論文　**Original Paper**

An Investigation on the Design Method and Manufacturing of Powertrain System for Student Formula Japan (SFJ) Vehicle ―Design Method of Powertrain System of Gasoline Engine with Turbocharger for Student Formula Japan (SFJ) Vehicle―

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Abstract: Student Formula Japan is one of the competitions of the manufacturing education for students, which is held by SAE (Society of Automotive Engineers). The competition consists of the static events and the dynamic events. The static events have three competitions of presentation, cost and design, and the dynamic events are acceleration, skid pad, autocross and endurance with fuel consumption. To get the higher rank in this competition, we must win the dynamic events with the high allotment points. For the purpose of the winning, we should have some advantages more than the other universities. We aimed at raising engine power and drivability. There are some ways to get engine power up such as the combustion improvement, the change of the cam profile and supercharging. We tried to install a commercial turbocharger to a normal aspirated multi-cylinder engine for Student Formula Japan under the SAE regulations. As the throttle and air restrictor must be located at upper streamside of the intake system, the piping from the exhaust manifold to the turbocharger is designed in order not to flow fuel into the turbocharger. Furthermore, we redesigned the intake air collector and the air restrictor in consideration of installation of the turbocharger to the standard engine. We considered sufficiently the results of the simulations using VES (Virtual Engine Simulation) and CFD (Computational Fluid Dynamics) simulation methods and verified them by the results of the experiments. As a result, we succeeded getting the higher engine performance by turbocharging than the normal engine, and the turbocharged engine can perform the higher and flat torque in the lower engine speed range, 5000 [r/min] to 8000 [r/min]. We made a Student Formula Japan vehicle that everyone can drive fast.

Key words: Power Unit, Engine Component / Student Formula Japan, Intake and Exhaust System, Turbocharged Engine, Computational Fluid Dynamics, Virtual Engine Simulation, Engine Performance, Turbocharger, Air Restrictor

1. Introduction

As for the Student Formula Japan (hereafter called "SFJ") competitions, the regulation was supposed to obligate to manufacture one vehicle in a year [01]–[08]. In other words, the planning, the development, the design, the manufacturing and the experiment must be finished in this period [9]–[16]. The team of our university has been attending two competitions; U.S.A. competition in May and Japan competition in September for these four years, since the first Japan combined team had participated in the SFJ competition 2000 year in U.S.A.. Therefore, our team has to finish all the works to manufacture a new vehicle in 5 months from October to next February [01]–[16].

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2. Flow of This Study

This study described the improvement of engine performance better than last year's engine performance at least, securing the durability, drivability and reliability to finish all the dynamic events of the competition and to exceed score in the previous year. We tried to install a commercial and typical turbo charger with a normal aspirated engine in order to get a higher engine performance.

For the purpose of this study, we performed the processes as follows,

[1] The problem and the improvement points, which are related with making a turbocharged engine based on a normal aspirated engine had to be found out by the implementation of the engine test to have simply incorporated a turbocharger into intake and exhaust system of the base engine.

[2] A turbocharger and an intake collector volume were selected by the experiment results. A collector shape and an air restrictor shape were designed using the CFD (Computational Fluid Dynamics) analysis [17], [18].

[3] The model for VES (Virtual Engine Simulation) was built by adjustment of the VES engine simulation results to the experiment results, according to the laying-out of intake and an exhaust system we should design [19]–[22]. The base specifications of a turbocharged engine for SFJ vehicle were determined through the numerical calculations of VES using the model [23]–[33], changing each engine element related with engine power.

[4] After decision of the engine specifications, turbocharged engine bench mark test were carried out in order to examine the engine performance as the computation results.

3. Numerical Consideration of Turbocharger Built-in

A turbocharger is a kind of air pump that reuses the energy of the exhaust gas that is emitted, being generally wasteful. The exhaust gas drives turbine rotor assembly (the hot side) at first. This assembly is connected to the compressor wheel (the cold side) one another. When the compressor wheel and the turbine rotor assembly are turned by exhaust gas, a great deal of compressed air is sent into the combustion chamber of the engine. That is, as a great deal of compressed air makes more fuel sent into a combustion chamber with air, the turbocharged engines can generate higher output than normal aspirated engines if compared with same displacement volume engine. On the other side, higher fuel consumption and brake powers of a turbocharged engine causes some problems such as higher cooling temperature and reliability. Furthermore, the regulation of SFJ describes on turbochargers and superchargers as follows [34]; "Turbochargers or superchargers are allowed if the competition team designs the application. Engines that have been designed for and originally equipped with a turbocharger are not allowed to compete with the turbo installed. The restrictor must be placed upstream of the compressor but after the carburetor or throttle valve. Thus, the only sequence allowed is throttle, restrictor, compressor, engine. Only ambient air may be used to cool an intercooler for engine oil". Refer to appendix I.

3.1 Sections and Layout of a Turbocharger

Presupposing that the turbo can be started up by the low boost-pressure power in the low engine speed range, we selected two kinds of small sized turbochargers with the smallest A/R (Area of air inlet part / Radius of turbine) made by IHI as shown in **Figure 1**, **Figure 2** and **Figure 3** shows the results of the engine bench tests with each test turbocharger in maximum boost pressure 160 [kPa]. However, because the maximum boost pressure of type A rose only to 150 [kPa], the difference with the maximum torque shown in this Figure was caused by boost pressure difference. **Figure 4** shows the experimental results of the internal pressure rising time of intake pipe. This figure also shows the rise change of the intake pipe pressure from engine operation condition maintained at intake pipe pressure 100 [kPa], 4000 [r/min] to the full load condition in the moment (time which makes full load is 0.2 [seconds]). In other words, these experimental results

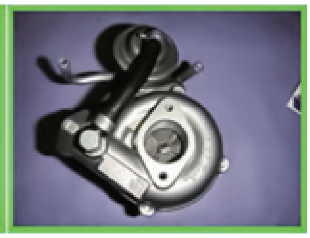


Figure 1 Test Turbocharger, IHI-Type A [A/R:7.0], Area/Radius=7.0 of Turbine.

Figure 2 Test Turbocharger, IHI-Type B [A/R:9.0], Area/Radius=9.0 of Turbine.

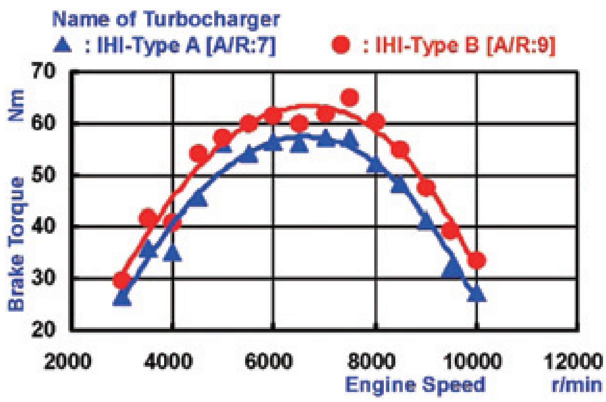


Figure 3 Engine Performance Difference Test Gasoline Engine with Turbocharger [Comparison of IHI-Type A [A/R:7.0] and IHI-Type B [A/R:9.0] Turbocharger].

