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Tuning for a Small Engine with a Supercharger

Shinji Kajiwara, Tadamasa Fukuoka

***Abstract*—**The formula project of Kinki University has been involved in the student Formula SAE of Japan (JSAE) since the second year the competition was held. The vehicle developed in the project uses a ZX-6R engine, which has been manufactured by Kawasaki Heavy Industries for the JSAE competition for the eighth time. The limited performance of the concept vehicle was improved through the development of a power train. The supercharger loading, engine dry sump, and engine cooling management of the vehicle were also enhanced. The supercharger loading enabled the vehicle to achieve a maximum output of 59.6 kW (80.6 PS)/9000 rpm and a maximum torque of 70.6 Nm (7.2 kgf m)/8000 rpm. We successfully achieved 90% of the engine's torque band (4000–10000 rpm) with 50% of the revolutions in regular engine use (2000–12000 rpm). Using a dry sump system, we periodically managed hydraulic pressure during engine operation. A system that controls engine stoppage when hydraulic pressure falls was also constructed. Using the dry sump system at 80 mm reduced the required engine load and the vehicle's center of gravity. Even when engine motion was suspended by the electromotive force exerted by the water pump, the circulation of cooling water was still possible. These findings enabled us to create a cooling system in accordance with the requirements of the competition.

***Keywords*—**Engine, combustion, cooling system, dry sump system, numerical simulation, power, torque, mechanical supercharger.

I. Introduction

KINKI University Formula Project has been participating in the student Formula SAE of Japan (JSAE) annually since the second year of the competition. Each year, the university implements various vehicle improvements, including the deployment of a turbocharger and a supercharger and the fabrication of a dry sump system. The active team members of the project are 5–15 students, who are all undergraduates. This paper discusses the engine development activities that have been undertaken for Kinki University's formula project since the institution's participation in the JSAE. Kinki University's formula machine is shown in Fig. 1.

Spark ignition (SI) high-performance engines for use in Formula SAE/student competitions have been developed and optimized [1]. The performance of a small engine, which has been tested in a variety of normally aspirated and forced induction modes on 98-RON pump gasoline, was compared [2]. The modes examined were (1) NA with carburetion, (2) NA with port fuel injection (PFI), (3) mild supercharging (SC) with PFI, and (4) high turbocharging (TC) with PFI. The results are significant in defining the limitations of small downsized SI engines, and power increases via intake boosting are needed to compensate for reduced swept volume. The experimental findings show that BSFC values in the order of 240 g/kWh or 34% could be achieved. We also achieved TC BMEP values in the region of 25 bar— the highest recorded for small engines running on pump gasoline.

Many variations in intake designs are currently implemented in Formula SAE. This study investigates whether one intake design exhibits performance that is superior to another and obtains insight into the nature of the airflow within a Formula SAE intake system [3]. To obtain the maximum range for usable torque, Helmholtz theory is used as basis for engine tuning [4]. A prototype engine intake design that incorporates a required 20-mm restrictor is currently being evaluated [5]. A distributed system for engine management is expected to comprise a comprehensive, predictable, and easily extendable platform, thereby presenting the possibility of incorporating additional features into an engine even at the racetrack [6].

Fig. 1 Overview of Kinki University's formula machine

II. Equipping the Vehicle with a Supercharger

We think that the power–weight ratio is the most important value in evaluating the performance of a power train during student formula event. We therefore deployed a supercharger, which can markedly improve the power–weight ratio of a vehicle. The example of autocross course layout is shown in Fig. 2. As for race course layout (autocross, endurance), the existence of numerous tight corners determines the deployment of a mechanical supercharger, which executes efficient SC even at low turns because the low rotary levels produced by engine revolutions account for 70% of the entire revolutions. The result of the chassis dynamo in NA is shown in Fig. 3. Because the 20-mm diameter air restrictor exerted considerable influence on engine operation, the maximum power generated was 37.5 kW (50 PS)/10000 rpm, the maximum torque achieved was 50 Nm/9000 rpm (5.1 kgf m), and 80% torque band was produced at 8500–10000 rpm. The result of the chassis dynamo equipped with a mechanical supercharger and

Shinji Kajiwara is with the Department of Mechanical Engineering, Kinki University, 3-4-1, Kowakae, Higashiosaka, Osaka, 577-8502, Japan (phone: +81-6-6730-5880 (ext. 5366); fax: +81-6-6727-2024; e-mail: kajiwara@mech.kindai.ac.jp).

electric oil pump is illustrated in Fig. 4. This configuration generated a maximum power of 59.6 kW (80.9 PS)/9000rpm, a maximum torque of 70.6 Nm (7.2 kgf m)/8000rpm, and 90% torque band at 4000–10000 rpm. SC pressure was set to 0.2 bar (200 kPa).

