Investigating the Effects on the Low Speed Response of a Pressure Charged IC Engine Through the Application of a Twin-Entry Turbine Housing

Alex Kusztelan, Denis Marchant, Yufeng Yao and Yawei Wang

Abstract In this study, one-dimensional analysis using AVL Boost software has been carried out on a series of compression and spark ignition engines utilizing a manufacturer fitted single-entry turbocharger and a modified twin-entry unit, the latter adopting two symmetrical turbine housing inlet ports. The model reconstruction using AVL Boost considers parameters that accurately represent the physical engine conditions including manifold geometry, turbocharger flow maps and combustion chamber characteristics. Model validations have been made for a standard single-entry turbocharger configuration to predict the maximum engine power and torque, in comparison with available manufacturer data and analytical calculations. Further studies concentrate on engine performance comparisons between single- and twin-entry turbochargers at low engine speed conditions, typically in a range of 1000–3000 RPM. Improvements in turbine shaft speed, engine power and torque have been achieved, thus implying improved low speed engine response. This study reveals the potential commercial benefits of adopting a twin-entry turbocharger and contribution to the academic community through this additional research.

Keywords Engine simulation · Internal combustion engine · Pressure charging · Turbine housing · Turbocharging · Twin-entry turbochargers

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1 Introduction

Turbochargers have been extensively used for "engine downsizing" practices as they can largely enhance the engines power and torque output without the need of increasing the swept volume of each cylinder. However, for turbocharged downsized diesel engines, the slower response of the turbine at low engine speeds, typically in a range of 1000–3000 RPM, appears to be a common problem. Various solutions have been proposed and studied, including variable geometry turbochargers (VGT), two-stage turbocharger and turbo-compounding methods. Both Arnold [1] and Hawley [2] observed that adopting a narrow vane angle within a VGT turbine housing at low engine speeds increases exhaust flow to the impeller, thus improving the boost performance of the compressor. Recently Chadwell and Walls [3] suggested a new technology known as a SuperTurbo to overcome the slow response of a turbocharger at low engine speeds. This type of turbocharger can be coupled to a continuously variable transmission (CVT) which is directly run via the crankshaft of the engine, thus allowing the turbocharger to act as a supercharger boosting device at lower engine speeds. Similar increases in performance using turbo-compounding methods are observed by Ishii [4] and Petitjean et al. [5]. Two-stage turbocharging as discussed by Watel et al. [6] uses high and low pressure turbochargers working in series to overcome the effects of reduced exhaust pressure encountered at low engine speeds. One method which has not been fully researched is the application of a twin-entry turbocharger with two turbine inlet ports. This arrangement may lead to an improved engine response at lower engine speeds, primarily due to the separated inlet port arrangement, thus avoiding the interactions between the differently pulsed exhaust gases inside the manifold, and enhancing the energy transfer from exhaust gas to the turbine impeller. In contrast to a single-entry turbocharger, a twin-entry turbine housing (as shown in Fig. 1) will better utilize the energy of the pulsating exhaust gas to boost the turbine performance which directly increases the rotational speed of the compressor impeller. For example a four-cylinder engine with a 1-3-4-2 firing order equipped with a single-entry turbocharger and 4 into 1 exhaust manifold will produce the following conditions: at the end of the exhaust stroke in cylinder 1 (i.e. when the piston is approaching top dead centre (TDC)), the momentum of the exhaust gas flowing into the manifold will scavenge the burnt gas out of the cylinder. In the meantime in cylinder 2, the exhaust valve is already open allowing for exhaust gas to enter the manifold as well. This means that the exhaust gas from cylinder 2 will interact with the flow of exhaust gas from cylinder 1, thus affecting the energy transfer to the turbine [7]. One solution to this problem is to adopt a twin-entry turbocharger with a split-pulse manifold that keeps the differently pulsed exhaust gasses separate, thus allowing the majority of the pulsating energy of the exhaust gas to be used by the impeller. This is not only more practical and economical but also provides a potential for improvement in the reduction of gaseous emissions. Twin-entry turbochargers have been widely used in industry for large-size engines, but limited research has been undertaken for medium-sized engines. Therefore more

Fig. 1 Turbocharger cutaway highlighting the twin-entry volute geometry [8]

studies are necessary to provide further insight into the key benefits, or otherwise, of adopting a twin-entry turbocharger as shown in this study.

