Increases in Low Speed Response of an IC Engine using a Twin-entry Turbocharger

A. Kusztelan, D. Marchant, Y. Yao, Y. Wang, S. Selcuk, A. Gaikwad

***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 variety, the latter adopting two 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, etc. Model validations have been made for a manufacturer 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 in terms of torque, shaft speed and compressor efficiency and at low engine speed conditions typically in a range of 1000-3000 RPM. It was found that on average engine response has been increased by 27.65%, 5.5%, 5.5% in terms of turbine shaft speed, engine power and torque, respectively, which implies improved "drivability" of the vehicle. This study reveals the potential benefits of adopting a twin-entry turbocharger and the findings would be useful for both industry and academic communities.**

***Index Terms*— Turbocharging, 1-D Simulation, Twin-entry Turbochargers**

I. 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. Chadwell and Walls [3] suggested a new technology known as a SuperTurbo to overcome the slow response of a turbocharger at low engine RPM. 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 in 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 figure 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 the 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 influence 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 now been used in industry for large-size engines, but limited research has been undertaken for medium-sized engines. Therefore more studies are necessary to provide further insight into the key benefits, or otherwise, of adopting a twin-entry turbocharger as shown in this paper.

Manuscript received March 18, 2012

A. Kusztelan, PhD Student, Kingston University London, Friars Avenue, London, SW15 3DW. (e-mail: K0306918@kingston.ac.uk)
D. Marchant, Faculty Registrar, Kingston University London, Penrhyn Road, Kingston Upon Thames, Surry KT1 2EE (e-mail: D.Marchant@kingston.ac.uk)
Y. Yao, Reader, Kingston University London, Friars Avenue, London, SW15 3DW (e-mail: Y.Yao@kingston.ac.uk)
Y. Wang, Senior Lecturer, Kingston University London, Friars Avenue, London, SW15 3DW (e-mail: Y.Wang@kingston.ac.uk)
S. Selcuk, MSc Student, Kingston University London, Friars Avenue, London, SW15 3DW. (e-mail: K1048944@kingston.ac.uk)
A. Gaikwad, MSc Student, Kingston University London, Friars Avenue, London, SW15 3DW. (e-mail: K1044290@kingston.ac.uk)

Fig. 1. Turbocharger cut-away highlighting the twin-entry volute geometry, allowing differently pulsed exhaust gases to remain separate**.**

II. ANALYSIS OF EXPERIMENTAL ENGINE MODELS

A. *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 (Figure 2). 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 with the same trim and area ratio. Table 1 gives the key parameters of the model required by the AVL Boost code [8]. It is worthwhile to point out that the purpose of choosing this type of engine is to fulfill the current trend of engine downsizing as frequently cited in engine technology international [9]. Two further engines are also modeled within 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 2.0L CI engine and a 1.8L SI engine respectively.

Table. 1. The main engine parameters as required by the Boost simulation code.

	1.5L DCi	2.0L CI	1.8L SI
Bore	**76 mm**	**85mm**	**81mm**
Stroke	**80.5mm**	**88mm**	**86mm**
Exhaust Valve Lift	**8.6mm**	**5.0mm**	**9.3mm**
Inlet Valve Lift	**8.0mm**	**4.6mm**	**7.67mm**
Compression Ratio	**17.9:1**	**18:1**	**9.5:1**
No. Of Cylinders	**4**	**4**	**4**
Valves Per Cyl.	**2**	**2**	**5**

B. *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 - 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 configurations and corresponding 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 length and diameter 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 [10] 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.

Fig. 2. Renault 1.5L DCi engine for the 1-D simulation analysis (www.Renault.com).

C. *Single and Twin-entry Turbocharger Models*

Figure 3 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 into1 geometry which will result in strong flow interactions and turbulent flow mixing of the pulsating exhaust gases [11]. 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.

