COMPARATIVE STUDY OF TURBINE SHAFT SPEED FOR TWO ALTERNATIVE TURBOCHARGER TYPES FITTED TO A LIGHT-DUTY CI ENGINE

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Abstract: *A common drawback associated with 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 turbocharger becomes effective. This study will therefore investigate changes in turbine speed during an engine speed range of 1000 - 4500 RPM. It is anticipated that the adoption of a twin-entry turbocharger may reduce the spooling time. The AVL Boost simulation code is used to construct a light duty compression ignition (CI) simulation model incorporating model geometry and refined engine parameters obtained experimentally from an identical test engine. The model is subsequently modified to include a twin-entry turbine housing and split-pulse exhaust manifold in conjunction with the original turbine impeller, compressor housing and utilising the same compressor flow map. Model validation is achieved by comparing power and torque values to those published by the engine manufacturer, and results published by Kusztelan et al [1]. This work will introduce data examining turbocharger shaft speed and engine performance to include brake mean effective pressure (BMEP) and volumetric efficiency (VE) obtained from the standard and twin-entry turbocharger simulation models measured across a specific engine operating range of 1000 – 4500 RPM.*

Keywords: ENGINE DOWNSIZING, COMPRESSION IGNITION ENGINES, PRESSURE CHARGING, TURBOCHARGER SPEED, TWIN-ENTRY TURBINE HOUSING

1. Introduction

Turbochargers are increasingly being used in conjunction with engine downsizing practices [2] as they contribute to the enhancement of power density for a given cylinder capacity, improve fuel consumption and contribute towards the reduction of gaseous exhaust emissions. However, the slow spooling response of the turbine (i.e. slow rotational speed) at low engine crankshaft speeds is a common problem which can result in turbo lag. This can lead to reduced engine response and performance. Adopting different types of pressure boosting devices such as variable geometry, two-stage and twin-entry turbochargers could increase and maintain a suitable turbine rotational speed at lower engine speeds therefore reducing the effect of this typical characteristic.

Twin-entry turbochargers (Figure 1) are commonly associated with large capacity spark ignition engines but the application of these turbochargers to smaller capacity, high speed diesel engines is now becoming more popular.

Twin-entry turbochargers are attributed to utilising the pulsating energy of the exhaust gas more efficiently. This occurs primarily due to the separation of the differently pulsed exhaust outlets, thus preventing the interactions between the individual gas flows [3]. An optimised energy transfer from exhaust gas to the turbine impeller may therefore be achieved during lower engine operating speeds of 1500 - 2500 RPM in contrast to the transfer of energy in single-entry turbocharger types. The use of this technology could consequently increase the rotational speed of the turbine at lower engine RPM's (revolutions per minute) resulting in more efficient compressor and turbine performance.

A one-dimensional (1-D) engine simulation software package is used to investigate the effect of turbocharger spooling speed on engine response and performance. The software accurately replicates an engine model using different configuration and operating conditions. This allows the user to quickly and precisely change variables to identify any initial problems early in the design and testing stages of engine development.

Fig. 1 *Twin-entry turbocharger with corresponding split-pulse manifold [4].*

2. Engine Simulation Model

For the purpose of this 1-D analysis the AVL Boost code is used to simulate a four-cylinder 1.5L DCi engine (see table 1) fitted with a factory supplied single-entry turbocharger as shown in figure 2. The simulation software is based on a series of models used to accurately replicate operating conditions, such as combustion and mechanical friction of the 1-D engine model.

The engine model is subsequently modified to include a twin-entry turbine housing and split-pulse exhaust manifold (figure 3) however still utilising the same turbine impeller, compressor and housing. As a result identical compressor flow maps are used on both simulations. A comparative analysis between the two simulated models is undertaken to investigate any differences in turbine speed across the measured test range.