2 Analysis of Experimental Engine Models

2.1 Engine Model

A commercially available downsized four-cylinder Renault 1.5L compression ignition (DCi) engine is used as a base engine for the 1-D simulation. The engine is fitted with a single-entry turbocharger as part of its standard specification. This factor is beneficial as a crucial aspect of the experimental criteria involves an analysis of a standard engine and the same engine equipped with a twin-entry turbine housing utilizing the same trim and area ratio. Table 1 gives the key parameters of the model required by the AVL Boost code [9]. It is worthwhile to point out that the purpose

Table 1 The key engine parameters as required by the AVL Boost simulation code

Parameters	1.5L DCi	2.0L CI	1.8L SI
Bore	76 mm	85 mm	81 mm
Stroke	80.5 mm	88 mm	86 mm
Exhaust valve lift	8.6 mm	5.0 mm	9.3 mm
Inlet valve lift	8.0 mm	4.6 mm	7.67 mm
Compression ratio	17.9:1	18:1	9.5:1
No. of cylinders	4	4	4
Valves per cyl.	2	2	5

of choosing this type of engine is to fulfill the current trend of engine downsizing as frequently cited in engine technology international [10]. Two further engines are also modeled using the AVL Boost code to evaluate whether the same observations could be met due to the application of a twin-entry turbocharger. These include a Peugeot 2.0L CI engine and Audi 1.8L SI engine respectively.

2.2 Engine Boundary Conditions

For the purpose of this study, the engine modeling is based on 100 simulation cycles using variable operating conditions of engine speed ranging between 1000 and 5000 RPM. Both the exhaust and the inlet valve lift profiles and dimensions are also defined using original data from the manufacturer to provide realistic operating conditions of the combustion cycle. Furthermore, identical compressor geometry and flow maps are used for both single- and twin-entry turbocharger configurations. This provides more accurate boundary conditions as the flow characteristics of the compressor will only be affected by the differences in turbine inlet and exhaust manifold geometry. It is essential to maintain the same compressor housing in order to derive accurate and convincing conclusions. In order to model engine operating conditions, the intake and the exhaust piping lengths and diameters of the physical engine are directly measured and replicated within the software. In conjunction with the Vibe combustion model, the Woschni heat transfer model and the Patton et al. friction model [11] are used to define the heat transfer conditions within the combustion chamber for each simulated engine RPM stage, which allows the AVL Boost code to accurately replicate a realistic compression ignition combustion cycle within a simulation environment.

2.3 Single and Twin-Entry Turbocharger Models

Figure 2 shows the complete simulation model of the Renault four-cylinder 1.5L DCi engine with a standard single-entry turbocharger configuration. The exhaust manifold has a 4 into 1 geometry which will result in strong flow interactions and turbulent flow mixing of the pulsating exhaust gases [12]. This implies that the energy transfer from the exhaust gases to the impeller of the turbine are not optimized, thus not realizing the full potential of the engine outputs, particularly power and torque.