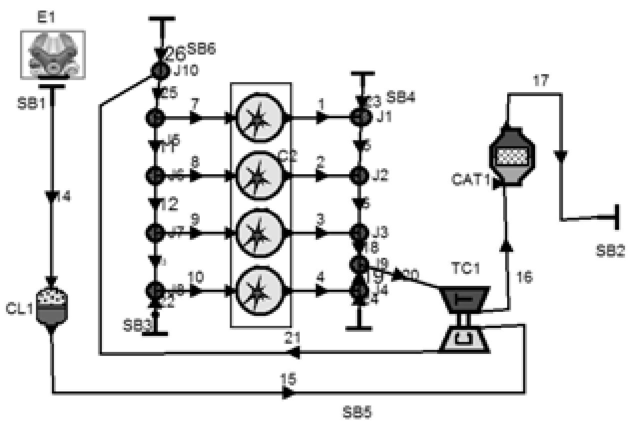

Fig. 3. AVL Boost model of a single-entry turbocharger configuration for the Renault 1.5 DCi engine.

In order to implement a twin-entry turbine housing in the above model, 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&4 and 2&3 to remain separate as highlighted in figure 4. Therefore the software will recognize the number of exhaust to turbine housing inputs being changed to a corresponding twin-entry configuration.

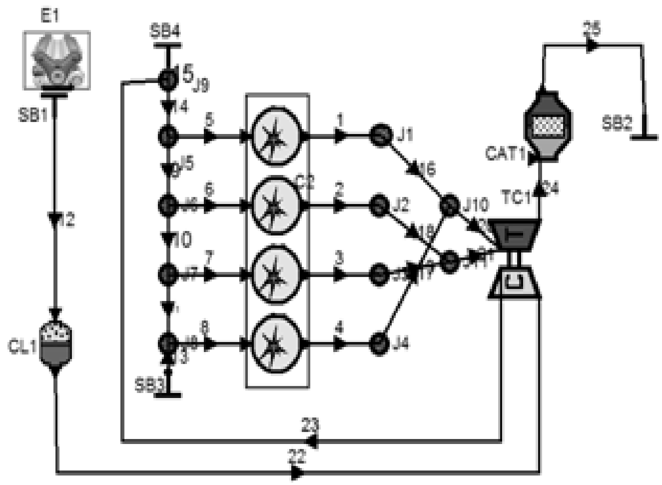

Fig. 4. AVL Boost model of a twin-entry turbocharger configuration for the Renault 1.5 DCi engine.

III. 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 compared to those provided by the manufacturer. Data such as peak engine power of 50kW at 4000 RPM with the BorgWarner KP35 single-entry turbocharger, and other key engine parameters as shown in Table 1, were used.

A. *Maximum Engine Power*

The engine model was run for 100 simulation cycles using the parameters described above and the engine performance results were compared. Figure 6 shows the power and torque output as a function of engine speed in a range of 1000 – 4500 RPM. It is clear that the simulated model engine has produced very accurate predictions of maximum power and torque output at an engine speed of 4000 RPM when compared to the manufacturer's data shown in figure 5. Similar model validations were performed for the 2.0L CI and 1.8L SI engines. A summary of the engine power and torque simulation results are given in table 2 and figure 7.

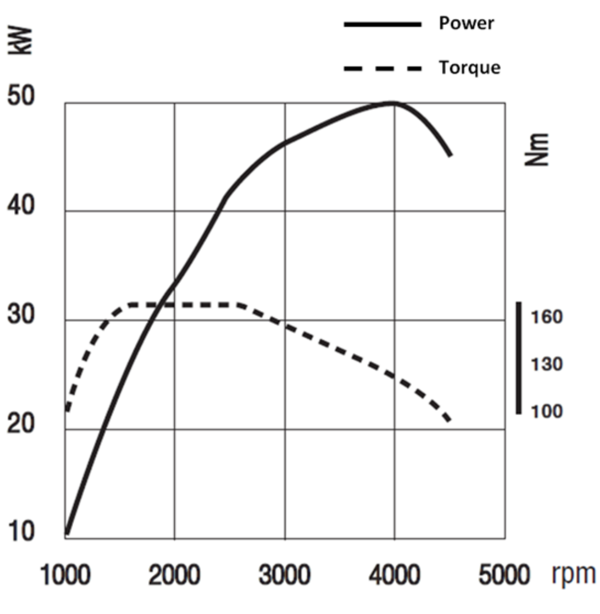

Fig. 5. Performance data of the Renault 1.5 DCi engine.

