

Single Sequential Turbocharger: A New Boosting Concept for Ultra-Low Emission Diesel Engines

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ABSTRACT

As turbocharged diesel engines have evolved to higher brake mean effective pressure (BMEP) levels, with ultra-low levels of regulated emissions, it has become increasingly difficult to match the turbine and compressor in conventional turbocharger and achieve the desired level of performance and responsiveness. Emissions regulations result in the use of ultra-high levels of Exhaust Gas Recirculation (EGR) to control NO_x and Diesel Particulate Filters (DPF) to control soot and particulate emissions. These technologies exacerbate the compressor/turbine matching issue by reducing the turbine corrected flow to as little as half the compressor corrected flow.

A new design concept is presented in this paper that significantly improves the performance of the turbine by changing the work/speed relationship of the compressor. This is accomplished by using a double-sided compressor and results in a secondary benefit of improved flow range of the compressor, achieved by stalling one of the compressor wheels. Due to the improved range of the compressor, the backward curvature can be reduced, which improves the pressure ratio capability.

When combined with a ball bearing cartridge, this concept results in a single turbo with a double-digit improvement of turbine efficiency, a high pressure, wide range compressor, in a small, compact package capable of providing performance unattainable in a conventional turbocharger configuration.

INTRODUCTION

U.S. emissions regulations for Medium-Duty and Heavy-Duty diesel engines starting in 2007 and phasing in fully in 2010, demand a 90% NO_x reduction compared to 2004 regulations. Simultaneously particulates must be reduced by 90%.

Particulate reduction is straightforward through the use of a diesel particulate trap. Implementation is less than straightforward due to consideration of factors associated with modeling and controlling active

regeneration. An increased and widely varying backpressure to the turbocharger turbine is the main effect on the gas exchange system.

NO_x reduction can be achieved by a number of means, mainly the use of exhaust gas recirculation (EGR), a Lean NO_x Trap (LNT), or Selective Catalytic Reduction (SCR). It is extremely difficult to achieve the full 90% reduction with any one technology; therefore a combination of at least two of the technologies will be used by most engine manufacturers. Both SCR and LNT systems generally cannot achieve more than 80% reduction in NO_x over a wide operating range, therefore EGR is used to bring the NO_x down to at least the 1 gram level where an LNT or SCR system can be used reach the regulation target.

To reach the ~1 gram level of NO_x, a substantial amount of EGR must be used—often in excess of 40% at part load and in excess of 20% at full load (for dynamometer certified engines). This level of exhaust gas recirculation has a significant impact on the gas exchange system; it negatively impacts the pumping loop work, adversely impacts the pressure ratio required by the turbocharger system to pump fresh air through the engine, and upsets the corrected flow balance of the compressor and turbine. The corrected flow imbalance results in poor turbine efficiency, which adds to the negative pumping work, further worsening the fuel consumption. It is a vicious loop!

The result is that a single turbocharger has difficulty meeting the demand of the engine to supply air and pump EGR efficiently, and provide transient response desired by customers.

Multi-turbo systems with complex valving can overcome these challenges but present additional challenges regarding packaging, under-hood heat rejection and cost. The thermal capacitance has a substantial impact on the essential NO_x after-treatment and DPF and can delay the heating of the after-treatment systems to the point they are not effective on the emissions test cycle.

Although the turbo-diesel engine has numerous advantages over the gasoline engine, such as fuel

economy and high torque, increased cost and complexity can reduce its competitiveness, especially in the ultra-low emissions U.S. market.

One can conclude that there is a significant market need for a turbocharger re-engineered to provide the functionality needed by these 2010 engines within the constraints of a single turbo package.

We will explore the fundamentals of this “vicious loop” and attempt to breakthrough these limitations with a new architecture.

TURBO MATCHING ISSUE

COMPRESSOR - First, let's examine the effect of EGR on the compressor. When running short loop EGR, the turbocharger “sees” the EGR as a reduction of the volumetric efficiency of the engine. This is not technically correct as the volumetric flow through the engine has not changed; however if we define another term, “fresh air volumetric efficiency”, then this becomes accurate. A 90% volumetric efficient engine running 10% EGR appears to the turbocharger as though the volumetric efficiency has dropped by 10+%. The “plus” is due to the impact of higher temperature of the EGR.

