

Simulation-based Assessment of Various Dual-Stage Boosting Systems in Terms of Performance and Fuel Economy Improvements

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ABSTRACT

Diesel engines have been used in large vehicles, locomotives and ships as more efficient alternatives to the gasoline engines. They have also been used in small passenger vehicle applications, but have not been as popular as in other applications until recently. The two main factors that kept them from becoming the major contender in the small passenger vehicle applications were the low power outputs and the noise levels.

A combination of improved mechanical technologies such as multiple injection, higher injection pressure, and advanced electronic control has mostly mitigated the problems associated with the noise level and changed the public notion of the Diesel engine technology in the latest generation of common-rail designs. The power output of the Diesel engines has also been improved substantially through the use of variable geometry turbines combined with the advanced fuel injection technology. However, recent trend in automotive industry suggests that the dual-stage boosting is also essential to further improve the power output of the Diesel engine to the level comparable to that of the other engine technologies. The advantage of the dual-stage boosting system over the single-stage system is the increase in the rated output while simultaneously improving the steady-state torque at low engine speeds and the transient response of the Diesel engine by rapidly building up boost pressure.

In this study, several different types of dual-stage boosting systems are evaluated with a physics-based

zero-dimensional Diesel engine system simulation in terms of their steady-state and transient performance characteristics and fuel economy improvements. The dual-stage boosting systems evaluated in the study include a boosting system with two fixed geometry turbochargers at both the high pressure and low pressure stages, and several hybrid boosting systems in which the high pressure turbocharger is replaced with a screw type supercharger, an electrical compressor, and a variable geometry turbocharger. A dual-stage boosting system with early intake valve closing (EIVC) strategy is also evaluated. Each alternative system exhibits unique tradeoffs and improvements in terms of performance and fuel economy compared to the dual-stage boosting system with fixed geometry turbochargers at both high and low stages.

INTRODUCTION

Dual-stage Turbocharging

A dual-stage boosting system has several advantages compared to a single-stage boosting system. The most prominent advantage is its ability to achieve very high BMEP through very high intake boost pressure [1, 2]. By configuring two turbochargers in series, good efficiency can be retained even at extremely high boost pressure. The limiting factor in this case is the peak cylinder pressure. Therefore, the intake manifold pressure must be regulated in order to keep the peak cylinder pressure from exceeding its mechanical limit.

Another advantage of dual-stage turbocharging is its improved transient behavior due to the combination of a small high pressure turbocharger and a much larger low pressure turbocharger [3]. The small turbocharger at the high pressure stage reacts very rapidly to the driver's request because of its smaller inertia and flow characteristics and the large turbocharger at the low pressure stage follows up as the air flow increases.

Choi et al. [4] shows much improved transient behavior of their engine with two-stage turbocharger compared to that with VGT as well as increased full load torque and emissions reduction. The increase in BSFC is very modest and in some cases even better BSFC is reported.

However, as the flow rate reaches the flow capacity of the smaller high pressure turbocharger, the flow becomes choked and the backpressure increases excessively. Thus, the flow must be bypassed, or the size of the high pressure turbocharger must increase to accommodate the flow and the transient behavior must be compromised somewhat.

The disadvantages of dual-stage turbocharging are additional cost, complexity and bulk, and poor part load performance at low speed [3]. The lower pressure ratio per stage compared to the single-stage system at a given part load condition, combined with the low flow rate results in less efficient operation of the dual-stage system. Therefore, the important requirements are to reduce complexity and bulk of the system by using minimum number of components, and to improve performance at part load low speed condition as well as at full load high speed condition by optimizing its matching.

An idea of adding another intercooler between high and low pressure compressors was also considered for higher efficiency but discarded since the savings in the compression work of the high pressure stage was not enough to offset the corresponding higher pressure losses in the circuit [5].

Hybrid Dual-Stage Boosting Systems

A hybrid dual-stage boosting system is a boosting system that consists of a number of different types of air charging devices connected in a series configuration. Such devices include exhaust-driven turbochargers, mechanically driven superchargers, and electrically driven superchargers. Although it is preferable to use exhaust driven turbochargers whenever possible if the ultimate efficiency is the main interest of building a dual-stage boosting system, they do not always provide sufficient boost over the entire operating range of an engine due to the non-linear boosting characteristics of centrifugal compressors.