In order to implement a twin-entry turbine housing in Fig. 2, a modified manifold configuration is introduced with a split-pulse design. By using the known firing order (1-3-4-2) of the original engine, the 4 into 1 manifold has been changed to allow for the exhaust gases from cylinders 1 and 4 and 2 and 3 to remain separate as highlighted in Fig. 3. Therefore the software will recognize the number of exhaust to turbine housing inputs being changed to a corresponding twin-entry configuration.
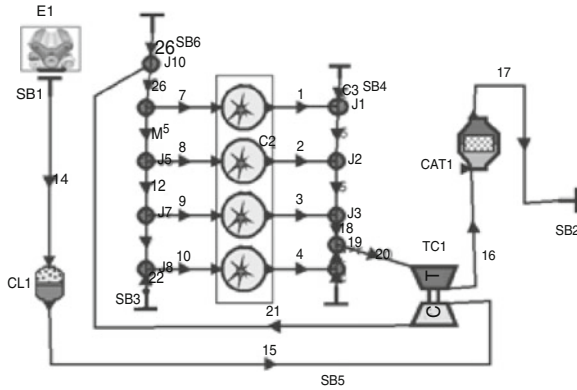
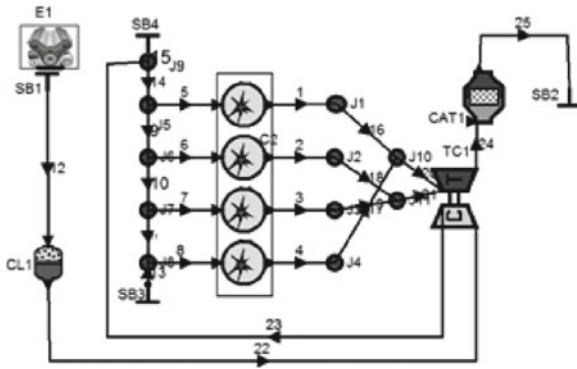**Fig. 2** AVL Boost model of the single-entry turbocharger configuration for the Renault 1.5L DCi engine

Fig. 3 AVL Boost model of the twin-entry turbocharger configuration for the Renault 1.5L DCi engine

3 AVL Boost Model Validation

Model validation has been performed using parameters of a standard Renault 1.5L DCi engine with a single-entry turbocharger and the results will be compared to those provided by the manufacturer. Data such as peak engine power of 50 kW at 4000 RPM with the BorgWarner KP35 single-entry turbocharger, and other key engine parameters as shown in Table 1, will be used.

3.1 Maximum Engine Power and Torque

The engine model was run for 100 simulation cycles using the parameters described above. To validate the model engines the performance results are compared to those
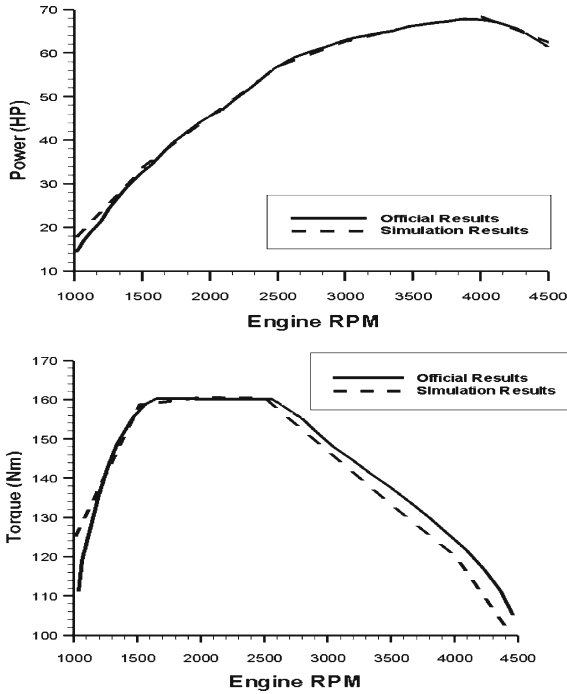
Fig. 4 AVL Boost simulated power and torque results in comparison with published manufacturing data of the 1.5L DCi Renault engine

published by the engines manufacturer as shown for the Renault 1.5L DCi engine in Fig. 4 displaying power and torque output results as a function of engine speed in a range of 1000–4500 RPM. It is clear that the simulated model Renault engine has produced very accurate predictions of peak power and torque values at an engine speed of 4000 and 1500 RPM respectively, when compared to the manufacturer's data. Similar model validations were performed for the Peugeot 2.0L CI (see Table 2) and Audi 1.8L SI engines (see Fig. 5). For the 1.8L SI engine however the torque results exhibit some discrepancies between the simulation and the manufacturer data, particularly the torque curves. This is likely to be attributed due to the inaccuracies of the combustion shape parameter which specifies the combustion characteristics within each cylinder in the AVL Boost simulation code. These characteristics are constantly changing within a running physical engine which means that a fixed number, as specified with the software cannot accurately represent a complete combustion definition.