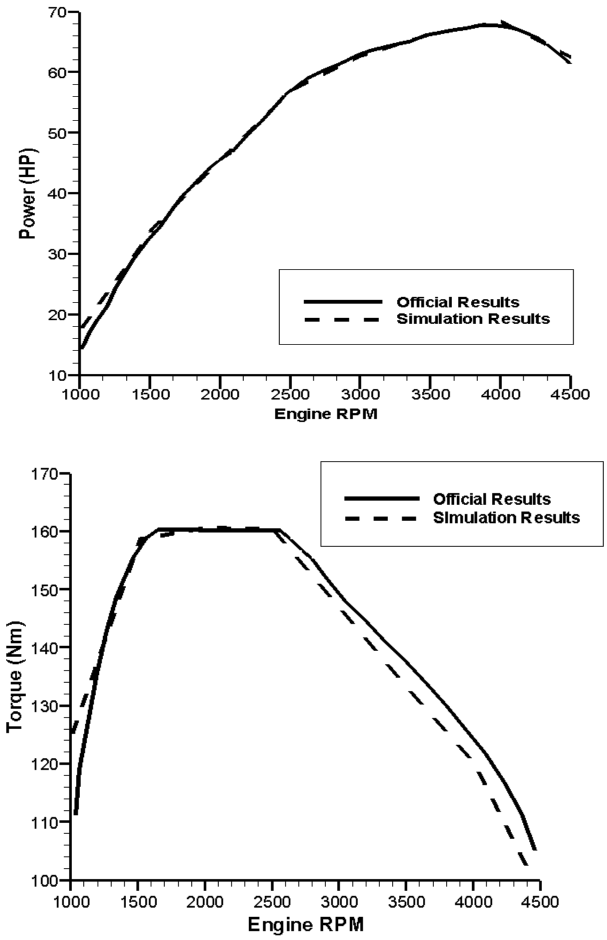

Fig. 6. AVL Boost simulated power and torque results in comparison with manufacturing data of Renault

It is clear that there is a good agreement between the simulated data and the officially published data for the 1.5L DCi engine. For the 1.8 SI engine (figure 7), 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 which means that a fixed number cannot be used to accurately represent a complete combustion definition.

Table 2. Manufacturer and simulated data acquired by the Boost code for the 2.0L compression ignition engine.

	Simulation Results	Official Data	% Difference
Max. Power	68 kW @ 4000 RPM	67 kW @ 4000 RPM	1kW [1.4% Error]
Max. Torque	220 Nm @ 2000 RPM	215 Nm @ 2000 RPM	5 Nm [2.3% Error]

Table 2 show 1.4% and 2.3% differences in power and torque results respectively indicating that the AVL Boost code has accurately re-produced the operational condition of the 2.0L compression ignition engine. The validation results acquired from the three engines clearly indicate that the AVL 1-D simulations have achieved reliable results considering that combustion, thermodynamic and heat transfer codes are used to simulate viable engine operation.

