Effectiveness of Mechanical Turbo Compounding in a Modern Heavy-Duty Diesel Engine

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ABSTRACT: Over the years, multiple engine manufacturers have offered mechanical turbo compounding on some engine models as a means to improve the efficiency of diesel engines. In theory, converting exhaust energy which would otherwise be wasted to shaft work seems like a technical path that no manufacturer should ignore especially in today’s world with the uncertainty of future CO2 emission regulations or fuel economy standards. This paper offers an independent examination of the effectiveness of the most recent implementation of mechanical turbo compounding on the Detroit Diesel (Daimler) DD15. In order to evaluate the effectiveness, an analysis procedure was developed for computing the tradeoff between power generated by the power turbine and losses incurred due to higher exhaust back-pressure as a result of the power turbine with consideration to the requirement for some minimum exhaust pressure to drive EGR.

KEY WORDS: (Standardized) Heat Engine, Performance, Turbocharger, Turbo-compound [A1]

1. INTRODUCTION

In the heavy-duty truck market, fuel consumption has always been a top priority. The global focus on climate change and CO2 emissions has placed additional emphasis on reducing fuel consumption for all vehicles, in particular diesel heavy-duty trucks which consume a significant fraction of fuel in the transportation sector. In the United States, implementation of new diesel engine exhaust emission standards in 2007 and 2010 has made it more difficult to improve fuel economy for new trucks due to the trade-offs required for low engine out emissions.

Improvements in the combustion process have been a key in maintaining or improving fuel economy and development engineers have also looked at other areas for improvements such as reducing friction, improving turbocharging, and exhaust energy recovery. One method of recovering exhaust energy is the use of a power turbine coupled either mechanically or hydraulically to the engine gear train, or to an alternator for generating electrical power (1, 2). At the present time, one of the few examples of turbo-compounding in a production engine is the Daimler (Detroit Diesel) DD15 engine which uses an axial turbine mechanically coupled through gearing and a fluid coupling to the engine crankshaft.

This paper describes an analysis procedure for determining the net benefit of turbo-compounding on engines using high-pressure loop EGR and the results of this analysis for the Daimler (Detroit Diesel) DD15 engine.

2. DAIMLER (DETROIT DIESEL) DD15 ENGINE

The test engine was a model year 2008 DD15 with the specifications shown in Table 1. The engine has a high-pressure loop, cooled, exhaust gas recirculation (EGR) system and a fixed geometry turbocharger with no wastegate. In series with the turbocharger, there is a close-coupled axial flow turbine which is connected via a fluid coupling and gear train to the engine crankshaft. A photograph of the axial turbine and turbocharger arrangement is shown in Figure 1.

The addition of the axial turbine in series with the turbocharger increases the exhaust back pressure on the turbocharger and increases the exhaust manifold (pre-turbine) pressure (PTP). The increase in exhaust manifold pressure provides a negative pressure gradient across the cylinder head to drive EGR (intake -exhaust). The large negative pressure gradient can be expected to increase the pumping losses (PMEP). However, since the axial turbine does extract some energy from the exhaust gas and transfers most of this energy to the crankshaft, does the power produced offset the losses in PMEP? In other words, is the “Net PMEP” (PMEP + effective MEP from Axial turbine) a gain or a loss?
Table 1. DD15 Engine Specifications

| Specification          | Value                  |
|------------------------|------------------------|
| Cylinder Arrangement   | In-Line 6              |
| Bore                   | 139 mm                 |
| Stroke                 | 163 mm                 |
| Displacement           | 14.8 L                 |
| Compression Ratio      | 18.4:1                 |
| Rated Power            | 354 kW (475 hp)        |
| Rated Speed            | 1800 rpm               |
| Peak Torque            | 2238 N-m (1651 lb-ft)  |
| Peak Torque Speed      | 1240 rpm               |
| Injection System       | Amplified Pressure Common Rail Fuel System (ACRS) |
| Turbocharger           | Fixed Geometry, with Turbo-compounding |
| Exhaust Treatment      | DOC and DPF            |
| Exhaust Gas Recirculation (EGR) | High-Pressure Loop, Cooled, Hot Side EGR Control Valve |
| Emission Regulation    | 2007 US Heavy-Duty Federal Test Procedure |
| Emission Levels        | NOx FEL = 1.55 g/kW-hr NMHC = 0.19 g/kW-hr PM = 0.013 g/kW-hr CO = 20.85 g/kW-hr |