Parameters	1.5L DCi K9K Engine
Bore	76mm
Stroke	80.5mm
Exhaust Valve Lift	8.6mm
Inlet Valve Lift	8.0mm
Compression Ratio	17.9:1
No. of Cylinders	4
Valves per Cylinder	2

Table.1 *Key engine parameters of the 1-D engine model*

Refined engine parameters and geometry are obtained experimentally from an identical test engine (figure 4) and incorporated into the 1-D engine model.

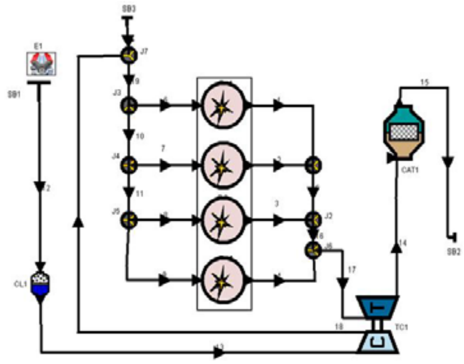

Fig. 2 AVL Boost model of the single-entry turbocharger configuration for the Renault 1.5L DCi engine.

In order to implement a twin-entry turbine housing as shown in figure 3, a modified manifold configuration with a split-pulse design is introduced. 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 [1] as highlighted in figure 3. The AVL Boost software code will therefore automatically recognise the increase in turbine housing inputs being changed to a corresponding twin-entry configuration.

A simplified twin-entry model is also selected within the software to reconfigure the operating conditions of the turbocharger. Selecting the simplified model implies that the software applies the same turbine map i.e. identical pressure ratios to both flow regions of the turbine housing. The flow interaction between both pipe regions can then be subsequently modelled by defining an inlet interference coefficient which allows the software to identify the cross section of both turbine volutes [5].

3. 1-D Model Validation

The boundary conditions of the model engine are refined utilizing data obtained from experimental tests using an in-house engine test cell dynamometer as shown in figure 4. Data recorded during a complete testing cycle is used to obtain a detailed overview of engine and turbocharger operating conditions and then subsequently defined within the 1-D AVL Boost engine model.

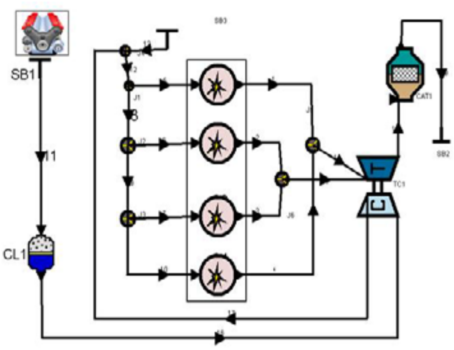

Fig. 3 AVL Boost model of the twin-entry turbocharger configuration for the 1.5L DCi engine.

The operational conditions of the test engine are analysed using an "in-house" designed fast sampling rate (30-40 kHz source frequency) data acquisition system capable of logging key parameters including turbocharger inlet/outlet temperature and pressure values for every 1° engine crank angle.

Fig. 4 Experimental test engine

Power and torque results recorded from the single-entry simulated 1-D engine model (figure 5) are compared to those provided by the manufacturer of the 1.5L DCi engine. It is clear from the graphs that the simulated engine model is producing similar performance output values as to those intended by the manufacturer and that the engine is operating correctly. There are however slight discrepancies between the simulated and published power and torque values which the authors attribute to the Vibe combustion model, Woschni heat transfer model and the Patton et al. friction model [6] used within the AVL Boost code. Thus explaining why both power and the torque curves recorded from the 1-D model engine show slightly lower performance characteristics but similar trends to those measured by the manufacturer during an operating range of 1000 – 4000 RPM. This indicates that the 1-D simulated engine model is working correctly and can therefore be modified to utilise a twin-entry turbocharger as explained in figure 6. A comparative study between both turbocharger configurations is used to analyse any changes in engine performance due to the adoption of the twin-entry turbocharger and corresponding split-pulse manifold. Specific engine performance parameters such as turbine shaft speed, compressor boost, BMEP and VE are used to determine changes in engine performance.