ISBN: 978-988-19252-2-0
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

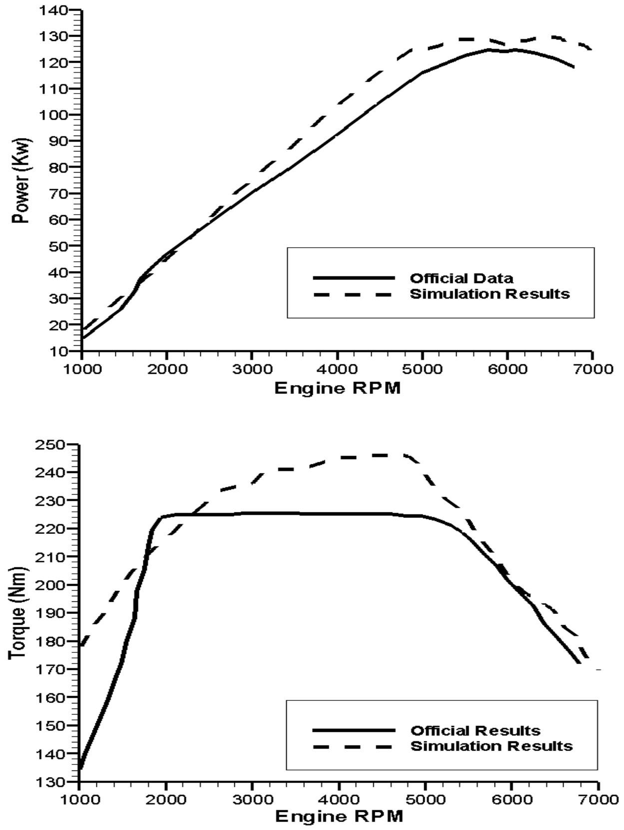

Fig. 7. Comparison between simulated engine and torque results for the 1.8L SI engine to the data specified by the engine manufacturer.

It was concluded based on the above validations that the simulated model engines equipped with the standard single-entry turbocharger are working correctly and therefore could be subsequently adapted to a twin entry turbine housing for a direct comparison analysis between the single- and the twin-entry turbocharger configurations as described in the next section.

IV. Simulation Results

Comprehensive studies will be made by comparing simulation results to reveal the potential changes in engine performance due to the adoption of a twin-entry turbocharger geometry, for results including engine torque, power and brake mean effective pressure (BMEP) etc. 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 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.

A. *Power and Torque Outputs*

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

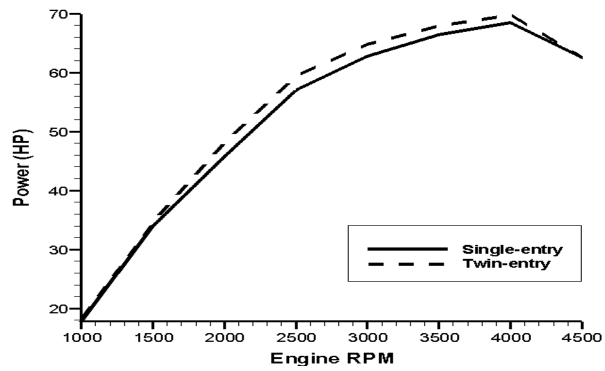

Fig. 8. Increased power output of the Renault 1.5L DCi engine in 1000-3500 RPM engine speed range using a twin-entry turbocharger.

Figure 8 shows the comparison of power output from both single- and twin-entry turbochargers. Unsurprisingly, the greatest gain in power output for the twin-entry configuration occurs at 2500 RPM, providing approximately 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. This power gain is approximately 4% compared to the benchmark data. Similar trends in an increase in engine output performance have been revealed by the AVL Boost test code where a single- and twin-entry turbocharger comparative analysis was performed on a 1.8L spark ignition engine as shown in figure 9.

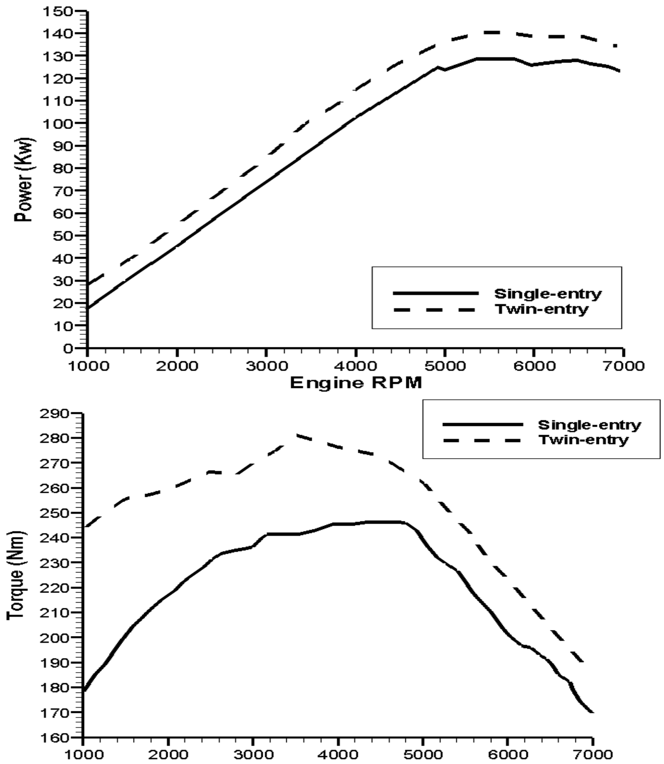

Fig. 9. AVL Boost simulation of increased power and torque a 1.8L SI engine using a twin-entry turbocharger