3. COMPARISON OF DD15 TO OTHER CLASS 8 TRUCK ENGINES

Other engines that have been benchmarked at SwRI include the 2007 Volvo D13, 2007 Cummins ISX, and the 2007 Caterpillar C15. The selected engines each had similar ratings of about 360 kW. The D13 and ISX engine have a high pressure loop EGR system with a variable geometry turbocharger to control the exhaust manifold pressure to drive EGR. The Caterpillar C15 engine was unique in that it had a low-pressure loop EGR system with two-stage turbocharging. These engines, along with the DD15, are representative of engine offerings for the U.S. Class 8 Truck market. A comparison of the cylinder head delta pressures at rated power and at a “cruise-like” condition are shown in Figure 2 for these engines. The DD15 engine, due to the turbo-compounding, has significantly higher $\Delta P$ across the cylinder head at rated power. At the part load “cruise” condition, the $\Delta P$ was lower than the D13 and ISX and similar to the C15. The high $\Delta P$ at rated for the DD15 translated directly into increased pumping losses relative to the other engines as shown in Figure 3. At the “cruise” condition, the PMEP’s were similar.

4. PERFORMANCE EVALUATION RESULTS FOR THE DD15

As noted above, the DD15 engine has a significant negative pressure across the cylinder head at rated power. The fixed geometry turbocharger in series with the axial flow turbine has the characteristic of increasing back pressure as volumetric exhaust flow increases. Thus, at higher speeds and loads one can expect the cylinder head differential pressure (intake manifold – exhaust manifold pressure) and the PMEP to increase. These parameters are shown in Figures 4 and 5, respectively, for the speed-torque operating map. As shown in the figures, both cylinder head $\Delta P$ and PMEP losses increase at higher engine speed and load. Of course this is also the region of the map where the axial turbine would produce useful power due to the available exhaust pressure and temperature.
The ratio of the axial turbine speed to the engine speed is one indicator that the axial turbine is producing power. Since the axial turbine is connected to the crankshaft via a fluid coupling and gear train, there is some slip in the fluid coupling and the turbine and engine speeds are related by the slip ratio and the gear ratio. If the axial turbine speed is greater than the engine speed multiplied by the gear ratio, the turbine is producing some power. Conversely, if the speed ratio is less than 1, the engine is providing power to the axial turbine. Figure 6 illustrates the range of operation where the axial turbine was producing power, above the contour label as 1.0. There was a region just below 1200 rpm where EGR is reduced and additional mass flow was therefore added to the exhaust which created a small peninsula where the slip ratio was above 1. So in the region where the turbine-engine speed ratio was greater than 1, the power of the axial turbine can be computed from the inlet and outlet temperatures and pressures, the fluid properties and the exhaust flow rate. Often an assumption can be made that the heat loss from the turbine housing can be neglected; however, in this case and elsewhere in the literature, since the expansion ratios are relatively small the heat loss can be a significant factor. The axial turbine expansion ratios are shown in Figure 7.

The heat loss from the turbocharger and axial turbine was estimated by the following:
1. Computing the overall turbocharger efficiency
2. Computing the compressor efficiency
3. Solving for the turbine efficiency (including mechanical efficiency)
4. Using the equation for turbine efficiency and the value computed in step 3, solve for the turbine exit temperature.
5. An estimate of the heat loss is represented by the difference between the temperature computed in step 4 and the measured temperature multiplied by the mass flow rate and specific heat.