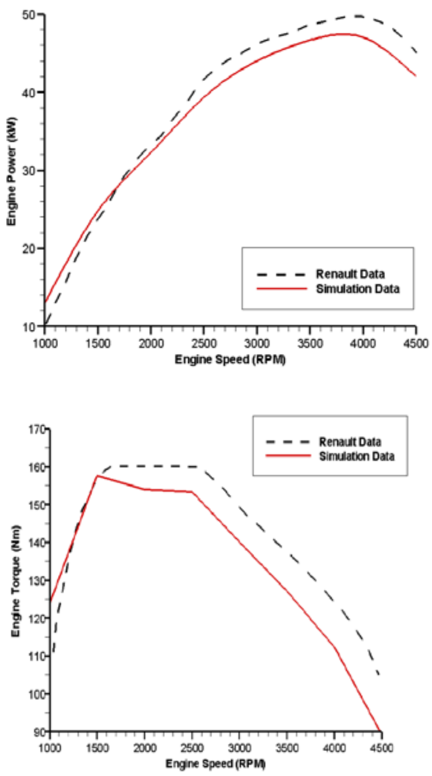

Fig. 5 AVL Boost simulated power and torque results compared to official manufacturer data [7].

4. Simulation Results

To demonstrate the potential benefits of the twin-entry turbine volute and its effects on turbine spooling time, a steady state (wide open throttle) test is run for 100 cycles at 500 RPM increments between an engine operating range of 1000 - 4500 RPM. At each RPM increment, engine parameters including air/fuel ratio and in-cylinder heat transfer conditions are defined in conjunction with key parameters that accurately represent the engine inlet/exhaust manifold geometry, valve lift profiles and compressor/turbine flow maps.

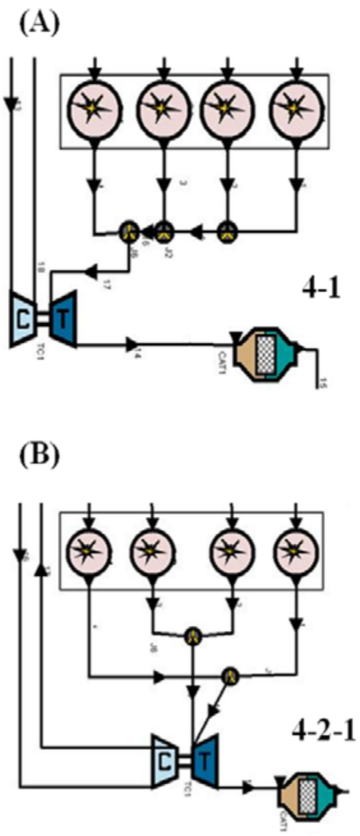

Fig. 6 *Image (A) shows a 4-1 exhaust manifold configuration used for standard turbine housings. Image (B) highlights the change in exhaust manifold configuration to allow for the dual inlet turbine housing geometry.*

4.1 Turbocharger Shaft Speed

Adoption of a twin-entry turbocharger has the potential to improve the energy transfer from the exhaust gases to the turbine impeller, leading to an increase in shaft speed for a given engine RPM. This will allow the turbocharger to achieve a reduced 'spooling time' resulting in improved compressor efficiency at lower engine speeds. Achieving greater turbine shaft speeds at reduced engine speeds will allow the compressor to operate in greater efficiency zone therefore increasing the volumetric flow rate.

Figure 7 shows the comparison of turbocharger shaft speeds between the single- and twin-entry configurations plotted against an engine speed range of 1000 - 4500 RPM. It is clear from the graph that at engine speeds in the range of 1000 – 4500 RPM the turbine shaft for the single-entry turbocharger reaches speeds in the range of 155 - 165K RPM. These high rotational speeds at low engine revolutions are not typical operating conditions for an actual turbocharged engine. This being due to the variable loading conditions experienced under normal engine operating conditions. As previously mentioned the AVL Boost software uses models to predict and simulate engine and turbocharger operating conditions.

