

Modeling of Two Charge-air Cooling Turbo-charging Systems for Spark Ignition Engines

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ABSTRACT

Two charge-air cooling turbo-charging systems, named Turbo-Cool and T²C, have been introduced. Turbo-Cool employs an air turbine expander/suction-compressor unit in the intake side and a VGT in the exhaust side, while T²C employs an air turbine expansion in the intake side and an exhaust VGT with a same shaft with the air turbine. A Nissan SR20DET turbocharged gasoline engine with the two charge-air cooling turbo-charging systems have been modeled using GT-SUITE 6.0 engine simulation code. Modeling results show that either Turbo-Cool or T²C must be combined with exhaust VGT. The air turbine in both systems can be either VGT or fixed. Modeling results also show that the power at WOT and rated engine speed of the engine with the proposed charge-air cooling turbo-charging systems can be increased by 20~30%, the maximum torque at WOT can be increased by 46~71%, and the BSFC at WOT and rated engine speed can be decreased by 4~5%. The large improvement of the torque at middle and low engine speed can result in an excellent torque back-up and correspondingly drivability for vehicle use.

INTRODUCTION

Fuel economy and engine power density are the two main concerns of engine researchers. Turbo-charging is an extraordinarily effective method for raising internal combustion (IC) engine power density [1, 2] as well as, theoretically, engine efficiency. Applying turbo-charging to piston engines in practice to realize this theoretical potential requires an effective matching of the turbocharger to the piston engine. In the case of diesel engines, matching of turbochargers with piston units has been satisfactorily, if not optimally, developed using a combination of exhaust-gas bypassing through waste-gate and control of fuel rate. As a result, turbocharged diesels enjoy a clear and irresistible superiority in power and efficiency over naturally-aspirated (NA) diesels in heavy-duty use (as well as in light-duty application), and are hugely successful in the marketplace [3].

But waste-gate system wastes some available exhaust gas energy. An alternative way to avoid the “waste” is a turbocharger with a turbine that has variable nozzle area capable of accommodating varying exhaust-gas mass flow. Such turbochargers are known as Variable Geometry Turbine (VGT). Although VGT turbochargers were originally developed for automotive gasoline engines, they have not been adopted for turbocharged gasoline engines [4]. Wang and Yang [5] examined this specific issue. They reported their modeling results for a gasoline engine with VGT and concluded that VGT for spark ignition engines must be supplemented by other measures, such as charge-air cooling systems. So Wang and Yang [5] proposed a Turbo-Cool charge-air cooling turbo-charging system. They demonstrated that Turbo-Cool could be a supplement for VGT, and VGT was essential for Turbo-Cool. Their modeling results showed that Turbo-Cool with exhaust VGT could improve performance of both fuel economy and power density relative the baseline turbocharged gasoline engine. In their study the air turbine was fixed. They didn't optimize the mass flow distribution in the two paths of Turbo-Cool unit. Also they didn't consider the effect of the effectiveness of heat exchanger of Turbo-Cool on performance of the engine.

The more important difference between turbocharged spark ignition (SI) engines and turbocharged diesel engines is that SI engines are prone to knock. Knock is an abnormal combustion phenomenon of homogenous charge spark ignition combustion engines, which is normally a propagating turbulent flame sweeping through “engine combustion chamber” towards the end-gas region. Following the discussion by Heywood [1], knock is the extremely rapid release of much of the energy contained in the end-gas ahead of the propagating turbulent flame, resulting in high local pressure—causing repeating reflective pressure waves. When the fuel-air mixture in a combustion chamber is compressed to sufficiently high pressure and temperature over a period, known as the time of auto-ignition, auto-ignition of the fuel-air mixture will take place. Normally, the propagating turbulent flame reaches the end-gas region before auto-

ignition. However, if the time of auto-ignition is shorter than the time it takes for the flame to reach the end-gas region, auto-ignition of the end-gas occurs before the arrival of propagating flame, resulting in knock.

In the case of turbocharged engines, the most widely used method of reducing the start of compression temperature is intercooling. However, the performance of the intercooler, and thus the extent of the charge-air temperature reduction, is limited by the temperature of the sink to which the heat is rejected and the effectiveness of the intercooler itself. With such a system it is not possible to reduce charge-air temperature below the sink temperature [6]. Turbo-Cool proposed by Wang and Yang [5, 7] can be an approach to cool charged air. It has a better cooling effect than intercooler. Taitt, Garner, Swain, Bassett, Pearson, and Turner [6] proposed an alternative charge-air intake conditioner system which used a turbine expander in the intake side. They finished their first step of study: thermodynamics analysis of performance characteristics of charge-air intake conditioner system.