6. A correlation for the heat loss as a function of the turbine inlet temperature was developed.

7. This correlation was used to estimate the heat loss from the axial turbine and used to adjust the turbine power and efficiency computations.

The axial turbine power is presented in Figure 8. As shown, the turbine power increased with increasing speed and load as expected, however, over much of the engine operating map the turbine produced little or no power. For comparison to the PMEP, the axial turbine power can be converted to the equivalent MEP by dividing by 24.75 (kW/bar). Using this conversion, one can see that the power turbine produces up to 1 bar in terms of MEP. The axial turbine benefit must be considered together with the additional pumping work that it imposes on the engine. This tradeoff is somewhat difficult to estimate since some negative pressure gradient across the cylinder head is required to drive EGR. In theory, if the engine was using a variable geometry turbocharger to control the pressure gradient to drive EGR, the optimum ΔP could always be obtained and the pumping losses minimized. In practice, since the DD15 is using a fixed geometry turbocharger, when the cylinder head ΔP is in excess of that required to drive EGR, the EGR valve must close, as shown in Figure 9, to get the desired EGR flow and the higher ΔP incurs a PMEP penalty. This marginal PMEP penalty (above that required for EGR) versus the power derived from the axial turbine is then the tradeoff that must be considered.

In order to assess this tradeoff, an estimate is required for the minimum cylinder head ΔP required to drive the desired amount of EGR. For the DD15 engine, the EGR flow rate was measured for each operating condition along with the EGR pressure, temperature, and ΔP across the EGR valve. This information along with the EGR valve position was used to develop an EGR orifice flow equation. Using this equation, the effect of opening the EGR valve for a fixed EGR flow rate on the required EGR ΔP could be determined. By accounting for the pressure drop across the EGR valve, cooler, and measurement venturi, the optimum exhaust manifold pressure can be computed from the measured intake manifold pressure. The difference between the optimum exhaust manifold pressure and the intake manifold pressure, (i.e. the optimum cylinder head ΔP), along with the engine speed was used to compute the minimum required PMEP for driving EGR using a curve fit of the measured PMEP data.

Figure 10 illustrates the analysis process for the most favorable speed and load condition. Shown in Figure 10 is the observed PMEP at the 1400 rpm, 100-percent load condition. An estimated minimum PMEP assuming the optimum cylinder head ΔP was computed as described above. The observed PMEP can be divided up into the minimum PMEP and the PMEP penalty for having excess exhaust back pressure. This PMEP penalty can be compared to the effective MEP recovered from the axial turbine (accounting for gear train losses which were assumed to be 5-percent). If the net MEP is positive, the axial turbine provides a net benefit, a negative remainder would indicate a net loss. As shown in Figure 10, the net PMEP accounting for the minimum ΔP and the axial turbine was positive for this operating condition. At a positive 0.5 bar MEP, the power turbine contributes about 2.5 percent of the overall engine power and would produce a similar improvement in fuel consumption.
5. SUMMARY

The performance and benefit of turbo-compounding on the Daimler (Detroit Diesel) DD15 engine were examined using an analysis procedure which considered both the additional power produced by the axial power turbine (corrected for heat loss) and losses that were incurred as a result of higher exhaust back-pressure. The fixed geometry turbocharger in combination with the axial power turbine creates additional backpressure on the engine which leads to additional pumping losses over and above the pumping losses that would be incurred to drive EGR. Thus when considering the net benefit of the axial power turbine on an engine using high-pressure loop EGR, the actual pumping losses and the minimum pumping loss required to drive EGR must be computed and compared to the power produced by the axial power turbine. The power produced from the turbo-compounding unit offsets the pumping losses in some regions of the operating map. Particularly, at 1400 rpm, 100-percent load, the axial turbine provides an estimated 2.5 percent increase in BMEP and a corresponding decrease in fuel consumption.

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