A Vibe function model is used to model the heat release characteristics of the engine. The start and duration of combustion is therefore defined as two constant shape parameters which are not dependant on engine speed and loading conditions [5].

It can be seen from the graph in figure 7 that the shaft speed of the twin-entry turbocharger increases across an engine speed range of 1000 - 4500 RPM, and most noticeably within the speed range of 2000 - 2500 RPM. This specific range is regarded as a common region of performance inefficiency for small capacity, high speed diesel engines which are typically used in “urban” driving environment. The simulation results show that there is an approximate 11% increase in shaft speed at 2500 RPM and an average increase throughout the complete RPM range of approximately 9.5%. This suggests that an employable amount of energy has been transferred from the exhaust gas to turbine impeller at lower engine crankshaft speeds.

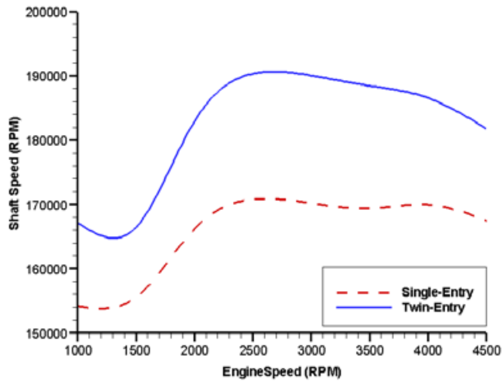

Fig. 7 *Comparison of shaft speed for both turbocharger configurations.*

4.2 Compressor Discharge Pressure: Boost

Figure 8 shows a comparison of compressor discharge pressure for both single- and twin-entry turbine housings utilizing an identical compressor flow map as shown in Figure 9. Achieving higher impeller shaft speeds indicates that the compressor can achieve optimum boost levels corresponding to an increased volumetric flow rate at lower crankshaft speeds.

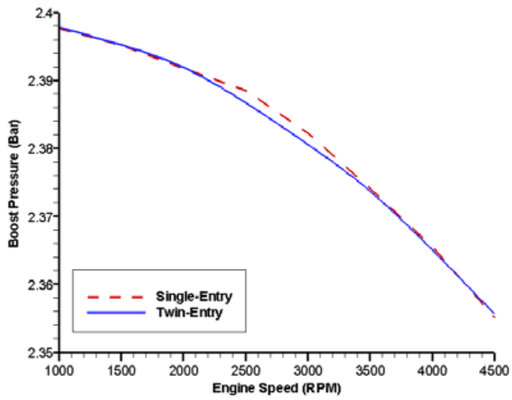

Fig. 8 *Comparison of Boost pressure for both turbocharger configurations.*

This could lead to a decrease in engine pumping losses and therefore increased VE allowing the engine to achieve greater power and torque outputs.