ISBN: 978-988-19252-2-0
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

Figure 10 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 Nm to 170 Nm. This considerable gain of 5.55% @ 2000 RPM is highly favorable as the engine response performance will have considerably improved during the low engine speed range of 1000-3500 RPM.

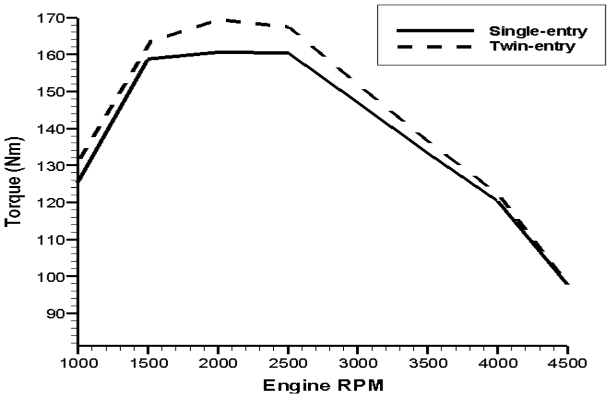

Fig. 10. Increased engine torque output due to the adoption of a twin-entry turbocharger on the Renault 1.5L DCi engine

The third simulation using a 2.0L Peugeot compression ignition engine was performed using the AVL Boost code to further illustrate the effect of a twin-entry turbine housing on the output performance characteristics of the engine. An average increase in power (18%) and torque (9%) from 1000 -3000 RPM are shown in figures 11 and 12.

B. *BMEP Improvement*

The additional air flow rate due to the twin-entry turbine configurations also causes an increase in the inlet manifold pressure, 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 equation 1.

$$Torque = \frac{BMEPx\ Swept\ Engine\ Volume}{2\pi} \qquad (1)$$

It is clear from the equation that increasing the BMEP of an engine results in increased torque characteristics, as shown in figures 10 and 12.

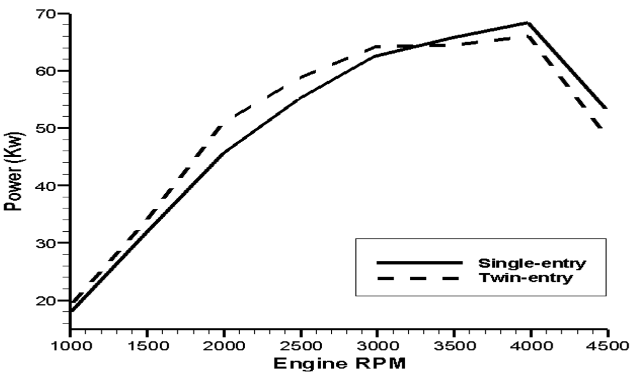

Fig. 11. Increased engine power output of a 2.0L CI engine using a twin-entry turbocharger

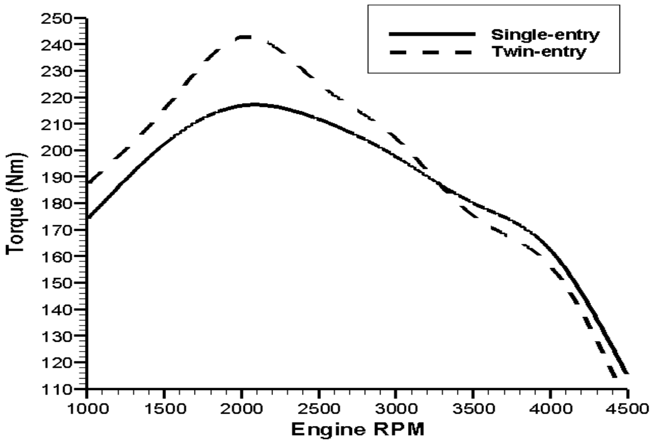

Fig. 12. Increased engine torque output of a 2.0L CI engine using a twin-entry turbocharger

Figure 13 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 @ 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.