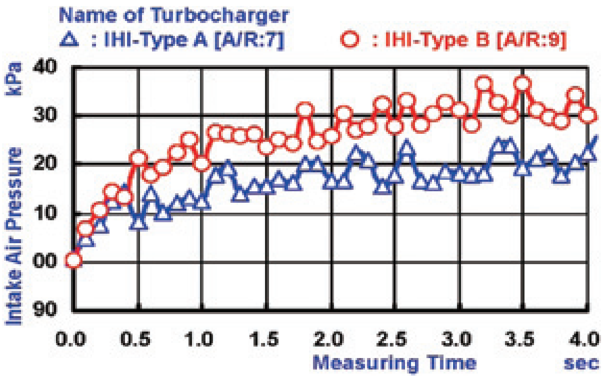


Figure 4 Response Difference between IHI-Type A [A/R:7.0] and IHI-Type B [A/R:9.0] Turbocharger at Engine Speed 4000 [r/min].

meant the response and the turbocharged performance of the test turbocharger. As the response of type B was superior to type A, type B can be said the turbo for SFJ which generates out higher engine performance.

3.2 Installation Layout of a Turbocharger

The installation-position of the turbocharger is determined the place near the gravity center of the vehicle, while the place must be located higher than the height of a side impact beam as the regulation specifies. **Figure 5** shows the installation location of a turbocharger. Although the exhaust manifold length was consequently limited, the layout of the intake manifold was designed at first.

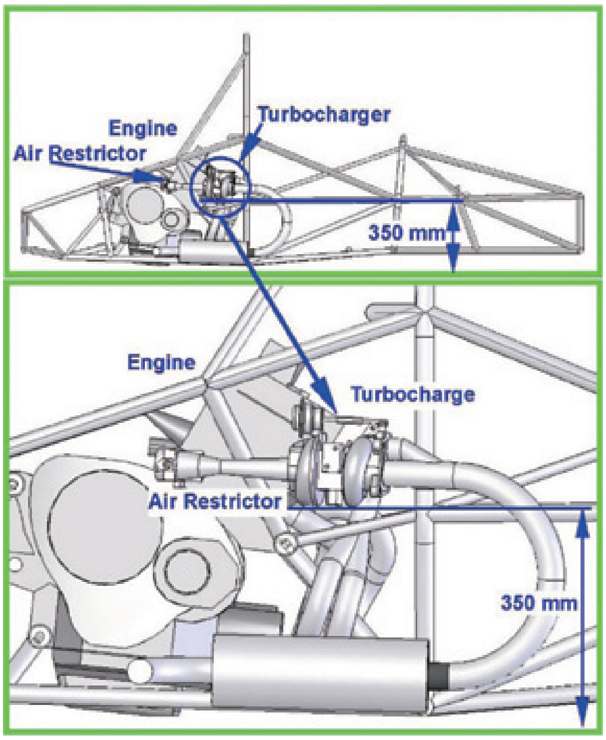


Figure 5 Layout of Installation Position of Gasoline Engine with Turbocharger, [Refer to 2017-18 Formula SAE® Rules].

4. Numerical Calculation of Intake System of Engine with Turbocharger by CFD

4.1 Regulation on Air Restrictor

The regulation for SFJ provides the limitation of intake air by using an air restrictor as follow; IC 1.6. Intake System Restrictor "In order to limit the power capability from the engine, a single circular restrictor must be placed in the intake system between the throttle and the engine and all engine airflow must pass through the restrictor. Any device that has the ability to throttle the engine downstream of the restrictor is prohibited. The maximum restrictor diameters are: Gasoline fueled cars-20.0 [mm] (0.7874 [inch]), E-85 fueled cars-19.0 [mm] (0.7480 [inch]). The restrictor must be located to facilitate measurement during the inspection process. The circular restricting cross section may not be movable or flexible in any way, e.g. the restrictor may not be part of the movable portion of a barrel throttle body. If more than one engine is used, the intake air for all engines must pass through the one restrictor" [34]. The air restrictor must be placed in the intake system between the throttle and the engine. Furthermore, turbocharger must be also arranged between the air restrictor and engine, which was different from a typical intake system layout shown in Figure 5. That is, since the arrangement of air restrictor, throttle and turbocharger was not typical, the performance improvement of the engine should be sufficiently investigated. Refer to appendix II.

4.2 CFD Numerical Calculation Code

CFD is the abbreviation of Computational Fluid Dynamics and is a software code for the computer simulation of the scientific technology which estimates the technological phenomenon of the flow, the heat transfer, the mass transport, the chemical reaction and so on. The computation means to solve the equation of the physical law using the values on the computer. Flowizard version 2.0.4, of which the computation principle uses the finite volume law, was used as the CFD analysis code for the this study [17], [18]. The area of the analysis object is divided into the finite cells by the discretization and each cell is replaced into the algebraic equation by the discretization which can compute conservation equations such as mass, momentum and energy on the computer. Air restrictor must be attached between the side of the engine and the throttle body as described in the regulation. Therefore, throttle body, air restrictor, turbocharger, and intake collector were arranged from upstream in the layout of the intake system.

4.3 Air Restrictor

(1) Numerical Calculation Conditions of CFD

Air inflow change by the shape of air restrictor had been previously studied on normal aspiration (NA) engine in the previous paper [03] – [08]. Therefore, it was decided to investigate the change of the flow by the change of the shape of air restrictor when introducing a turbocharger. The computation parameters were the thickness of the butterfly valve at the wide open throttle, atmosphere pressure, inlet pressure and air mass as incompressible fluid. Also, flow on the side of outlet was restricted by the entrance diameter, 32 [mm] of the turbocharger. These analysis conditions are shown in **Table 1** and the used soft wares are shown in **Table 2**.

Table 1 Numerical Calculation Conditions of Air Restrictor, [CFD, Flowizard Version 2.0.4].

Inlet	Analysis Condition	Velocity Inlet
	Initial Condition	60 m/s
Outlet	Analysis Condition	Out Flow

Table 2 Uses Soft wares, [CAD and CFD Numerical Calculation Code].