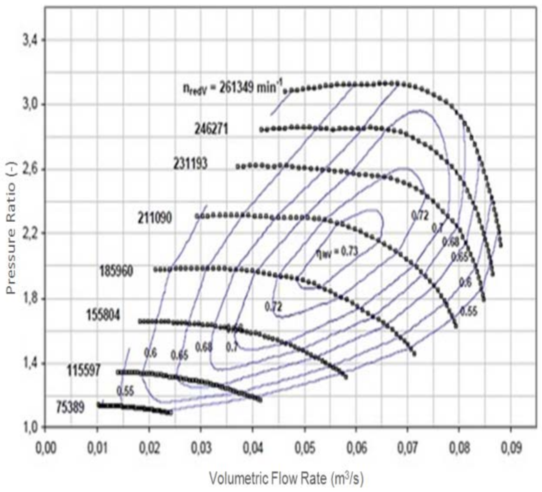

Fig. 9 BorgWarner KP35 compressor flow map

4.3 Volumetric Efficiency

VE is typically used to quantify the rate of change between the air flow into and out of the engines combustion chamber. A percentage is therefore adopted to quantify this rate of change which ideally should lie above 100%. Engines exhibiting VE values above the 100% benchmark indicate a greater rate of change hence, providing increased performance. Naturally aspirated engines are normally unable to significantly exceed this 100% benchmark. Forced induction methods are therefore frequently employed to achieve a higher volumetric efficiency. A typical turbocharged engine will achieve a VE greater than 100% [8].

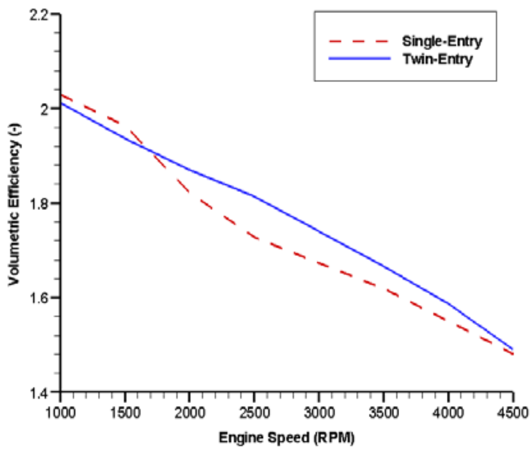

Fig. 10 *An approximate 23% increase in VE at 2500 RPM shows a clear improvement in engine output and response during low engine speed conditions.*

This increase in VE results in a greater pressure acting on the piston crown enhancing the torque characteristics of the engine. This is possible because the energy loss usually associated with the rate of airflow change within the combustion chamber decreases, thus reducing the overall pumping losses of the engine. Figure 10 indicates that the twin-entry turbocharger configuration increases the VE in comparison to the single-entry data. The improved compressor performance has resulted in an approximate 5% increase of VE at 2500 RPM with an overall improvement of 3.6% within the 2000 - 3500 RPM crankshaft speed range. These results demonstrate why a greater torque is achieved at lower engine speeds by an engine equipped with a twin-entry turbine housing.

4.5 BMEP Improvement

It is apparent that the adoption of a twin-entry turbine housing may result in increased impeller speeds at a given engine speed range (2000 - 2500 RPM), consequently allowing the compressor to reach its optimum regulated boost pressure in a reduced amount of time. This can lead to an improvement in volumetric efficiency and BMEP at similar engine speeds. BMEP is a parameter used to determine the torque characteristics of an internal combustion engine as shown in equation 1.

$$Torque = \frac{BMEP \times Swept\ \ Engine\ \ Volume}{2\pi} \quad (1)$$

As with the increase of VE noted at 2500 RPM a maximum increase in BMEP of approximately 1.5 Bar is observed at the same engine speed as shown in figure 11. The twin-entry configured engine model achieves an approximate 10% increase in BMEP calculated during the 2000 – 3500 RPM range resulting in an improvement in engine torque.

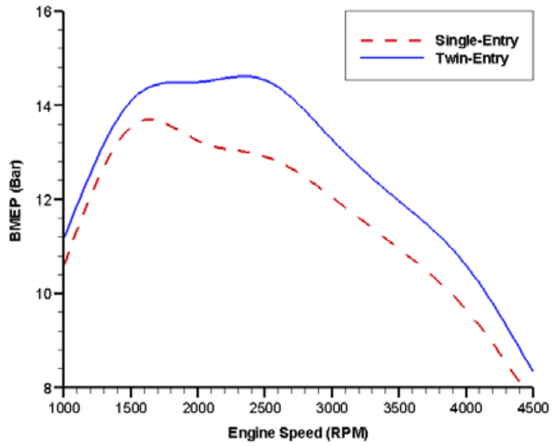

Fig. 11 *Comparison between single and twin-entry engine BMEP results of 1.5L DCi engine.*

4.3 1.5L DCi Performance: Power and Torque

The benefits of increasing turbine shaft speed at low engine crankshaft RPM is the decrease in "spooling time", increase in BMEP and VE, resulting in a corresponding increase in engine power and torque.