By adopting the supercharger, we increased the torque band to approximately twice that of NA. Comparison of torque curve is shown in Fig. 5. The torque band was approximately 60%, which accounts for 80% of the torque achieved with the typical number of revolutions (4000–12000 rpm). For NA, the power–weight ratio was 5.6 PS/kg, which was substantially improved to 3.8 PS/kg by the supercharger. We were also able to run approximately 75% of an entire course with an 80% torque band; this percentage is higher than the run data obtained in the autocross assumption course.

The power axis of the supercharger extended the oil pump axis of the ZX-6R engine. Super charger output power shaft is shown in Fig. 6. The initiation of supercharger deployment necessitated a large moment for a bearing because of the extended oil pump axis, and the power axis was damaged. We reduced the moment on the basis of a review of the bearing's dimension clearance, which markedly increased the service life of the axial component. The CAD model of engine compartment is shown in Fig. 7. Specification of the formula machine is shown in Table I.

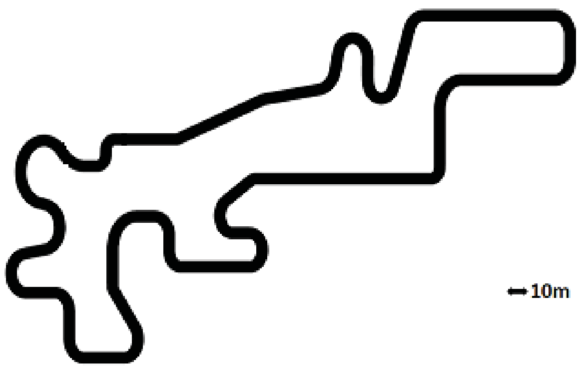

Fig. 2 Layout of the endurance course

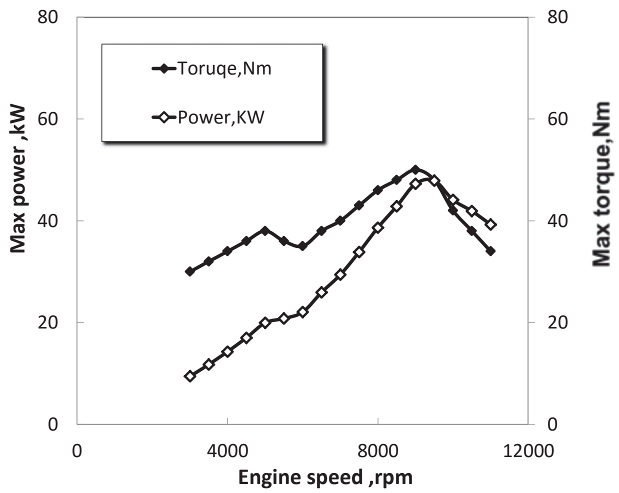

Fig. 3 Output power and torque in rear wheels of NA engine

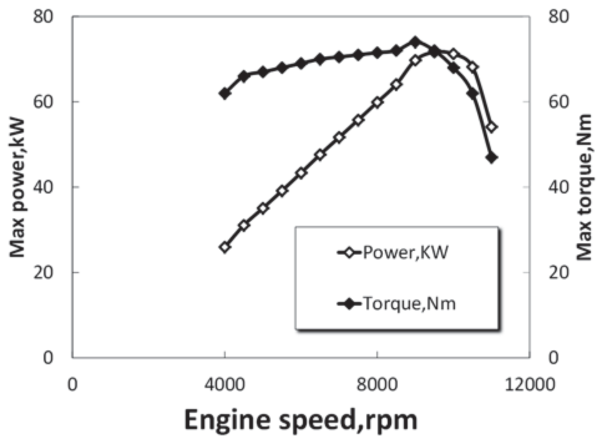

Fig. 4 Output power and torque in rear wheels of S/C and the electric oil pump engine