CAD Software	Solid Works 2004
CFD Software	Flowizard 2.0.4

(2) Air Restrictor Shape

In the regulation of Formula SAE, intake air must absolutely flow into air restrictor with 20 [mm] diameter. If the flow loss influenced by the shape of air restrictor is minimized, the engine output will be improved. Because an appropriate diffuser angle was found 3.5 [degree] in the study last year. The basic model is shown in **Figure 6**. After one kind of diffuser angle was selected on the base of these results, we decided to consider the influence of pressure difference occurred by the length of the inducer part. It was assumed that the entrance diameter of air restrictor is 40 [mm], that is the exit diameter of the throttle body, and the exit diameter of air restrictor is 32 [mm] that is the entrance diameter of the turbocharger. Assuming that the length of the inducer is 35 [mm] when changing a diffuser angle, and diffuser was fixed when changing inducer length, the numerical computations are done. The numerical calculated results were evaluated by using the difference pressure between entrance and exit of the air restrictor and the exit flow rate. **Figure 6** to **Figure 16** show the diffuser angle of the test air restrictors including the base air restrictor in the previous paper [01] – [08].

Figure 6 Basic CFD Model (Three-Dimensional Solid CAD Model) [Shape] of Air Restrictor for SFJ Vehicle

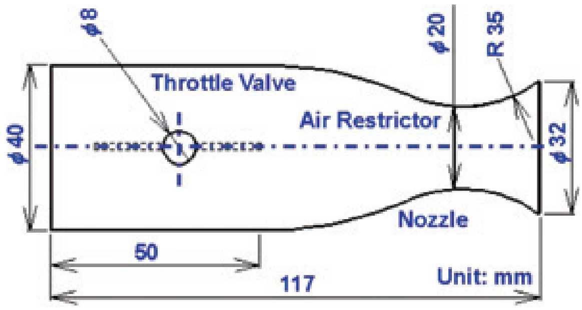


Figure 7 Dimension of Air Restrictor (Three-Dimensional Solid CAD Model) for SFJ Vehicle [Shape: 01, Radius: 30 [mm]].

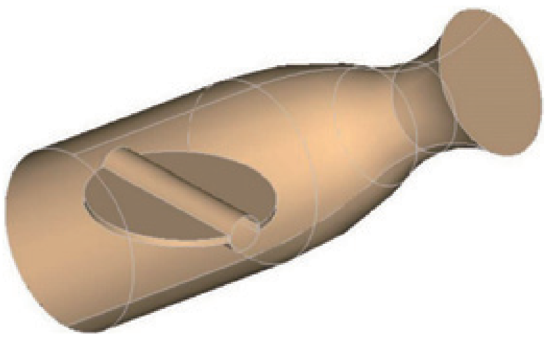


Figure 8 Numerical Calculation Model of Air Restrictor by the CFD for SFJ Vehicle, [Shape: 01, Radius: 30 [mm]].

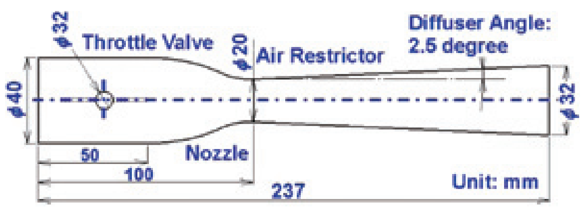


Figure 9 Dimension of Air Restrictor (Three-Dimensional Solid CAD Model) [Shape 02, Diffuser Angle: 2.5 [degree]]

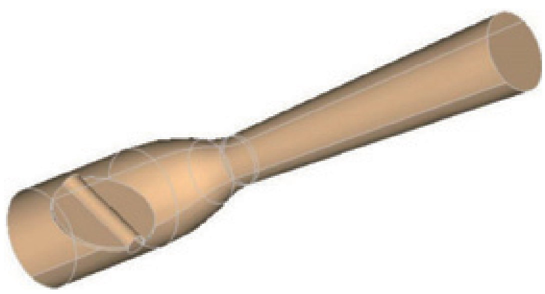


Figure 10 Numerical Calculation Model of Air Restrictor by the CFD for SFJ Vehicle, [Shape 02, Diffuser Angle: 2.5 [degree]]

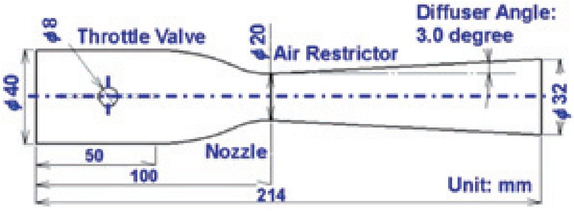


Figure 11 Dimension of Air Restrictor (Three-Dimensional Solid CAD Model) [Shape 03, Diffuser Angle: 3.0 [degree]]

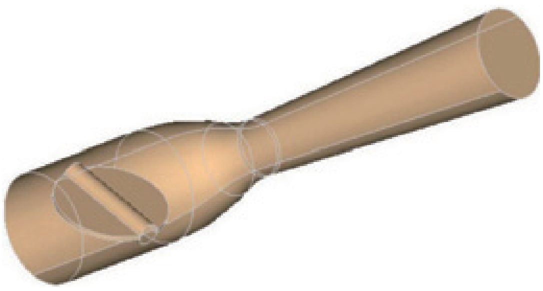


Figure 12 Numerical Calculation Model of Air Restrictor by CFD for SFJ Vehicle, [Shape 03, Diffuser Angle: 3.0 [degree]]

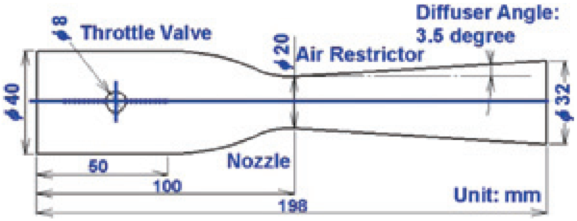


Figure 13 Dimension of Air Restrictor (Three-Dimensional Solid CAD Model) [Shape 04, Diffuser Angle: 3.5 [degree]]

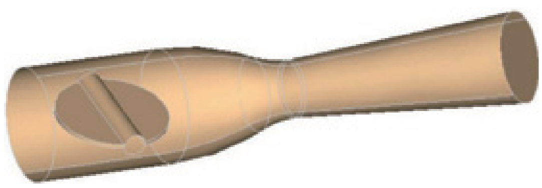


Figure 14 Numerical Calculation Model of Air Restrictor by CFD for SFJ Vehicle, [Shape 04, Diffuser Angle: 3.5 [degree]]

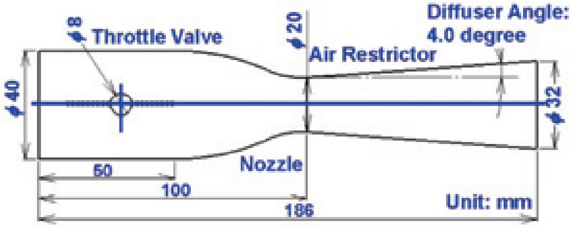


Figure 15 Dimension of Air Restrictor (Three-Dimensional Solid CAD Model) [Shape 05, Diffuser Angle: 4.0 [degree]]