In this paper, based on the study of Wang and Yang [5], modeling of Turbo-Cool with both exhaust VGT and air turbine VGT and optimization of the mass flow distribution in the two paths of Turbo-Cool unit and optimization of the effectiveness of the heat exchanger of Turbo-Cool unit, have been performed. Also based on the charge-air intake conditioner system proposed by Taitt, Garner, Swain, Bassett, Pearson and Turner [6], a new turbine expansion charge-air cooling system with a simple structure, named T^2C , is introduced. T^2C considers the effect of exhaust system, especially the back pressure, on the performance of the engine. For T^2C , modeling of three cases, say T^2C with both exhaust VGT and air turbine VGT, T^2C with exhaust VGT and fixed air turbine and T^2C without any VGT, has been performed. This paper will begin with a description of the Turbo-Cool and T^2C charge-air cooling turbo-charging system. Then the models in the simulation code and the parameters of the baseline turbocharged gasoline engine are briefly described. In the last part, the simulation results and some discussions will be.

DESCRIPTIONS OF TURBO-COOL AND T^2C CHARGE-AIR COOLING TURBO-CHARGING SYSTEMS

Achieving higher engine effective expansion ratio (EER) by higher gas expansion through exhaust-gas-driven turbine is constrained by the knock-limit, which depends on the temperature of the end-gas. The end-gas temperature and pressure are related to the temperature and pressure of the mixture at the start of the combustion (SOC), and the gas state at SOC is, in turn, related to that at the start of piston compression. It is useful to direct investigations to the intake side of the piston engine.

TURBO-COOL CHARGE-AIR COOLING TURBO-CHARGING SYSTEM

One approach is to cool the charge-air of the intake side. Two types of charge-air cooling turbo-charging systems are considered in this paper. The first charge-air cooling turbo-charging system is depicted in figure 1. It is called Turbo-Cool charge-air cooling turbo-charging system [5, 7]: incorporating, on the exhaust side of the engine, a VGT turbocharger and, on the intake side of the engine, the apparatus, now named as the Turbo-Cool unit. Turbo-Cool charge-air cooling turbo-charging system utilizes exhaust-gas expansion in turbine to achieve higher engine EER in a straightforward manner, similar to the higher engine EER achieved in a turbocompound engine. The novelty of this system is that the excess power from higher exhaust-gas expansion through turbine is used for cooling intake charge air. The waste gate can be replaced by this system.

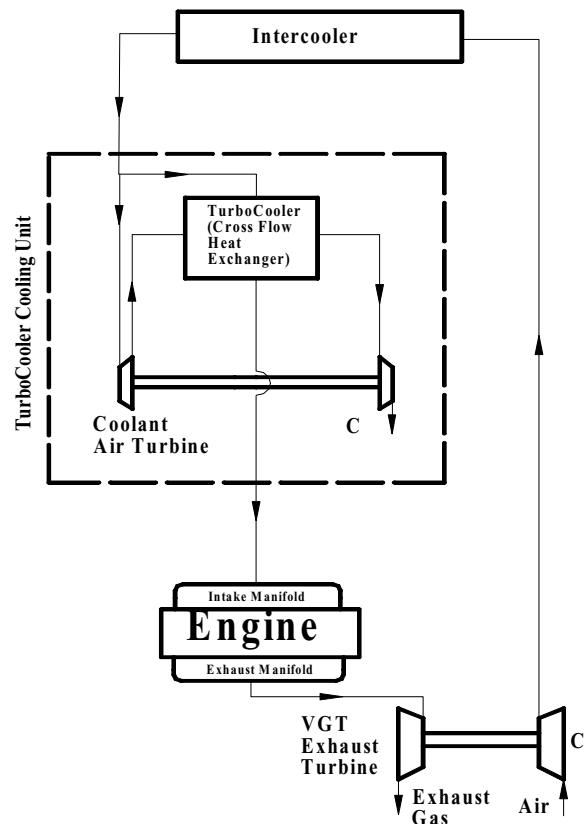


Figure 1 Turbo-Cool charge-air cooling turbo-charging system

Referring to Figure 1, the air stream exiting the intercooler on the intake side is divided into two paths: an air stream of the first path as the charge air, and an air stream of the second path. The first air stream is conveyed to the Turbo-Cooler heat exchanger. The second air stream is in communication with the inlet of a turbine expander of an expander/suction-compressor unit. This second air stream undergoes an expansion cooling process in the turbine expander. The exit pressure from the expander is determined by the balance between the turbine expander and the suction

compressor of the expander/suction-compressor unit. The energy expended by the second air stream in driving the turbine expander reduces the temperature of the second air stream. This low-temperature air stream leaving the turbine expander is then conveyed to the Turbo-Cooler heat exchanger, wherein heat transfer takes place from the first air stream of the charge air to the low-temperature second air stream. Exiting the heat exchanger, the second air stream is in communication with the inlet of the suction-compressor of the expander/suction-compressor unit. Then the second air stream undergoes a compression heating process in the suction-compressor, powered by the expander. The resulting temperature of the second air stream discharging from the turbocharging system is moderately above ambient air temperature, and the second air stream discharge represents only a moderate loss of available energy. The first air stream exiting from the Turbo-Cooler heat exchanger is conditioned to a lower temperature and is conveyed to the engine intake manifold as the engine charge air.