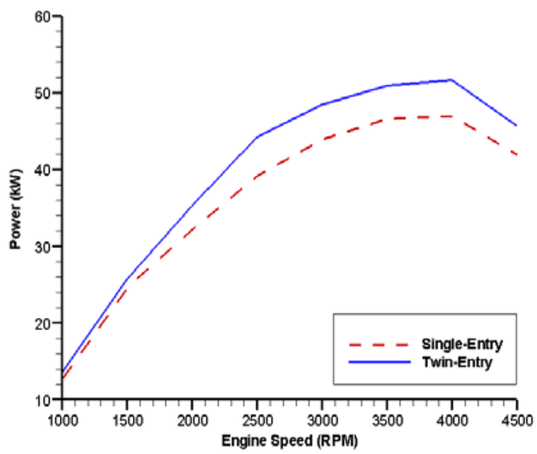

Fig. 12 *Increased power output of the Renault 1.5L DCi engine in 1000 - 3500 RPM engine speed range using a twin-entry turbocharger.*

Figure 12 shows a comparison of power output from both single- and twin-entry turbocharger models. The largest increase of power output for the twin-entry configuration is observed at 2500 RPM, indicating an approximate 12.5% improvement in the power produced which is most likely the result of an increase in compressor efficiency due to improved energy transfer from the exhaust gases to the turbine impeller. When comparing the difference in power over the complete RPM test range the twin-entry turbocharger model achieves an approximate 8.5% increase over the manufacturer's published data. Figure 13 shows a comparison of engine torque demonstrating a clear improvement over the complete simulation range.

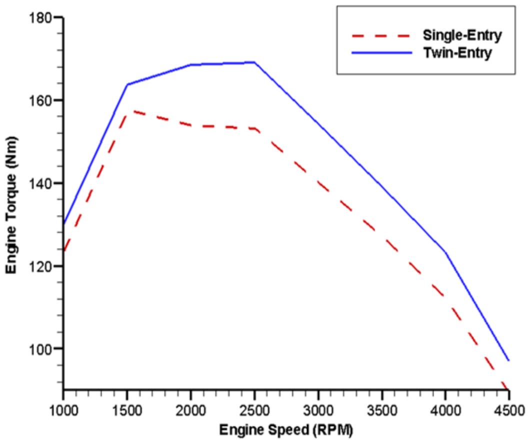

Fig. 13 *Increased engine torque output due to the adoption of a twin-entry turbocharger on the 1.5L DCi engine.*

A maximum increase of power (5kW) and torque (15.7Nm) is observed at 2500 RPM. Torque output is improved by approximately 8.5% throughout the complete RPM range and is particularly apparent during the low engine speed range of 1000 - 2500 RPM which indicates an average increase of 7.5 %.

Conclusions

The AVL Boost 1-D simulation has demonstrated potential spooling speed improvements due to the adoption of a twin-entry turbocharger with its corresponding split-pulse exhaust manifold.

The results showed that the application of a twin-entry design utilising identical volutes in a symmetrical arrangement enhances the turbine shaft speed during low engine speed conditions. The most significant improvement being observed at 2500 RPM showing an overall 11% increase in turbine shaft speed, resulting in an overall improvement of 3.6% and 10% respectfully in volumetric efficiency and BMEP within the 2000 - 3500 RPM crankshaft speed range. An overall average of approximately 8.5% in power and torque within the complete engine simulation RPM range (1000 - 4500 RPM) is also noted due to the increase in turbine speed.

The study clearly indicates that increases in turbine shaft speed, VE, BMEP, power and torque are achieved through the adoption of a twin-entry turbocharger.

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