At low speed, turbochargers do not deliver sufficient boost as the centrifugal compressors require substantially high rotational speed in order to produce meaningful boost. This is where the other types of charging systems can enhance the overall boosting characteristics of the system. For example, a mechanically driven supercharger connected in series with an exhaust driven turbocharger can provide strong boost even at low engine speed taking advantage of its direct connection to the output shaft of the engine. The combined effect of the boosting system is more consistent boost over the entire operating range of the engine at the cost of slightly less overall efficiency.

Electrically driven superchargers are even better in this regard since they are completely decoupled from the engine and have the freedom to spin at any rate regardless of the engine operating condition. With this feature, the hybrid boosting system can become an effective method not only to increase torque at low engine speed, but also to reduce smoke by ensuring sufficient air volume. However, the electric motor and power supply system are the limiting factors [6, 7]. The battery and the generator cannot supply sufficient power for the electric compressor to maintain air flow as the engine displacement volume increases.

In general, the hybridization of the boosting system results in fewer compromises in the boosting system matching. With the hybrid components enhancing the low speed performance, the turbocharger needs to perform efficiently only within the high speed range, which in turn improves the overall efficiency of the system.

The drawbacks of these types of hybrid systems include higher cost, complexity of the boosting system matching, and compatibility with the EGR system. With the hybrid dual-stage boosting system, the exhaust manifold pressure is often lower than the intake manifold pressure as there is only one turbine at the exhaust side but there are two compressors in intake side. Therefore, it is necessary to route the EGR feed line into upstream of the compressors. This may lead to the component fouling and the reduction of the component durability due to the particulate contents and the corrosive compounds in the exhaust gas [8].

Dual-Stage Boosting System Configurations

DUAL-STAGE TURBOCHARGER WITH BYPASS VALVE - Figure 1 shows the dual-stage turbocharging system with a bypass valve. In this configuration, two turbochargers are connected in series and a bypass valve is placed across the high pressure turbine to limit the boost by bypassing part of the exhaust gas from the exhaust manifold to the larger low pressure turbine. The low pressure turbine is much larger than the high pressure turbine in order to avoid excessive shaft speed at high engine speed when the exhaust gas needs to be bypassed.

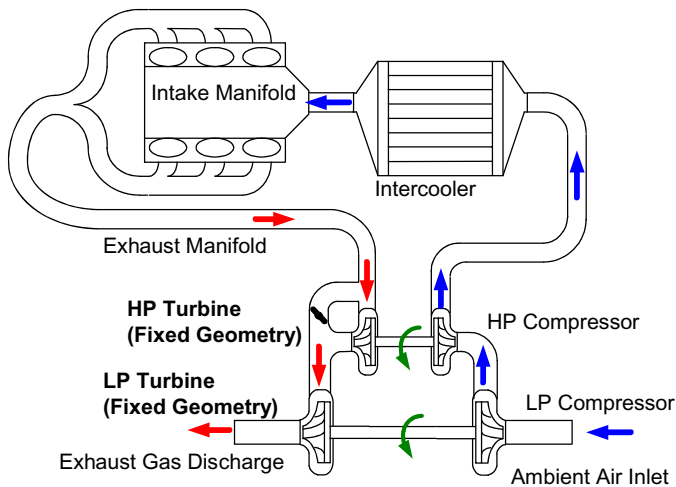


Figure 1. Dual-stage turbocharging system

A very small turbine can be used at the high pressure stage as the bypass valve opens up at high engine speed and prevents turbine choking. At low engine speed, the smaller turbine provides more power to the compressor wheel and as a result the compressor operates more efficiently. This helps the engine develop more torque at low speed by supplying more air.

The bypass valve can also be placed across the low pressure turbine instead of the high pressure turbine. However, this method is less effective since it requires larger turbine at the high pressure stage [9]. Even though this method allows smaller turbine at the low pressure stage, the benefit of having a smaller low pressure turbine is not enough to offset the shortcomings of a larger turbine at the high pressure stage as the performance of the boosting system is more sensitive to the size of the high pressure turbine.

HYBRID BOOSTING SYSTEM WITH SCREW TYPE SUPERCHARGER – The most appealing asset of a supercharger is its simplicity. It is easy to install and doesn't require any complicated control in most cases. It draws power through mechanical connection to the crankshaft, and its rotational speed is directly proportional to the rotational speed of the engine. It is this linearity that makes designing and predicting its boosting characteristics relatively easier than turbocharging. However, superchargers are inherently less efficient since they consume considerably more power than turbochargers [10].