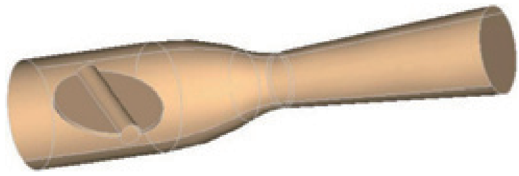
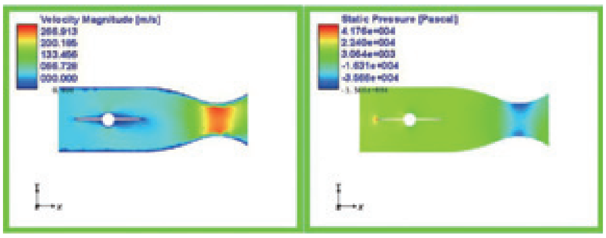


Figure 16 Numerical Calculation Model of Air Restrictor by CFD for SFJ Vehicle, [Shape 05, Diffuser Angle: 4.0 [degree]]

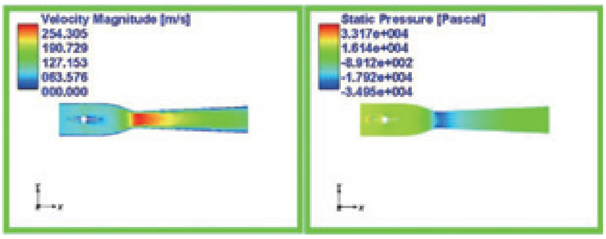
(3) CFD Numerical Calculation Results

The CFD results of the shapes of the air restrictors are shown in **Figure 17** to **Figure 22**. The results in **Table 3** show that the diffuser angle of 3.5 [degree] was the minimum pressure difference between inlet and outlet of the test air restrictors. These results were as similar as the results of the air restrictors with the normal aspirated engine as mentioned in the previous paper [03]−[08]. If flow velocity



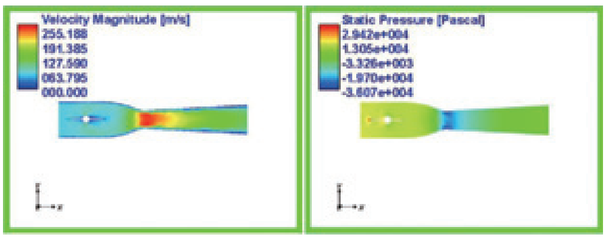
Velocity Magnitude [m/s] Static Pressure [Pa]

Figure 17 Numerical Calculation Results of Air Restrictor by CFD for SFJ Vehicle, [Shape 01, Radius 30 [mm]]



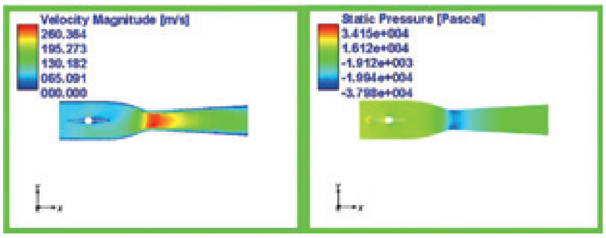
Velocity Magnitude [m/s] Static Pressure [Pa]

Figure 18 Numerical Calculation Results of Air Restrictor by CFD for SFJ Vehicle, [Shape 02, Diffuser Angle: 2.5 [degree]]



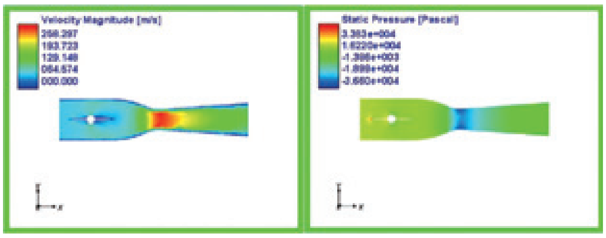
Velocity Magnitude [m/s] Static Pressure [Pa]

Figure 19 Numerical Calculation Results of Air Restrictor by CFD for SFJ Vehicle, [Shape 03, Diffuser Angle: 3.0 [degree]]



Velocity Magnitude [m/s] Static Pressure [Pa]

Figure 20 Numerical Calculation Results of Air Restrictor by CFD for SFJ Vehicle, [Shape 04, Diffuser Angle: 3.5 [degree]]



Velocity Magnitude [m/s] Static Pressure [Pa]

Figure 21 Numerical Calculation Results of Air Restrictor by CFD for SFJ Vehicle, [Shape 05, Diffuser Angle: 4.0 [degree]]

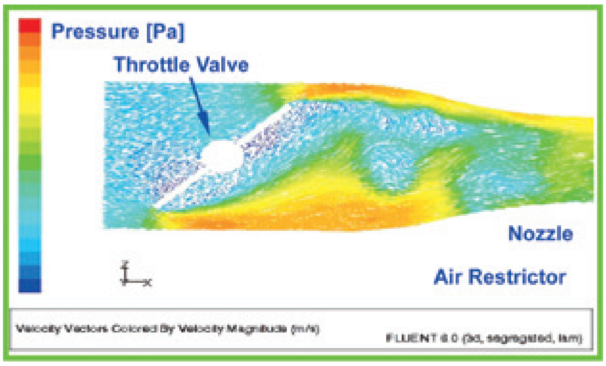


Figure 22 Pressure Distribution of Air Restrictor by CFD for SFJ Vehicle, [Throttle Valve Open Rate: 50 [%], Shape 04, Diffuser Angle: 3.5 [degree]]

Table 3 Pressure Differences between Back and Forth of Air Restrictor with Different Diffuser Angle. [Shape: 04, Diffuser Angle: 3.5 [degree], Pressure Difference: 09420 [Pa]].

CAD Software	Solid Works 2004
CFD Software	Flowizard 2.0.4

in air restrictor varies, it was confirmed that the air restrictor with diffuser angle 3.5 [degree] was better than any test air restrictor judging that the turbulence in the outlet of this air restrictor was smaller than the other air restrictors.