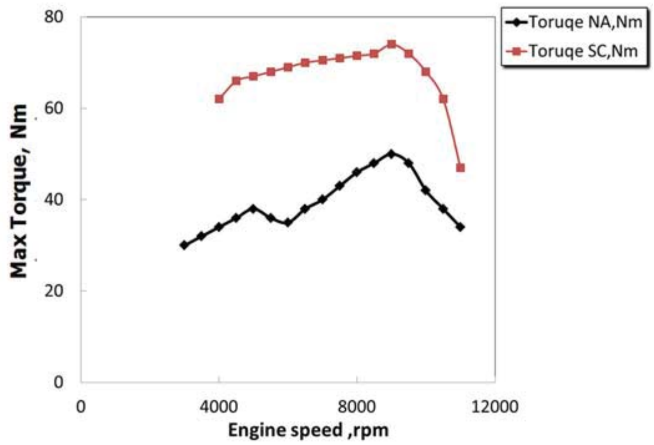

Fig. 5 Comparison of the output torque

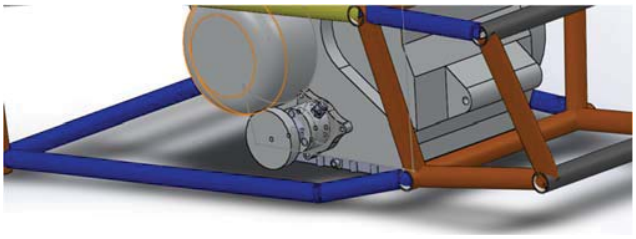

Fig. 6 Super charger output power shaft

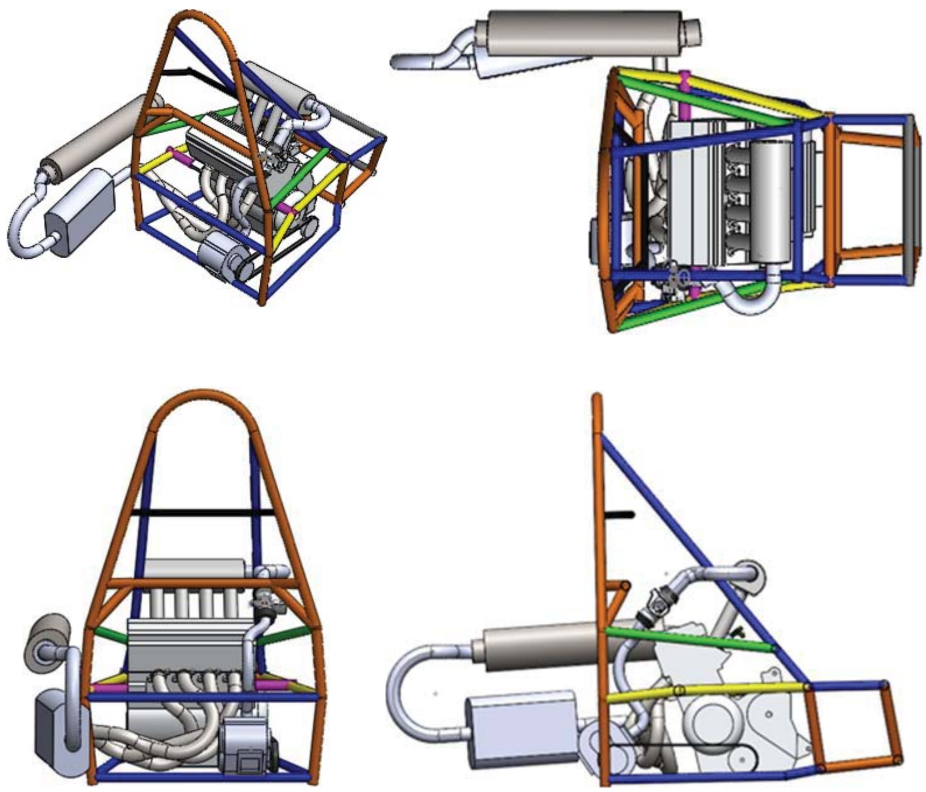

Fig. 7 CAD model of engine compartment

TABLE I
SPECIFICATION OF JSAE MACHINE

Dimensions	Front	Rear
Overall Length, Width, Height	2780mm, 1400mm, 1140mm	
Wheelbase	1600mm	
Track	1200mm	1200mm
Suspension Parameters	**Front**	**Rear**
Suspension Type	Double unequal length A-arm. Pull rod actuated spring and damper.	Double unequal length A-arm. Pull rod actuated spring and damper.
Tire Size and Compound Type	Hoosier R25B	Hoosier R25B
Wheels	6 inch wide, 1 pc Al Rim, 1.5 inch pos.offset, RAYS	6 inch wide, 1 pc Al Rim, 1.5 inch pos.offset, RAYS
Body Frame		
Frame Construction	Steel Tubular Space Frame	
Material	STKM 13A, STKM 11A steel round tubing.	
Powertrain		
Manufacturer/Model	Kawasaki ZX-6R 2010/ ZX600PE	
Bore/Stroke/Cylinders/Displacement	67mm bore / 42.5mm stroke / 4 cylinder / 599cc	
Compression Ratio	13.3:1	
Compressor / Manufacturer Model	Mechanical Supercherger / Aisin AMR500	
Compressor Specification	500cc / rev with 2 lobe roters	

III. PREVENTING KNOCKING BY USING AN EXHAUST LAYOUT

A supercharged engine often suffers from knocking problems. To determine the exhaust system layout, the pressure change around the intake and exhaust valves of the engine system was analyzed using the BOOST 1D Cycle Simulation CAE tool (AVL Inc.). We adopted the exhaust layout. Simulating supercharged engines is difficult because of the extensive influence of intake pressure, exhaust gas temperature, and other similar variables. Given that our purpose was to ascertain the exhaust layout, a 1D simulation of natural aspiration, and not SC, was