Assuming constant volumetric efficiency, lines of constant engine speed can be plotted on the compressor map as shown in Fig. 1. Additional EGR, without a change in air to fuel ratio move the operating point vertically on the map to the intersecting point with the lower volumetric efficiency line. In reality, since the engine is running lean, there is excess oxygen in the exhaust, therefore EGR brings oxygen and the “fresh air” to fuel ratio can be lowered somewhat e.g. by keeping fuel to oxygen ratio constant. This is effect is shown in Fig. 1 by the curved dashed line.

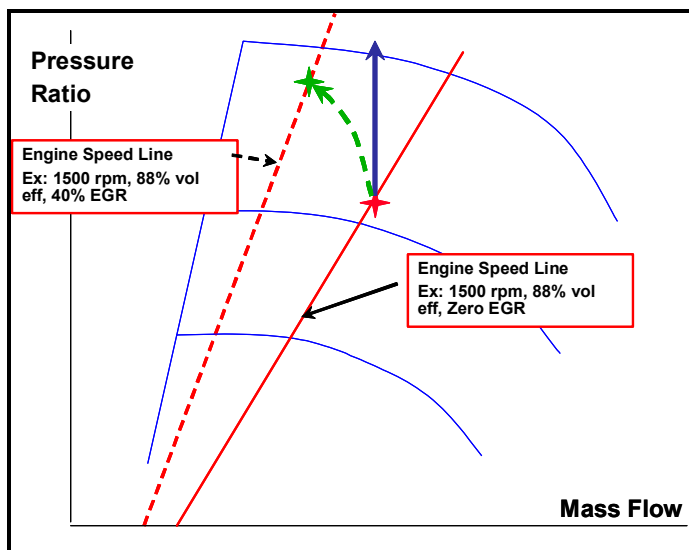


Figure 1

One can see that the operating point with high EGR is shifted to a higher pressure ratio and lower mass flow. This effect places an additional pressure ratio requirement (and thus higher speed) on the compressor. EGR is not used uniformly at all times and can be shut off entirely at very high altitudes or extreme high or low ambient temperatures. Thus the full non-EGR mass flow requirement must be met as well as the low flow/high pressure ratio point due to high EGR at low speeds. The demands of this may exceed the flow range (operating space between surge and choke) of the compressor.

To summarize, EGR raises the compressor pressure ratio (and thus speed) required as well as increasing the flow range requirement. Modest levels of EGR can be accommodated within the state-of-the-art compressor envelope—extreme levels required by 2010 engines cannot.

TURBINE – the additional pressure ratio resulting from the EGR increases the work required by the turbine to pump the fresh air and EGR through the engine. The additional work is generated through a higher expansion ratio for the turbine.

The NOx and particulate after-treatment further raises the exhaust manifold pressure. Assume for a moment that there is a 50 millibar restriction due to the after-treatment and a 3/1 expansion ratio. The exhaust manifold pressure is raised by (3*50+50) or 200 mb!

The increased expansion ratio due to EGR pumping plus the increase in exhaust manifold pressure due to after-treatment combine to significantly reduce the corrected flow of the turbine.

Mass flow quantities for compressors and turbines are generally “corrected” according to one of the following equations:

$$\dot{m}_{corr} = \dot{m}^* \sqrt{(T/T_{ref})} / P/P_{ref} \quad 1)$$

or

$$\dot{m}_{corr} = \dot{m}^* \sqrt{T/P} \quad 2)$$

Using either equation negates the impact of fluid density which essentially converts the mass flow to a volumetric basis, which is in concert with the characteristics of turbomachinery.

The result of the extreme increase in the exhaust manifold pressure is to reduce the corrected mass flow of the turbine. Looking at 2010 engine requirements, the corrected mass flow of the turbine can be as low as ½ the compressor corrected mass flow. Either an extremely low flow turbine must be designed or a smaller frame size turbine used. Before we try to resolve the combustion, let us look at some other turbine parameters.

A well known turbine design parameter is U/C_o or blade speed ratio. Fig. 2¹ from Watson and Janota shows the efficiency of a turbine as a function of the blade speed ratio.