5. Intake Air Cooling by the Secondary Injector

5.1 Effectively of the Secondary Injector

One of the problems caused by the turbocharged engine was the rise of the intake air temperature. Because the rise of the intake air temperature was the direct cause which connects to the decline of the engine performance, an intercooler was usually attached to the turbocharged engine in order to cool the intake air. However, the demerits of intercooler are the larger installing space, the increase of the engine weight, and the aggravation of the engine response. Since this cooling system was not appropriate to the SFJ vehicle, the other system had to be investigated. Therefore, we investigated whether the cooling system by the secondary injector applied for last year's vehicle can also support turbocharged engine. This cooling system cooled intake air using the vaporization heat of the fuel sprayed from the secondary injector and the vaporized fuel can support better combustion. Firstly, the experiments were carried out to examine the change of the air charge temperature to the rise of the engine speed as shown in **Figure 23**. Since these results were obtained from the experiment in the water temperature and oil temperature's being constant, the air charge temperature was rising with the time elapse. Secondly, we examined cooling effect of the injection quantity and mounting position of the secondary injector on intake air temperature. As shown in **Figure 24** and **Figure 25**, the fuel injection only by the secondary injector at full load operation was very effective for the cooling of intake air. Moreover, the improvement of the engine performance requires promoting fuel atomization in order to raise the combustion efficiency because the mounting position of the secondary injector was far from the induction port. **Figure 26** shows the effect of the mounting position difference of the secondary injector on the cooling of the intake air. These results indicate that the secondary injector mounted at 200 [mm] can effectively cool the intake air more than that mounted at 500 [mm]. The other parts such as fuel system,

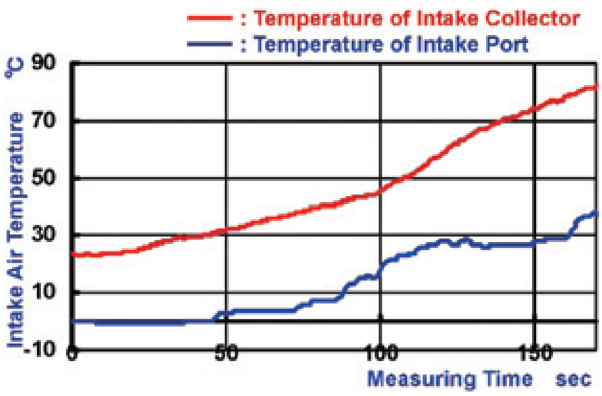


Figure 23 Change of the Air Charge Temperature in the Intake Collector and Entrance of Intake Port to Engine Speed [Experimental Results of Engine Benchmark Test at Full Load]

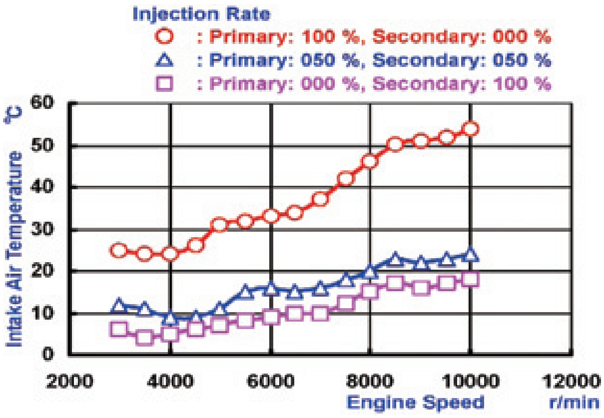


Figure 24 Rise of Intake Air Temperature at Intake Port when Changing the Injection Rate between the Primary and Secondary Injectors, [Comparison of [Primary Injection Rate: 100 [%], Secondary Injection Rate: 000 [%], [Primary Injection Rate: 050 [%], Secondary Injection Rate: 050 [%] and [Primary Injection Rate: 000 [%], Secondary Injection Rate: 100 [%]].

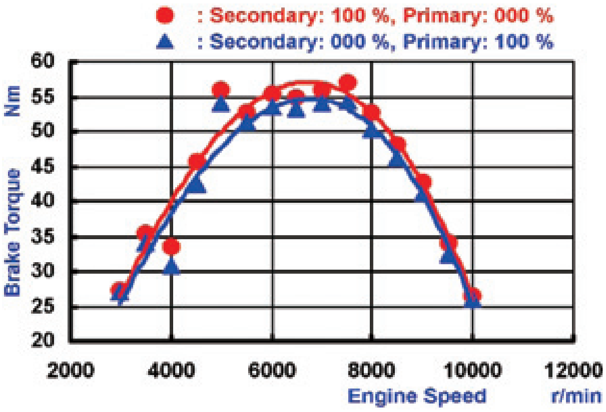


Figure 25 Rise of Brake Torque when Changing the Injection Rate between the Primary and Secondary Injectors, [Comparison of [Primary Injection Rate: 000 [%], Secondary Injection Rate: 100 [%] and [Primary Injection Rate: 100 [%], Secondary Injection Rate: 000 [%]].

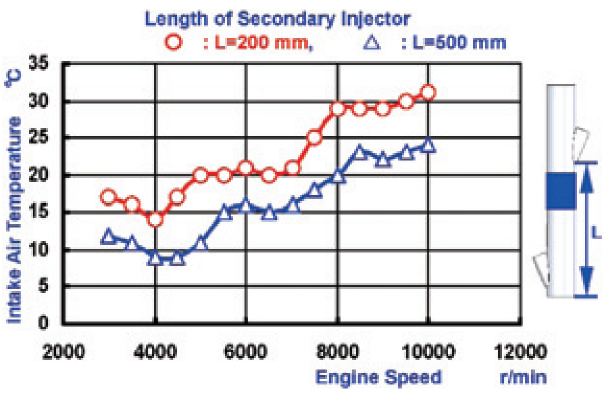


Figure 26 Rise of Intake Air Temperature at Intake Port when Changing the Secondary Injector Position, [Comparison of Length: 200 [mm] and Length: 500 [mm]].

intake system and so on may not limit the ideal mounting position as shown in the results. However, the mounting position should follow the experimental results to avoid engine overheat in vehicle running.

6. Engine Performance Simulation by the VES

The regulation of SFJ obligates to attach an air restrictor within caliber 20 [mm] to an engine of which displacement volume is equal to or less than 610 [cc]. As running course consists of straight lines equal to or less than 100 [m] and many corners, the acceleration performance in the corner exit and the turn ability in the curve are more important than the maximum speed. Busyness of the gearshift changing also leads to the time loss in consideration of the course layout. Therefore, to run the vehicle faster requires an engine with higher torque band in wide engine speed range. The usual engine speed was from 4000 [r/min] to 10000 [r/min] when running in the course in the last year competition. An engine for SFJ needs torque characteristics with torque increasing in low engine speed and a flat torque curve in the higher engine speed. In order to get the above-mentioned engine performance for the shortest development term, we decided to use a simulation software, VES [19]–[22], which can estimate the performance characteristics of the engine with variation of design factors and reduce the number of the engine bench tests.

6.1 VES Numerical Calculation Model

The numerical calculation model that can express the engine performance test results must be built on VES. The input data in each of length, caliber, curve of the pipes had to be entered in the intake system and the exhaust system. The computational conditions must be equal to the conditions of the engine performance test. The ignition timing was assumed as averaged value obtained from the base engine, because the actual ignition timing varied by influence of the inlet-exhaust pipe length and the air charge temperature. Moreover, the temperature, the humidity, the atmospheric pressure, the engine specifications (cam profile, bore, stroke, compression ratio, A/F (Ratio of air, fuel) and so on) must be entered. The purpose of VES in this study was to investigate the influence of engine design parameters on engine performance more than coordination between experimental values and simulated values in the abridgment of the design period. Flow rate and friction loss by Intake valve and exhaust valve, and air resistance were calculated by VES. **Figure A-2** shows the computation model. Refer to appendix III.