Table 2 shows that there is only a 1.4 % and 2.3 % increase in peak power and torque results for the simulated 2.0L CI Peugeot engine indicating that the AVL Boost code has accurately re-produced the operational condition of the 2.0L compression ignition engine.

Table 2 Manufacturer and simulated data acquired by the Boost code for the Peugeot 2.0 L compression ignition engine [13]

	Simulation results	Official data	Error in %
Max power	68 kW@4000 RPM	67 kW@4000 RPM	1 kW [1.4 % error]
Max torque	220 Nm@2000 RPM	215 Nm@2000 RPM	5 Nm [2.3 % error]
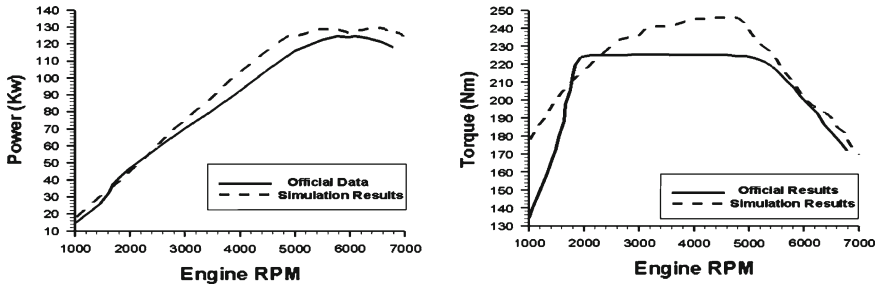
Fig. 5 Comparison between simulated engine and torque results for the Audi 1.8 L SI engine to the data specified by the engine manufacturer [13]

The validation results acquired from the three engines clearly indicate that the AVL Boost 1-D simulations have achieved reliable results considering that combustion, thermodynamic and heat transfer models are used to simulate viable engine operation.

Based on the above validations, it was concluded that the simulated model engines equipped with the standard single-entry turbocharger are working correctly. The models can therefore be subsequently adapted to a twin-entry turbine housing as described in Sect. 2.3. A direct comparison analysis between the single- and twin-entry turbocharger configurations will be used to conduct comprehensive studies concentrating on potential changes in engine performance due to the adoption of a twin-entry turbocharger geometry. These will include engine torque, power and brake mean effective pressure (BMEP).

4 Simulation Results

The engine response at low engine speeds is an area of primary interest when analyzing the application of twin-entry turbochargers for downsized engines. A common problem for turbochargers is the response time that the turbine needs to reach sufficient impeller speeds often known as "spooling time", in order for the compressor to work effectively, i.e. to produce sufficient boost. Having a long "spooling time" means the engine is susceptible to a long time delay in responsiveness, so-called 'turbo-lag', before the effect of the turbocharger becomes effective. It was therefore decided that the engine characteristics in a range of 1000–3500 RPM would be

Fig. 6 Increased power out-
put of the Renault 1.5L DCi
engine in 1000–4000 RPM
engine speed range using a
twin-entry turbocharger
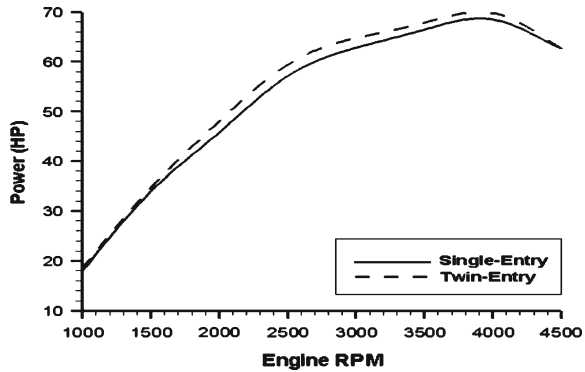
closely investigated as this is the range where the 'spooling time' and the 'turbo-lag'
have the greatest effect. It is expected that the adoption of a twin-entry turbocharger
could reduce these undesirable characteristics.