U is the tip speed of the turbine, or the wheel circumference times the speed. C_o is the isentropic spouting velocity of the gas which is essentially the gas potential energy converted into units of velocity. The turbine inlet temperature and expansion ratio of the turbine are the key variables—note that they are size and flow independent.

$$U/C_o = (D \cdot N \cdot \pi) / \sqrt{2 c_p T_{01} [1 - (P_s/P_{01})^{(\gamma-1)/\gamma}]} \quad 3)$$

where

- N turbo speed
- D turbine diameter
- T_{01} turbine inlet temperature
- P_s turbine inlet pressure
- P_{01} turbine exit pressure
- c_p specific heat at constant pressure
- γ ratio of specific heats

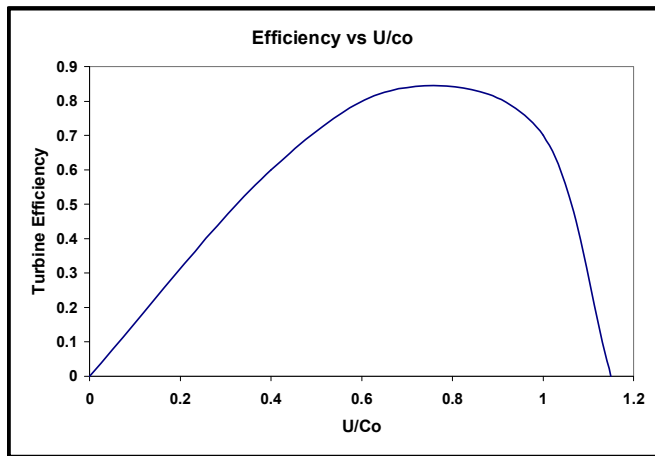


Figure 2

Turbines for gas turbine engines are designed to operate at the optimum blade speed ratio, however turbochargers are generally not. Turbochargers are required to be highly dynamic and quick transient response is essential so the diameter of the turbine is often significantly smaller than optimum. Turbines designed for the best blade speed ratio through the operating range of the engine would be efficient but sluggish.

The flow of a turbine is proportional to the square of the diameter and the inertia is proportional to the diameter to the 4th power. During the formative years of automotive turbocharger, it didn't take a rocket scientist to figure out the optimum response comes from the smallest, highest specific speed turbine possible.

The result of decades of industry development down this path is shown in Fig. 3. The blade speed ratio across the lug line for a typical VNT turbocharger running on a Light Truck diesel is shown. Combining Fig. 2 and 3, one can see that the efficiency of the turbine at low engine speeds will be poor, resulting in poor transient response as well as a limited ability to drive EGR and achieve the target air fuel ratio.

If the turbine could be redesigned to have a higher blade speed ratio, the lug line could be balanced about the peak of the blade speed ratio curve shown in Fig. 2.

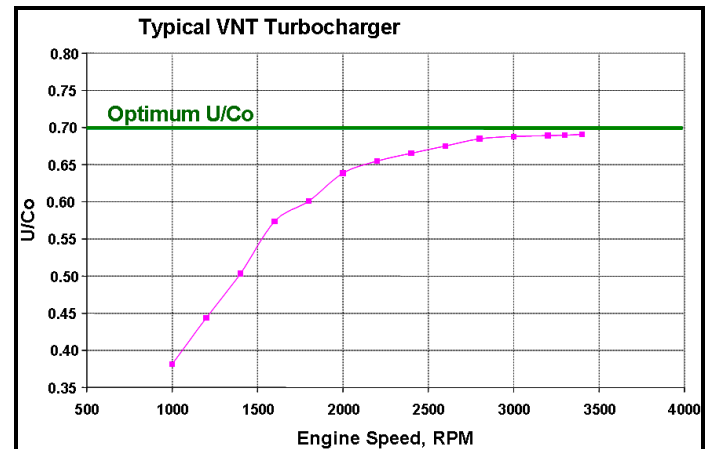


Figure 3

Referring back to equation 2), C_o is fixed by the engine operating condition (exhaust temperature and expansion ratio), thus the tip speed of the turbine must be increased to achieve a higher blade speed ratio. The compressor controls the speed that the turbo must operate at so the only independent variable is the diameter of the turbine.

Accordingly, to increase the blade speed ratio, we must increase the diameter of the turbine. However, earlier we determined the corrected flow of the turbine was substantially reduced which would indicate a decrease in size. It appears there are conflicting requirements for the turbine designer!

Perhaps we should revisit the compressor since it controls the turbo speed. To increase turbo speed for a given mass flow and pressure ratio demand by the engine, the compressor could be downsized while maintaining the inducer area, which sets the maximum flow. However, compressors are generally matched near the maximum trim (inducer area) to achieve maximum efficiency, so very little downsizing can be done.

Another possibility is to increase the backward curvature of the compressor. This reduces the pressure coefficient of the wheel and improves the flow range. However, many wheel designs already have 30-45 degrees of backward curvature, which limits the potential of this design change. Additional backward curvature also