6.2 Coordination of the VES Numerical Calculation and the Engine Performance Test

Figure 27 and **Table 4** show the curves of the base engine performance and the specifications of the parameters. **Figure 28** shows the comparison between the results of the engine performance tests and the performance simulation results by VES. The lines in **Figure 27** show the experimental values. The line in **Figure 28** indicates the simulation values to compare between the experimental values and the simulation ones. The torque values from 4000 [r/min] to 6000 [r/min] are varied by the influence of the turbocharging. Although 12 [%] of errors occur in the maximum, the tendency can approximately express the results of the experimental results in usual engine speed range. This VES model is useful to get the engine performance on the computer and provided useful information to carry out the engine bench tests in the limited schedule, even if a designed engine part should be changed.

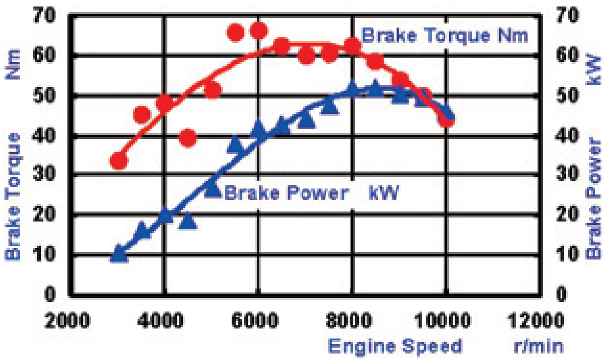


Figure 27 Performance Curves of Base Engine [The Ferformance Curves of Development Base Engine, [Experimental Results of Engine Benchmark Test at Full Load], Brake Torque and Brake Power].

Table 4 Main Specifications of Base Engine [PC-37E Gasoline Engine with RHB 31 Turbocharger].

Items	Specifications
Engine Name	PC-37E
Compression Ratio	10.5
Camshaft	EU Used
Name of Turbocharger	RHB 31
Diameter of Throttle	40 mm
Length of Intake Runner	500 mm [ϕ37 mm]
Length of First Exhaust Runner	450 mm
Length of Second Exhaust Runner	300 mm
Length of Third Exhaust Runner	100 mm
Volume of Intake Collector	1.5 Litter

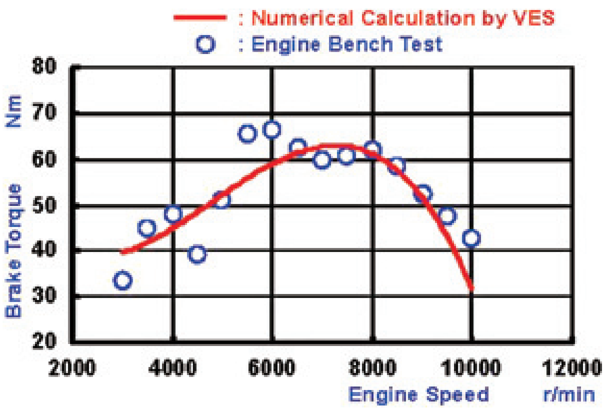


Figure 28 Comparison between Numerical Calculation by VES and Engine Benchmark Test [Engine Performance Curves of Development Base Engine, Comparison of Numerical Calculation Simulation by VES and Engine Benchmark Test]

6.3 Design-Limit Values

Considering the engine mounting space of the vehicle this year, the parts of the intake system and the exhaust system were limited in size, length and weight. The details are as follows.

[1] Throttle:

Considering the engine performance tests of the engine with a throttle diameter and the flow of air restrictor by CFD as test parameters, caliber 40 [mm] and length 20 [mm] of the throttle body was determined.

[2] Air Restrictor:

Inducer length 45 [mm], diffuser angle 3.5 [degree] and length of air restrictor 148 [mm] are determined from the results of the preceding chapter. Moreover, entrance diameter 40 [mm] that was equal to the exit diameter of the throttle, and exit diameter 32 [mm] that was entrance diameter of the turbocharger were determined.

[3] Intake Collector Volume:

Intake collector volume was adopted 1.5 [Litter] from the numerical calculation results using CFD and the engine performance tests.

[4] Intake Runner:

Considering the layout of an intake system, the intake runner lengths are computed every 50 [mm] in 200 [mm] to 600 [mm]. Also, the intake runner diameter was 37 [mm] according to the intake port diameter.

[5] First Exhaust Runner:

The first exhaust runner length varied from 300 [mm] to 450 [mm] by engine space and engine built-in layout. Moreover, caliber of the exhaust runner is 30.5 [mm] as same as caliber of intake runner in consideration of exhaust port caliber.

[6] Second Exhaust Runner:

The second exhaust runner length varied from 100 [mm] to 300 [mm] by engine mounted space and engine built-in layout. Moreover, minimum length 100 [mm] was determined in consideration of set part with the first exhaust runner and the second exhaust runner. The caliber was 30.5

[mm] as same as the caliber of the first exhaust runner.

[7] Third Exhaust Runner:

The third exhaust runner length varied from 100 [mm] to 300 [mm] by engine space and engine built-in layout. Moreover, minimum length 100 [mm] was determined in consideration of set part with the second exhaust runner and the third exhaust runner. The caliber was 30.5 [mm] as same as each caliber of the first exhaust runner and the second exhaust runner. The exit caliber to turbocharger was tapered diameter from 30.5 [mm] to 28 [mm] among the 10 [mm] length to adjust to entrance diameter 28 [mm] of the turbocharger.

6.4 Effect of each Element of Intake and Exhaust System on Engine Performance

[1] Effect of Intake Runner Length on Engine Performance

Figure 29 shows the relationship between brake torque and the intake runner length. These results indicate that the intake length was an effective factor on the engine performance by matching the wave length of intake pulsation to the intake runner length. In the simulation results of engine performance by VES, in the range of 200 [mm] <L <400 [mm], the peak value of the shaft torque and the maximum engine speed are close to 70 [Nm] and 8000 [r / min] There is no noticeable change. On the other hand, in the range of 450 [mm] <L <600 [mm], the peak value of the axial torque decreases and the maximum engine rotational speed tends to shift to the low rotational speed region. With the fuel injected from the primary injector (for example, in-cylinder gasoline direct injection (GDI) engine), in this respect, the in-cylinder gas temperature tends to decrease. Also, no phenomenon of gasoline knock was seen.

[2] Effect of Exhaust Runner Length on Engine Performance

Figure 30, **Figure 31** and **Figure 32** show the effect of each exhaust runner length on the engine performance. These lengths have effect on the engine performance of the turbocharged engine hardly. However, the second exhaust runner length had a little improvement of the engine performance in higher engine speed range than 7000 [r/min]. It seemed to be a reason that the exhaust resistances at combined part between the first runner and the second runner were reduced with increase of the first runner length or the second runner length.