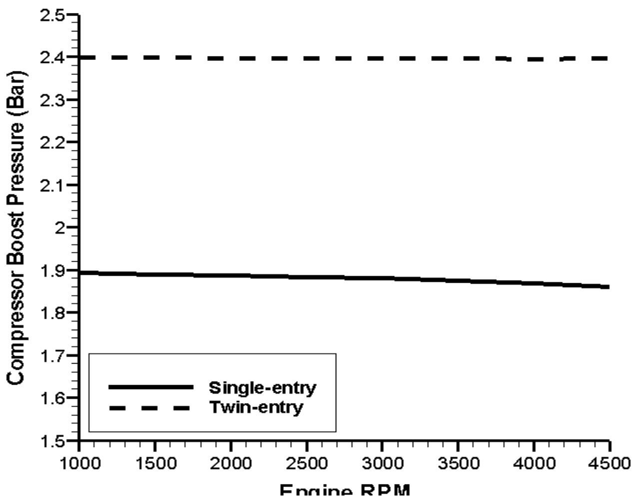

Fig. 13. Compressor "boost" performance gain increasing the manifold pressure and therefore BMEP.

It is apparent from the discussed results that small gains in the performance of the compressor will result in a greater engine performance output, e.g. the increase in compressor air flow resulting in a consequently larger VE. A similar observation can be made for the BMEP results as shown in figure 14, where a maximum BMEP increase of approx. 1 Bar is observed at 2000 RPM. Overall, there is considerable increase in BMEP over an 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 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.

ISBN: 978-988-19252-2-0
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

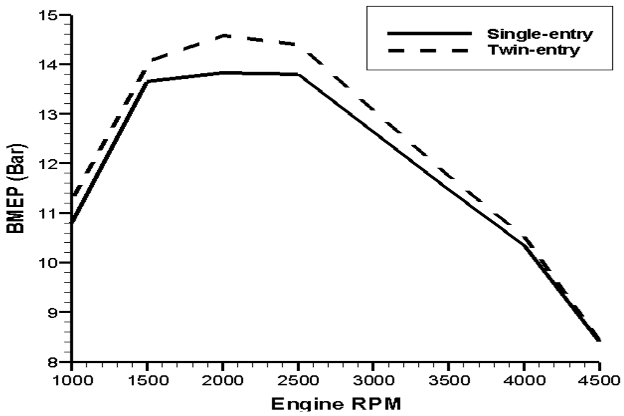

Fig. 14. Comparison between single and twin-entry engine BMEP results of Renault 1.5L DCi engine

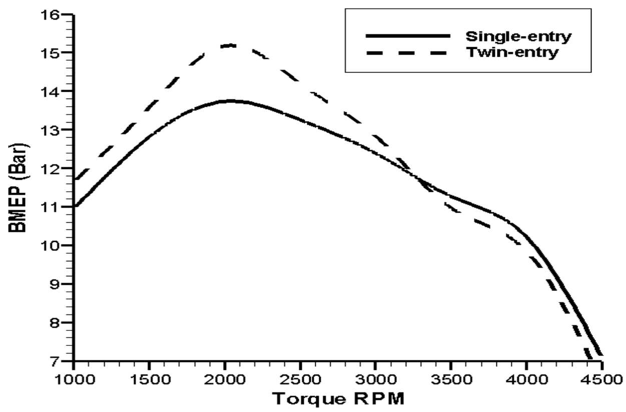

Fig. 15. A 10% improvement in BMEP exhibited in the 2.0L CI engine using a twin-entry turbocharger.