T²C CHARGE-AIR COOLING TURBO-CHARGING SYSTEM

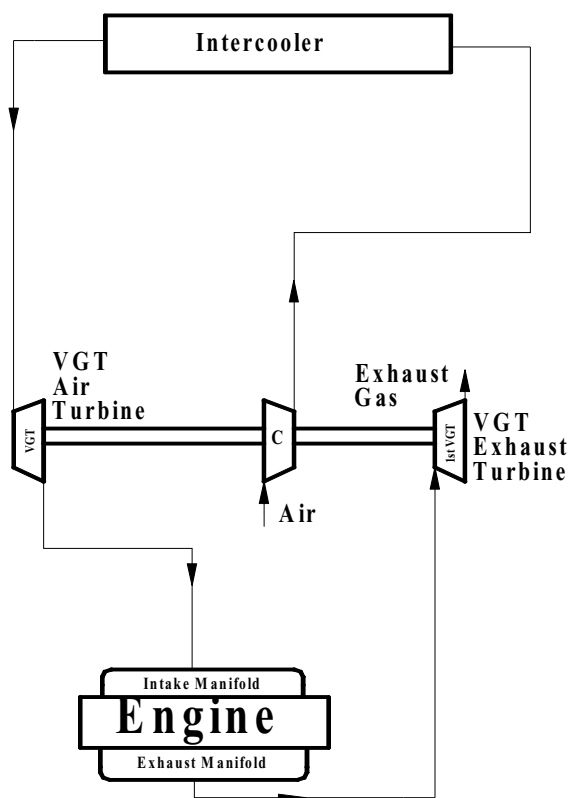


Figure 2 T²C charge-air cooling turbo-charging system

The second charge-air cooling turbo-charging system is depicted in figure 2. It differs from a conventional system by expanding the intake air after the intercooler to achieve a further reduction in temperature and pressure, along with a corresponding recovery of work. It also differs from the charge-air intake conditioner system investigated by Turner et al. [8] and Taitt et al. [6]. The

use of turbine expansion in T²C is different in two respects: the synergistic use of turbine expansion cooling and VGT exhaust turbine, and the air turbine expansion power being absorbed by compressor.

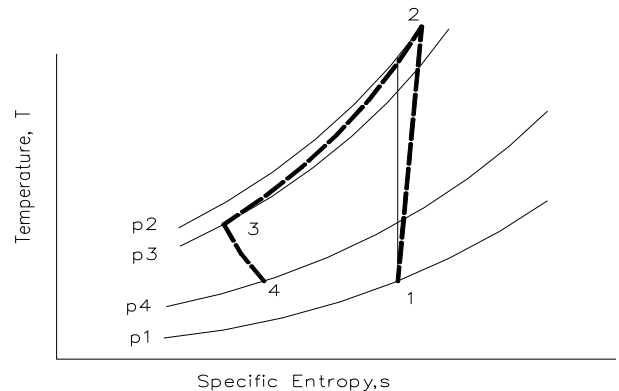


Figure 3 T-s diagram for the T²C charge-air cooling turbo-charging system

The principle of turbine expansion operation can be illustrated on the temperature-entropy (T-S) diagram in figure 3. Points 1 to 4 represent a real process. Air is compressed from points 1 to 2. Heat is rejected to point 3, from where it is expanded to the desired engine inlet condition at point 4. From figure 3 it can be seen that the charge air temperature reduction is no longer limited by the performance of the intercooler and may be cooled below the sink temperature, and careful control of the system allows a constant or higher charge air density (either high temperature, high pressure or low temperature, low pressure) [6]. Note that both designs in figure 1 and figure 2 have the option in that the air turbine (shown as a VGT kind in figure 2) can be either a fixed geometry kind or a VGT kind. The cooling effect, in the T²C charge-air cooling turbo-charging system, results from the expansion cooling of compressed and inter-cooled air. In T²C charge-air cooling turbo-charging system the compressor is driven by both the exhaust turbine and the air turbine.

The structure of T²C charge-air cooling turbo-charging system shown in figure 2 is simple. But if we don't have this kind of combined turbocharger for modeling and/or experiments, a thermodynamically-equivalent system-configuration of T²C can be used, which is show in figure 4.