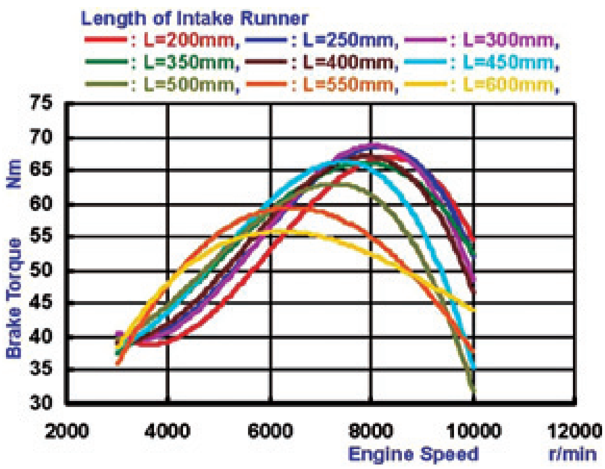


Figure 29 Relationship between Brake Torque and Length of Intake Runner, [The Length of Intake Runner: L-200 to 600 [mm], Step 50 [mm], Engine Performance Simulation by the VES for SFJ Vehicle].

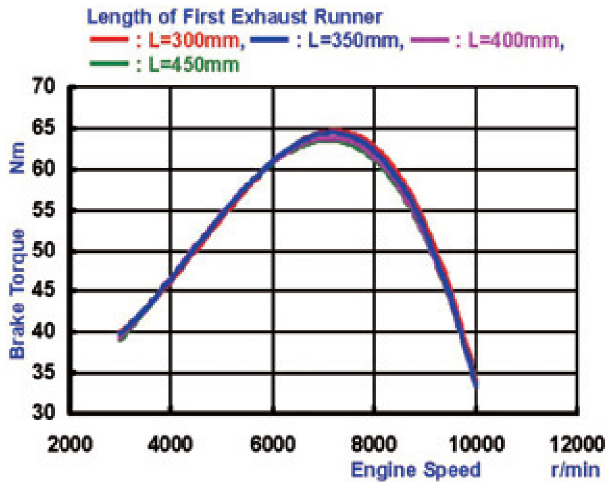


Figure 30 Relationship between Brake Torque and Length of First Exhaust Runner, [The Length of First Exhaust Runner: 300 to 450 [mm], Step 50 [mm], The Length of Secondary, Thirdly Exhaust Runner: Constant, Engine Performance Simulation by the VES for SFJ Vehicle].

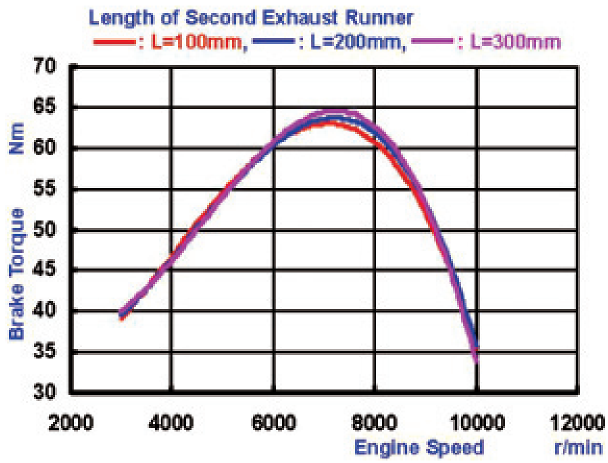


Figure 31 Relationship between Brake Torque and Length of Second Exhaust Runner, [The Length of Secondary Exhaust Runner: 100 to 300 [mm], Step 100 [mm], The Length of Firstly, Thirdly Exhaust Runner: Constant, Engine Performance Simulation by the VES for SFJ Vehicle].

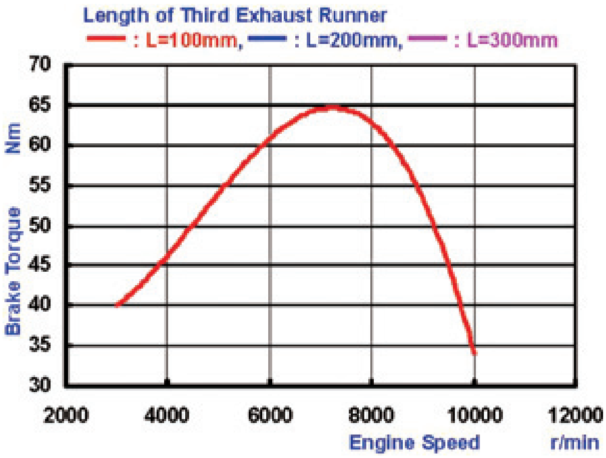


Figure 32 Relationship between Brake Torque and Length of Third Exhaust Runner, [The Length of Thirdly Exhaust Runner: 100 to 300 [mm], Step 100 [mm], The Length of Firstly, Secondary Exhaust Runner: Constant, Engine Performance Simulation by the VES for SFJ Vehicle].

6.5 Verification of Engine Performance by the VES

These numerical calculated results on engine performance by using VES had to be verified whether they can express actually engine performances obtained from the engine bench test. The implementation of engine bench test was decided to verify the combination of the design factors with the best engine performance in the calculation results. That is, if the test results exceed the target values, it can be said that these computation results were successful. **Figure 33** shows comparison between the test results and the target values. The performance curve of the engine, which was improved by the design factors based on the numerical calculation data obtained from VES, exceeded a target value sufficiently the values of the last years highest engine performance, because we judged that we should get the flat torque curve in our driving engine speed range 4000 [r/min] to 9000 [r/min]. In this research, the design concept is the flat shaft torque and shaft output of a gasoline engine with a turbocharger. From the simulation result of VES in Fig. 29, the shaft torque tends to be improved by shortening the length of intake runner. This depends on the performance (A / R) of the turbocharger alone. However, we must also consider the regulations of the competition.

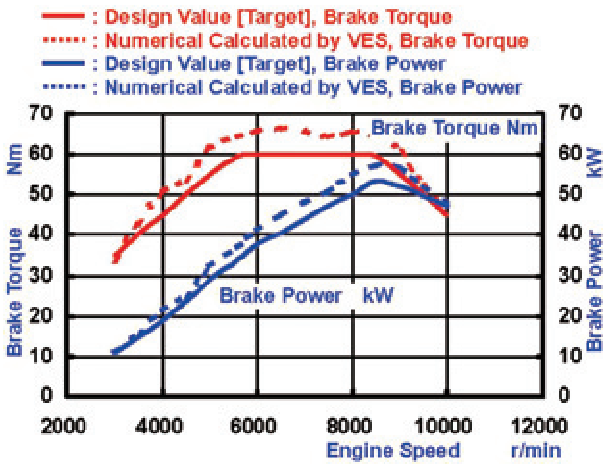


Figure 33 Comparison between Numerical Calculation by the VES and Design Values, [The Comparison of Design Value [Target] and Numerical Calculation Simulation by the VES for SFJ Vehicle].