The most common types of superchargers are the roots blower and the Lysholm compressor. They are also known as the roots type supercharger and screw type supercharger. Both are displacement pumps. The screw type supercharger has two rotors, a male and a female, forming a set of chamber between themselves and the housing. The volume of the chamber changes during the rotation and thus compresses the air that is trapped inside the chamber, which enables high pressure ratio and highly efficient supercharging even at low speed. On

the other hand, this requires the shapes of the two male and female rotors to be very complex and difficult to manufacture. Thus, the cost of the screw type compressor is considerably higher than that of the roots type compressor, making it less common.

Unlike the screw type supercharger, the roots type supercharger has two similar rotors forming discrete volumes but with constant volume. Modern roots type superchargers have three lobes instead of two, and twisted lobes have become more common.

The screw type supercharger has higher adiabatic (η_{ad}) and total (η_{total}) efficiency than the roots type supercharger. Takabe et al. [11] shows $\eta_{total} = 70\%$ for the screw type supercharger and $\eta_{total} = 50\%$ for the roots type supercharger, in spite of the lower mechanical efficiency of the screw type supercharger due to the higher step up gear ratio. The difference in efficiency comes from their distinctive compression mechanisms. Unlike the near adiabatic internal compression of the screw type supercharger, for the roots type supercharger, the compression takes place at the outlet as the air exits the blower. This condition is more comparable to the isochoric condition (constant volume process) than the adiabatic condition, thus resulting in lower efficiency and higher heat loss.

However, the actual efficiency of the screw type supercharger varies with engine operating condition. In order to fully utilize the higher efficiency of the screw type supercharger, the charge pressure at the compressor outlet has to be same as the intake manifold pressure. Otherwise, the discharged air must go through either isochoric compression or isochoric expansion inside the intake manifold, which is known to be less efficient. As a result, the overall efficiency becomes somewhere between adiabatic and isochoric efficiency.

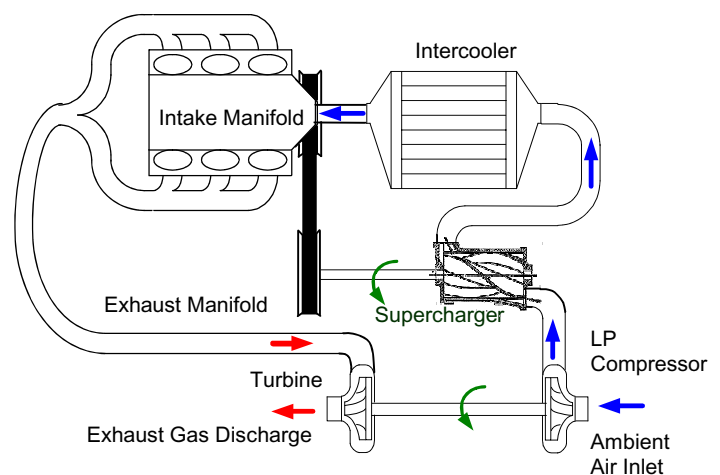


Figure 2. Hybrid dual-stage boosting system with mechanically driven screw type supercharger

The hybrid dual-stage boosting system with a screw type supercharger is illustrated in Figure 2. In this configuration, the turbocharger at the high pressure stage is replaced by a mechanically driven screw type supercharger.

DUAL-STAGE TURBOCHARGING SYSTEM WITH EIVC – With an EIVC strategy, the intake valve closes part of the way through the induction stroke to prevent any further air from entering the cylinder, thus restricting the trapped air mass. Once the intake valve has closed, the pressure inside the cylinder will decrease as the piston moves toward the bottom dead center (BDC), but the work done by the piston to expand the air will be recovered at the compression stroke as the trapped air will act as a gas spring. This also makes the effective compression ratio lower than the geometric compression ratio. In SI engine application, an EIVC strategy offers the possibility of throttleless operation throughout the engine speed and load range, and considerable improvement in fuel economy can be achieved by eliminating or reducing throttle [12].

In Diesel engine application, the EIVC strategy can be used as a mean to reduce the peak cylinder pressure. Since considerable portion of the compression is carried out in the compressor and the charge is cooled afterward by intercooler, the final temperature of the charge reached after the compression stroke will be lower than that of the charge compressed only in the cylinder [13]. This results in lower peak cylinder pressure and temperature during the combustion process. The EIVC cycle can also reduce NOx emissions when it is applied to Diesel engines with high boost level [5].

