil can effectively reduce the friction coefficient of friction pair engine. By adding fatty acid ester and MoDTC compound with anti-friction sample today can achieve an engine friction torque to reduce the rate of 8.6%, achieving better anti-friction effect. The fuel consumption test results showed that the fuel saving effect of adding two friction improvers can reach 2.01%.

(3) GED durability test was carried out on the sample⑧ as well as the GETC test results showed that in the whole test process, there was no abnormal in torque, power, piston leakage and oil consumption. Engine parts and performance status were all normal after dismantling, indicating the energy-saving low friction oil sample⑧ had engineering applicability.

ADDITIONAL INFORMATION

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The full name of acronyms in this paper were shown in the TABLE X as follows:

TABLE X. The full name of acronyms

NO.	ACRONYM	FULL NAME
1.	WLTC	Worldwide harmonized Light vehicles Test Cycle
2.	MoDTC	Molybdenum Dithiocarbamate
3.	GED	Global Engine Durability Test
4.	GETC	Global Engine Thermal Cycle
5.	OEM	Original Equipment Manufacturer
6.	GMW	GM Worldwide Engineering Standard
7.	HFRR	High Frequency Reciprocating Rig
8.	MTM	Mini Traction Machine
9.	KV	Kinematic Viscosity
10.	TBN	Total Base Number
11.	WF	Weight Factor
12.	FT	Friction Torque
13.	REO	Reference Oil
14.	FEI	Fuel Economy Improvement
15.	FC	Fuel Consumption

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