7. Summary

The engine performance improvements in our previous studies were repeatedly carried out on the trial and error for a long time. Achievement of performance improvement of turbocharged engine in short design time, which was our first try, means that VES was a useful tool for engine design. The following knowledge were obtained from this study;

[1] As each engine design factor could be treated as the independent parameter on VES, VES is a very useful tool to analyze the effect of each factor on the engine performance at the short time.

[2] It succeeded to make a high-speed type normal aspirated engine for motorcycles turbocharged. This fact on basis of the experiments shows that the engine performance could be improved in order to drive a SFJ vehicle more easily.

[3] The approach of this study made it possible to make the engine performance better than the target performance.

However, some engine troubles might occur through the vehicle running tests. We planned to implement some trouble shootings and improve the engine performance by repeating a running test in the near future.

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Appendix I

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IC1.5 Throttle and Throttle Actuation

IC1.5.1 Carburetor/Throttle Body The car must be equipped with a carburetor or throttle body. The carburetor or throttle body may be of any size or design.

IC1.5.2 Throttle Actuation The throttle must be actuated mechanically, i.e. via a cable or a rod system, unless IC1.11 - IC1.16 is followed for Electronic Throttle Control which replaces the rest of IC1.5

IC1.5.3 The throttle cable or rod must have smooth operation, and must not have the possibility of binding or sticking.

IC1.5.4 The throttle actuation system must use at least two（2）return springs located at the throttle body, so that the failure of any component of the throttle system will not prevent the throttle returning to the closed position. Throttle Position Sensors（TPS）are NOT acceptable as return springs.

IC1.5.5 Throttle cables must be at least 50.8［mm］（2［inches］）from any exhaust system component and out of the exhaust stream.

IC1.5.6 A positive pedal stop must be incorporated on the throttle pedal to prevent over stressing the throttle cable or actuation system.

IC1.5.7 The throttle pedal cable must be protected from being bent or kinked by the driver's foot when it is operated by the driver or when the driver enters or exits the vehicle.

IC1.5.8 If the throttle system contains any mechanism that could become jammed, for example a gear mechanism, then this must be covered to prevent ingress of any debris.

IC1.5.9 Carburetors are not allowed on boosted applications.

Appendix II

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IC1.6 Intake System Restrictor

IC1.6.1 In order to limit the power capability from the engine, a single circular restrictor must be placed in the intake system and all engine airflow must pass through the restrictor. The only allowed sequence of components are the following: a. For naturally aspirated engines, the sequence must be（see Fig 1）: throttle body, restrictor, and engine. b. For turbocharged or supercharged engines, the sequence must be（see Fig 2）: restrictor, compressor, throttle body, engine.

IC1.6.2 The maximum restrictor diameters at any time during the competition are: a. Gasoline fueled cars – 20.0 mm（0.7874 inch）b. E-85 fueled cars – 19.0［mm］（0.7480［inch］）

IC1.6.3 The restrictor must be located to facilitate measurement during the inspection process.

IC1.6.4 The circular restricting cross section may NOT be movable or flexible in any way, e.g. the restrictor may not be part of the movable portion of a barrel throttle body.

IC1.6.5 If more than one engine is used, the intake air for all engines must pass through the one restrictor.

Appendix III

Figure A-2 shows the numerical calculation model of engine performance by the virtual engine simulation.

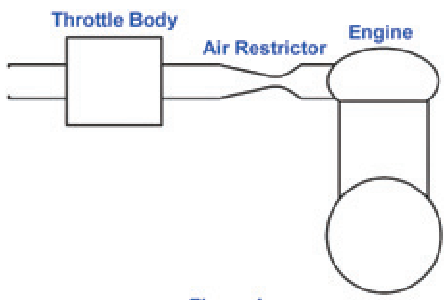

Figure 1

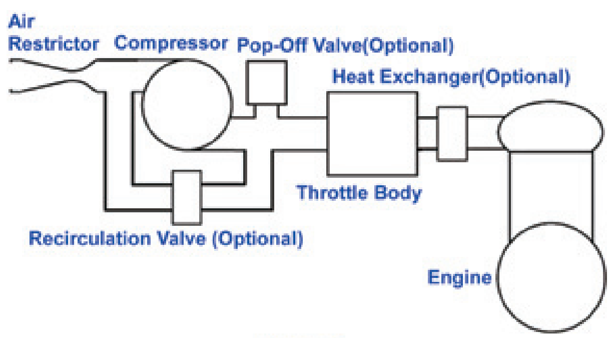

Figure 2

Figure A-1 Intake System Restrictor *A

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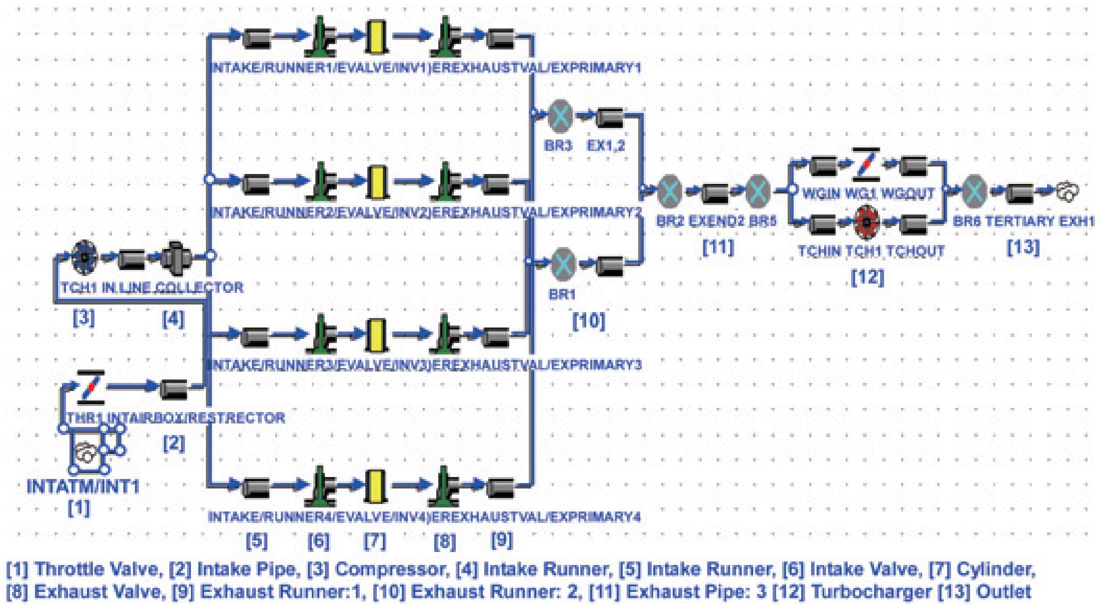

Figure A-2 Numerical Calculation Model of Engine Performance by the Virtual Engine Simulation