4.1 Power and Torque Outputs

The main benefit of increasing the spooling time of the turbocharger during low
engine crankshaft speeds is the improvement in time required for the compressor
to reach its optimum boost output. This implies the increase of engine power and
torque.

Figure 6 shows the comparison of power output from both single- and twin-entry
turbochargers for the Renault 1.5 DCi engine. The greatest gain in power output
for the twin-entry configuration was observed at 2500 RPM, providing approxi-
mately 2.5 HP of extra power output. This power gain is resultant from the increase
in compressor performance due to the improved energy transfer from the exhaust
gases to the turbine impeller. When calculated over the complete RPM range (i.e.
1000–4500 RPM) the twin-entry configuration produces 3.27 % greater power when
compared to the benchmark data. Figure 7 shows the comparison of engine torque
acquired from the simulation of the Renault 1.5L DCi engine. Adopting a twin-entry
turbine housing has clearly improved the torque characteristics of the engine. For
example at 2000 RPM the torque has increased from approximately 160–170 Nm.
This increase of 5.55 % @ 2000 RPM is highly favorable as the engine response
performance will have noticeably improved during the low engine speed range of
1000–3500 RPM.

Similar trends in an increase in engine power output have been revealed by the AVL
Boost simulation code where a single- and a twin-entry turbocharger comparative
analysis was performed on the Audi 1.8L SI engine as shown in Fig. 8.

The third simulation using a Peugeot 2.0L CI engine was performed using the
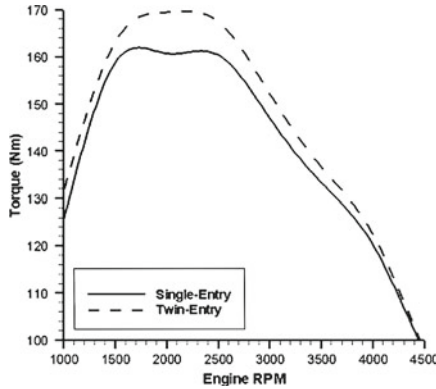AVL Boost code to further illustrate the effect of a twin-entry turbine housing on

Fig. 7 Increased engine torque output due to the adoption of a twin-entry turbocharger on the Renault 1.5L DCi engine
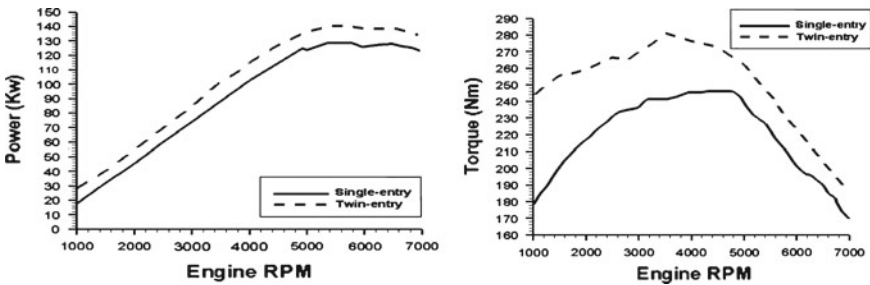

Fig. 8 Increased power and torque outputs for the Audi 1.8L SI engine using a twin-entry turbocharger [13]

the output performance characteristics of the engine. An average increase in power (7.76 %) and torque (7.52 %) calculated from 1000 to 3000 RPM are shown in Fig. 9.