With the lower peak cylinder pressure, the injection timing can be advanced more compared to the conventional engine, which leads to better overall system efficiency. The overall thermal efficiency of the engine is also improved as a result of better utilization of the available energy in the exhaust gas as the expansion stroke is longer than the effective compression stroke. It can be also used to regulate the turbocharger operation by controlling the trapped air charge in the combustion chamber, thus EIVC allows the turbocharger to operate without the bypass valve.

In the EIVC cycle engine with dual-stage turbocharger, the bypass valve is eliminated and the boost is controlled by varying the intake valve closing at medium to high engine speed. At the engine speed of 2000 rpm and higher, the intake valve is closed earlier at 160° after top dead center, and the overall effect is an engine cycle with much longer expansion stroke. A simulation based iterative method was used to select the optimal intake valve timing along with careful matching of the turbocharger since the turbocharger matching and the intake valve timing affect each other. The system configuration is shown in Figure 3.

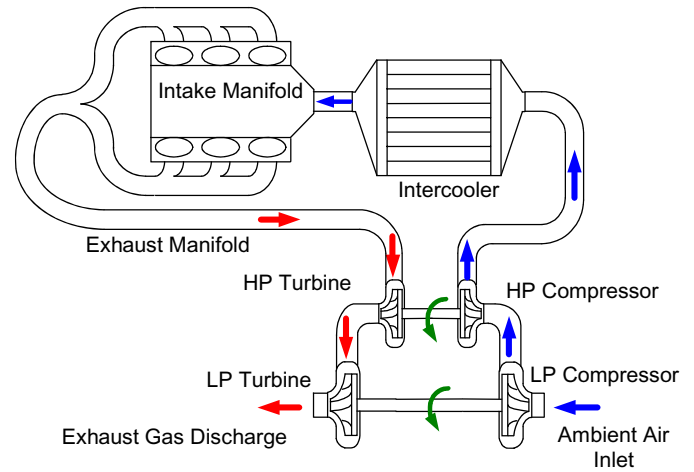


Figure 3. Dual-stage turbocharging system without bypass valve utilizing EIVC

DUAL-STAGE TURBOCHARGER WITH VGT – In a conventional dual-stage turbocharging system with fixed geometry turbines at both high and low pressure stage, it is desirable to have a small high pressure turbocharger in order to improve system transient response to sudden changes in speed and load [14]. However, the small high pressure turbocharger restricts the gas flow at higher engine speed, and typically some of the gas flow must be bypassed to prevent choking and over-boosting. However, by replacing one of the fixed geometry turbines with the variable geometry turbine (VGT), which typically has larger flowing capacity when the inlet vanes are fully opened, the bypass valve can be effectively eliminated as the boost can be controlled by adjusting the vane position of the VGT.

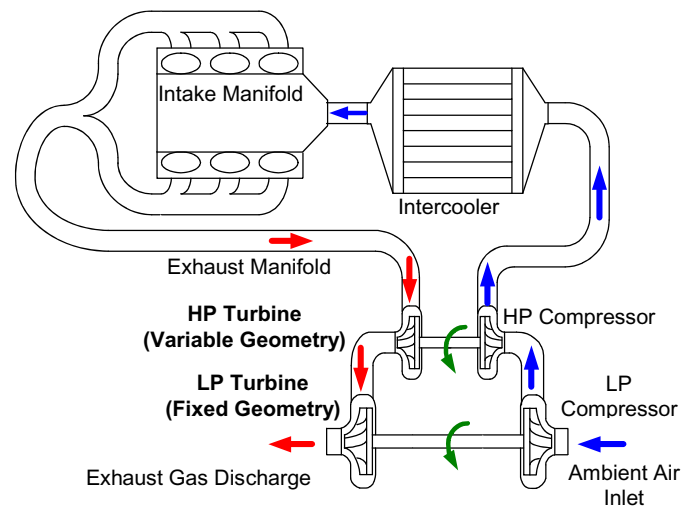


Figure 4. Dual-stage turbocharger with variable geometry turbine at the high pressure stage

In this hybrid configuration, the fixed geometry turbine at the high pressure stage is replaced by VGT as shown in Figure 4. Initial study showed that it is more effective to