performed. The pressure change around the intake and exhaust valves is shown in Fig. 8. Because of the exhaust internal pressure exerted by the opening of the exhaust valve and the overlap caused by the opening of the inlet valve, gas was siphoned by the pulsation of a blowdown, and the scavenging effect improved. When the internal pressure of the intake pipe was overlapped, the efficient air intake for which air intake pulsation was utilized served as positive pressure. This phenomenon indicates that converting the internal pressure of a cylinder into a negative value in the overlap period prevents knocking. When the diameter and length of exhaust pipes were optimized, the suction effect of exhaust pulsation caused the residual gas to reduce the residual gas in the cylinder. This finding demonstrates that advancing ignition timing is possible.

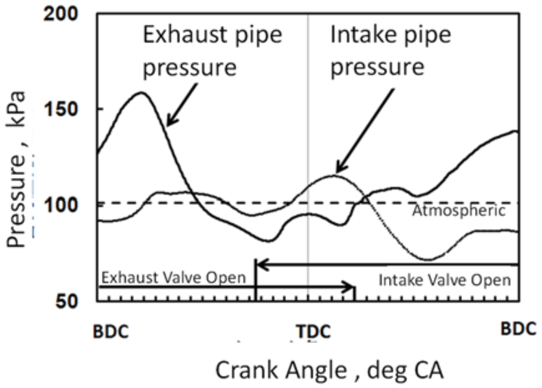

Fig. 8 Change of pressure of inlet-exhaust pipe

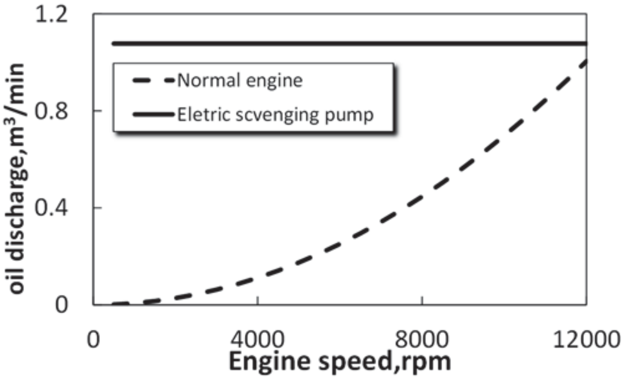

Fig. 9 Relationship between oil discharge rate and engine speed

IV. Dry Sump of the Engine

A low center of gravity is regarded as a significant technological element of a vehicle. The JSAE continues to use courses with autocross, endurance, and tight high-speed corners. During a run, therefore, a vehicle must always contend with turning lateral acceleration, necessitating control over the oil deflection caused by the aforementioned acceleration. On the basis of these those observations, we changed the engine's lubrication mechanism from a wet sump to a dry sump. The dry sump prolonged lubrication and caused greater pressure loss than that caused by the wet sump. Accordingly, we re-estimated the amount of loss in the lubrication and used a different scavenging pump.