4.2 BMEP Improvement

The additional air flow rate due to the twin-entry turbine configurations also causes an increase in compressor efficiency i.e. compressor discharge pressure, which not only improves the volumetric efficiency (VE) of the engine but also the Break Mean Effective Pressure (BMEP). BMEP is another important parameter used to characterize the performance of engine output and is related to torque as shown in Eq. 1.

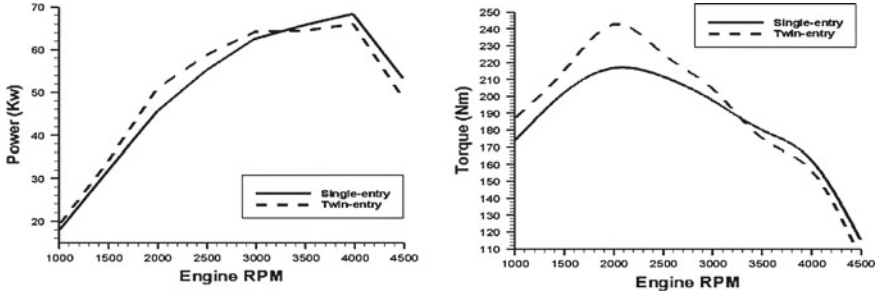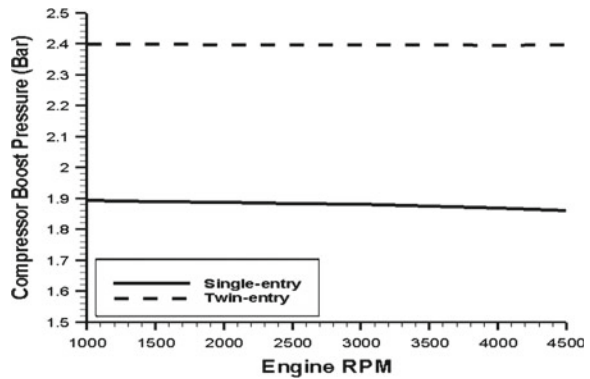$$Torque = \frac{BMEP \times Swept\ Engine\ Volume}{2\pi} \tag{1}$$

Fig. 9 Increased engine power and torque output for the Peugeot 2.0L CI engine using a twin-entry turbocharger [13]

Fig. 10 Compressor "boost" performance gain increasing the manifold pressure and therefore BMEP

It is clear from the equation that increasing the BMEP of an engine also results in increased torque characteristics, as previously shown in Figs. 8 and 9.

Figure 10 shows a comparison of compressor discharge pressure variations for both single- and twin-entry turbocharger configurations used on the 1.5L DCi engine. A clear increase in engine boost due to the twin-entry turbine housing is illustrated. The increase in pressure, although relatively small (0.5 Bar at 2500 RPM) is more favorable as any large rise in discharge boost pressure may lead to a surge condition which could deem it inappropriate for practical application. It is apparent from the discussed results that small gains in the performance of the compressor will provide an improved overall engine performance output, e.g. the increase in compressor air flow resulting in a consequently larger VE and therefore BMEP as shown in Fig. 11. A maximum BMEP increase of approximately 1 Bar is observed at 2000 RPM.