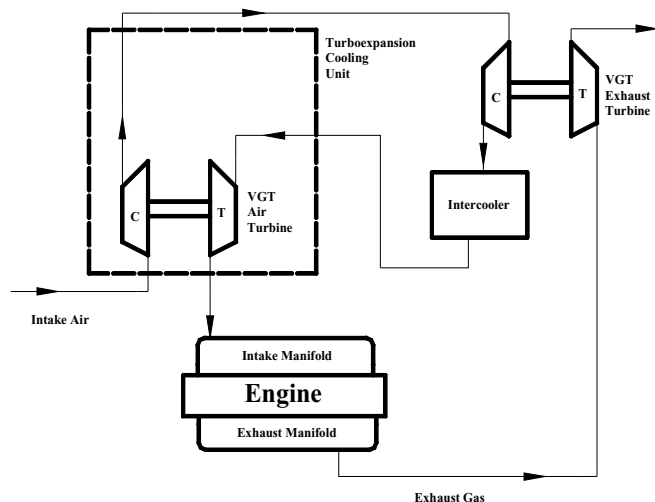


Figure 4 The thermodynamically-equivalent system-configuration of T²C

MODELS USED IN THE SIMULATION CODE AND PARAMETERS OF THE BASELINE TURBOCHARGED GASOLINE ENGINE

Engine performance modeling is used to study the potential application of the above two charge-air cooling turbo-charging systems. The Gamma Technologies' GT-SUITE commercial code (Gamma Technologies 2003) [9] is used for the modeling study of this paper. In this code, the one-dimensional unsteady gas flow model is used for the modeling of the intake and exhaust piping system. For the model discretization, the time steps should be chosen such that the Courant number is less than or equal to 0.8. The Courant number is defined as:

$$\text{Courant number} = \frac{\Delta t}{\Delta x} (|\vec{u}| + c) \quad (1)$$

Where Δt is the time step, Δx is the discretized element length, $\vec{u}$ is the fluid velocity, and c is the speed of sound. For the injection for SI engines, the fuel is injected at a rate that imposes the specified fuel-to-air ratio at all times. Air flow rate is measured at the point of injection. A SI Weibe combustion model is used. In this combustion model the burn rate or heat release is prescribed by the user with SI Weibe function. Inputs for the anchor angle (typically at 50% burn), duration (typically from 10% to 90%), and the Weibe exponent are required. In this simulation code, users can also add their own NO_x and knock models. Explicit knock model is not used in the current study: the investigation in this paper interprets the propensity for knock on the simple basis of comparing the in-cylinder mixture temperature at SOC and the peak in-cylinder temperature with the baseline turbocharged engine values. A more refined combustion model, NO_x and knock models will be considered in the next step of research. Such models will take into consideration of the pressure effect, which is neglected in this study. For the cylinder wall heat transfer, a Woschini heat transfer model is used.

The baseline engine is a Nissan SR20DET turbocharged gasoline engine. The parameters of the baseline engine are listed in table 1. The baseline engine is a four-stroke four-cylinder engine. Each cylinder has two intake valves and two exhaust valves. The displacement of this engine is 2.0 liter. The NA version of this engine is the Nissan SR20DET stripped off all turbocharger and intercooler related components/accessories, and the compression ratio of which is increased to 9.5 (from its original 8.5).

Table 1 Parameters of the baseline turbocharged SI engine

Parameter	Unit	Data
Engine Type		SR20DET DOHC 16V Turbo
Intake Valve Seat Diameter	mm	34.0~34.2
Exhaust Valve Seat Diameter	mm	30.0~30.2
Exhaust Valve Opening	Crank Angle Degree (BBDC)	53
Exhaust Valve Closing	Crank Angle Degree (ATDC)	7
Intake Valve Opening	Crank Angle Degree (BTDC)	-6
Intake Valve Closing	Crank Angle Degree (ABDC)	66
Throttle Body Bore	mm	60
Bore	mm	86
Stroke	mm	86
Connecting Rod Length	mm	136.3
Compression Ratio		8.5
Rated Power	kW	152.87
Rated Speed	rev/min (rpm)	6000
Firing Order		1-3-4-2
Compressor Type		T-25, 60 trim, 56mm BCI-1
Turbine Type		T-25, 62 trim, 53.8mm 0.64 A/R turbine housing
Ignition Timing-basic[static]		15 ± 2 °CA BTDC @ 800 rpm

MODELING RESULTS AND DISCUSSIONS

All results in the following are obtained with both the in-cylinder mixture temperature at SOC at full load (or called WOT) and the peak in-cylinder temperature at full load are kept at or below those respective temperatures of the baseline turbocharged engine at all rpm. Also the turbine entry temperature at full load is kept at or below the temperature of the baseline turbocharged engine at rated rpm.

MODELING RESULTS OF TURBO-COOL WITH BOTH EXHAUST VGT AND AIR TURBINE VGT

Based on the previous study [5], modeling of Turbo-Cool with both exhaust VGT and air turbine VGT and optimization of the mass flow distribution in the two paths of Turbo-Cool unit and optimization of the effectiveness