First, an actual scavenging pump rate was necessary. The volume discharge to a scavenging pump took 1 minute at *V*,

$$V = n \cdot V' \tag{1}$$

V: Quantity of discharge volume, m^3/min; *n*: Engine speed, RPM; *V'*: One discharge volume, m^3.

When demand speed from quantity of discharge volume *v*;

$$v = \frac{V}{60 \cdot A} \tag{2}$$

v: Flow velocity, m/s; *A*: Cross section, m^2.

The oil pressure in the main gallery was determined using Bernoulli's theorem, which is expressed as;

$$P = \frac{1}{2}\rho v^2 - \rho g h - \Delta h \tag{3}$$

P: Oil pressure, Pa; *h*: Pitch difference, m; ρ: Density, kg/m^3; *g*: Acceleration of gravity, m/s^2; Δh: Pressure loss, Pa.

The discharge volume was expected to be greater than that achieved with a typical scavenging pump. On the basis of the calculations, we adopted an electric scavenging pump that can exert an oil pressure of about 14,000 rpm. The oil discharge rate during engine running is shown in Fig. 9.

V. Cooling Management of the Engine

Heat capacity considerably increases with technological advancements in engine design. Correspondingly, we extensively modified the cooling system of our vehicle. With respect to the deployment of the new radiator, the water pump was motorized by the temperature control of the radiator fan. Applying these improvements to the supercharger caused difficulties in maintaining a constant water temperature in an engine with an actual radiator. We therefore designed a new radiator by using a heat transfer expression.

$$Q = K \cdot A \cdot \Delta t \tag{4}$$

Q: Heat release rate, W; *K*: Thermal conductivity, W/m^2 K; *A*: All radiator surface area, m^2; Δt: Mean temperature difference of water and air, K.

Because we overestimated the heat transfer rate of a new radiator, cooling performance constituted a key concern. Nevertheless, stable cooling performance was maintained in the deployment of the engine during the second year of the competition. Such performance was achieved by changing the value of the heat transfer rate under driving conditions for the new radiator with a motorized water pump.

A student formula engine must be stopped during a run if a substantial difference in temperature arises in the mechanical water pump in cases where engine shutdown occurs between changes in the internal engine's temperature and the temperature measurement point of the ECU. An electric water pump was used because conclusive data on engine start characteristics were unavailable. In a cooling course, water can be circulated using an electric water pump even if an engine stops. This enables the reduction of the difference between the internal engine temperature and the temperature measurement point of the ECU. The results show that engine start characteristics were achieved.

The thermostat was removed to secure the quantity of circulated water. The starting and ending temperatures (85 and 75 °C, respectively) of the radiator fan were established using the ECU. The water temperature after the engine was started is shown in Fig. 10. Thereby, the water temperature was stabilized at 75–85 °C. The electric water pump also partly contributed to the water temperature.

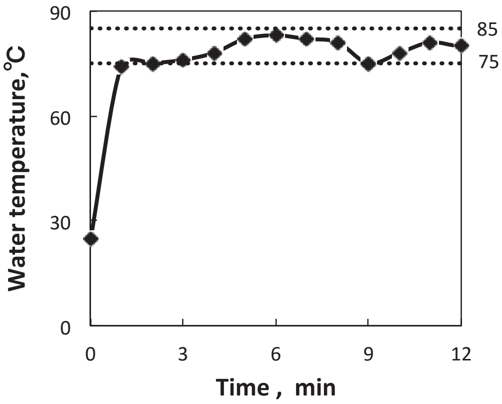

Fig. 10 Water temperature after engine started

VI. Summary3D/Conclusions

1) A maximum power of 59.6 kW (80.6 PS) /9000 rpm and a maximum torque of 70.6 Nm (7.2 kgfm) /8000 rpm were achieved by adding a supercharger.
2) Engine mounting position was lowered by 80 mm using a dry sump system.
3) An ECU-controlled radiator fan was sufficient to stabilize the water temperature at around 75-85°C.

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