An improvement in BMEP is also noted from the 2.0L CI engine as shown in figure 15. There is an increase in BMEP from 13.75 to 15.25 Bar which equates to a 10% improvement at 2000 RPM due to the addition of a twin-entry turbine housing. The simulation results acquired from the 1.8L spark ignition engine (figure 16) also showed that the BMEP increased by 15.9% 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 are shown in table 3.

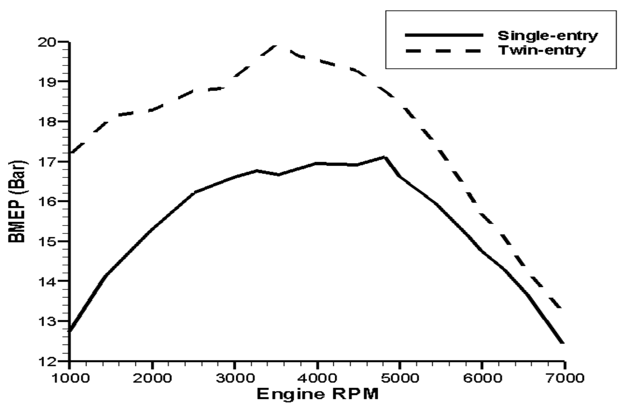

Fig. 16. A 15.9% BMEP improvement exhibited by the 1.8L SI engine

Table 3. Summary of the engine performance increase due to the application of a twin-entry turbocharger.

	1.5L DCi Engine	2.0L CI Engine	1.8L SI Engine
Power	5.55 %	18 %	14.8 %
Torque	5.55 %	9.03 %	14.0%
BMEP	4.20 %	8.2 %	15.9%

V. 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 1.5L DCi Renault engine show that the application of a twin-entry volute design enhances the performance of the engine when operating during low RPM conditions, the most effectiveness being observed from 1500-3000 RPM showing a maximum 27.65% increase in turbine shaft speed and a maximum 4.2% increase in BMEP. Both engine torque and power performance also increased by 5.55% at 2000 RPM resulting in an average performance increase of 4% during 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.

VI. Future Work

Further refinements to the Renault model are also currently being undertaken including the application of exhaust gas recirculation (EGR) and the use of experimentally acquired data to refine the operating parameters of the AVL Boost 1-D model.

References

[1] Arnold D. (2004). Turbocharging Technologies to Meet Critical Performance Demands of Ultra-Low Emissions Diesel Engines. SAE International Technical Paper Series, 2004-01-1359.
[2] Hawley J., Wallace F., Cox A., Horrocks R. and Bird G. (1999). Variable geometry turbocharging for lower emissions and improved torque characteristics. Proceedings of the Institution of Mechanical Engineers, Part D: J. of Automobile Engineering; 213(2): 145-159.
[3] Chadwell C.J. and Walls M. (2010). Analysis of a SuperTurbocharged Downsized Engine Using 1-D CFD Simulation. SAE International Technical Paper Series, 2010-01-1231.
[4] Ishii M. (2009). System Optimization of a Turbo-Compound Engine. SAE International Technical Paper Series, 2009-01-1940.
[5] Petitjean D., Bernardini L., Middlemass C. and Shahed S.M. (2004). Advanced gasoline engine turbocharging technology for fuel economy improvements. SAE International Technical Paper Series, 2004-01-0988.
[6] Watel E., Pagot A., Pacaud P, and Schmitt J. (2010). Matching and Evaluating Methods for Euro 6 and Efficient Two-stage Turbocharging Diesel Engine. SAE International Technical Paper Series, 2010-01-1229.
[7] Aghaali H. and Hajilouy-Benisi A. (2008). Experimental modelling of twin-entry radial turbine. Iranian Journal of Science & Technology, Transaction B, Engineering, Vol.32, No. B6, pp. 571-584.
[8] AVL Boost Theory Manual, version, year, etc.
[9] Weissbaeck M. (2011). "Diesel Downsizing", Engine Technology International, January, pp. 26-28
[10] AVL Boost User Manual, version, year, etc
[11] Hillier, V. and Coombes, P. (2004). Fundamentals of motor vehicle technology. 5th ed. Cheltenham, Nelson Thornes Ltd.