Overall, there is a noticeable increase (4.03 %) in BMEP over an engine speed RPM range of 1000–3500 RPM which is crucial for performance and response of the engine in urban driving environments. It is evident that with only a 0.5 Bar increase in compressor boost pressure the twin-entry configured engine can achieve a 1 Bar
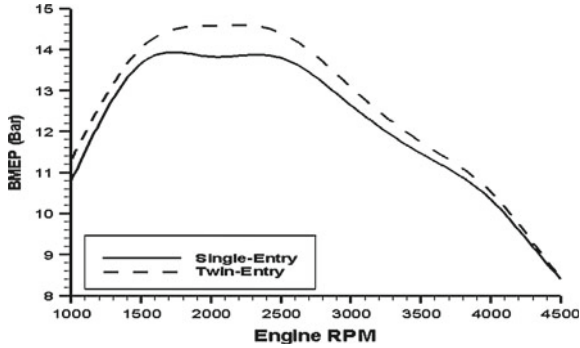
Fig. 11 Comparison between single and twin-entry engine BMEP results of Renault 1.5L DCi engine
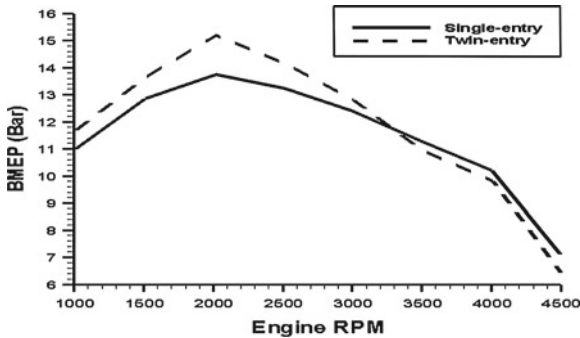

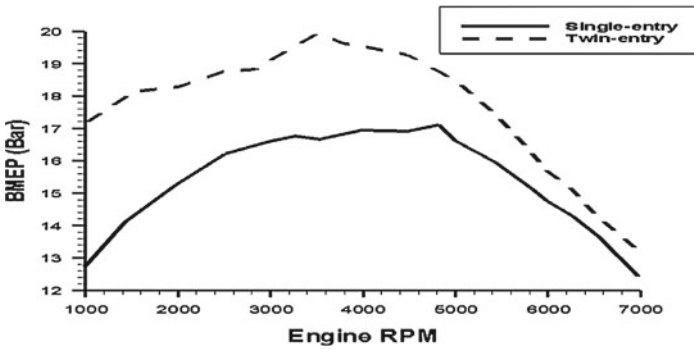**Fig. 12** A 11 % improvement at 2000RPM in BMEP for the Peugeot 2.0L CI engine [13]

Fig. 13 A 22 % improvement in BMEP exhibited by the Audi 1.8L SI engine [13]

increase in the BMEP. This therefore clearly shows that the adoption of a twin-entry turbine housing is more beneficial than a single-entry one.

An improvement in BMEP is also noted from the Peugeot 2.0L CI engine as shown in Fig. 12. There is an overall increase in BMEP from 13.75 to 15.25 Bar at 2000 RPM

Table 3 Average increase in engine performance calculated during 1000–3500 RPM due to the application of a twin-entry turbocharger

	1.5L DCi engine (%)	2.0L CI engine (%)	1.8L SI engine (%)
Power	3.99	7.76	26.46
Torque	4.04	7.52	22.23
BMEP	4.03	6.81	22.43

which equates to a 10.91 % improvement due to the addition of a twin-entry turbine housing. The simulation results acquired from the Audi 1.8L SI engine Fig. 13 also showed that the BMEP increased by 22.43 % during 1000–3500 RPM after the engine model was modified to accept the twin-entry turbocharger configuration.

A summary of the improvements exhibited by the AVL Boost simulation for all three engines is shown in Table 3.

5 Conclusions

The AVL Boost engine simulation code has demonstrated potential performance improvements on a variety of engines due to the adoption of a twin-entry turbocharger with a corresponding split-pulse manifold. The results for the Renault 1.5L DCi engine show that the application of a symmetrical twin-entry volute design enhances the performance of the engine when operating during low RPM conditions, the most effectiveness being observed from 1500 to 3000 RPM showing a maximum 4.03 % increase in BMEP. Both engine torque and power performance also increased by approximately 5.5 % at 2000 RPM resulting in an average performance increase of 4 % within the 1000–3500 engine RPM range. The addition of the extra torque and power is more beneficial during low engine speeds as the turbocharger delay time will be reduced making the engine more responsive to driver input. The "drivability" of the vehicle has therefore also improved.

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