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A Review of Novel Turbocharger Concepts for Enhancements in Energy Efficiency

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Abstract

Turbochargers are extensively used throughout the automotive industry as they can enhance the output of an internal combustion (IC) engine without the need to increase its cylinder capacity. The application of such a mechanical device enables automotive manufacturers to adopt smaller displacement engines, commonly known as "engine downsizing". Historically, turbochargers were often used to increase the potential of an already powerful IC engine, e.g. those used in Motorsport. The emphasis today is to provide a feasible engineering solution to manufacturing economics and "greener" road vehicles. It is because of these reasons that turbochargers are now becoming much more popular in industry applications. The aim of this paper is to provide a review on the current testing techniques and validation methods used to analyse different turbocharger types and designs, with discussions on future trends.

Keywords: *Automotive turbocharger, engine downsizing, twin-entry and VGT turbochargers*

1. Introduction

The application of turbochargers enables manufacturers to use smaller displacement engines. This is possible because the engine performance is related to the force acting upon the piston which produces work and therefore torque. A turbocharger unit is comprised of two main components: a turbine and a compressor, and its purpose is to increase the volumetric efficiency of the combustion chamber. The compressor of the turbocharger uses air from the ambient atmosphere and increases its density through the rotating impeller blade passages. The resultant high density airflow then enters the engine combustion chamber to mix with the fuel. Due to the increased air density (hence higher mass flow rate), the brake mean effective pressure acting upon the piston crown is enhanced. This will increase the force acting upon the piston which means the engine can produce more torque and therefore power. The enhanced combustion process will generate a more energetic pulsating exhaust gas which flows via the exhaust manifold into the turbine which drives the compressor. The turbine of the turbocharger produces a high back pressure on the exhaust manifold which results in the exhaust gas pressure being higher than the atmospheric pressure. The energy generated due to the expansion of the exhaust gas is then used to rotate the turbine impeller which directly drives the compressor, completing the cycle.
For example a four cylinder engine with a 1-2-4-3 firing order and a single-entry turbocharger will produce the following conditions. At the end of the exhaust stroke in cylinder 1 (i.e. when the piston is approaching top dead center (TDC)) the momentum of the exhaust gas flowing into the manifold will scavenge the burnt gas out of the cylinder. However in cylinder 2 the exhaust valve is already open allowing for exhaust gas to enter the manifold as well. This means the exhaust pulse from cylinder 2 will influence the flow of exhaust gas from cylinder 1, thus affecting the energy transfer to the turbine *(www.garrett.com)*. In contrast to a single-entry turbocharger, a twin-entry turbine housing (as shown in figure 1) will better utilize the pulsating energy of the exhaust gas, thus boosting the performance of the turbine which directly increases the impeller rotating speed of the compressor. Not only is this more practical and economical but also provides a potential for improvement in the reduction of gaseous emissions.

2. Current Status of Auto-Turbocharger Researches

2.1. Experimental Tests

A topic of significance is that different test methods have been used for analysing the performance of a turbocharger. Typically turbochargers are normally tested as an on-engine application using engine test benches coupled with a dynamometer. Another method that can be applied to analyse the turbocharger unit is by using test benches along with other experimental techniques. In the latter case, either 'hot' or 'cold' exhaust gas can be used to drive the turbine of a turbocharger.

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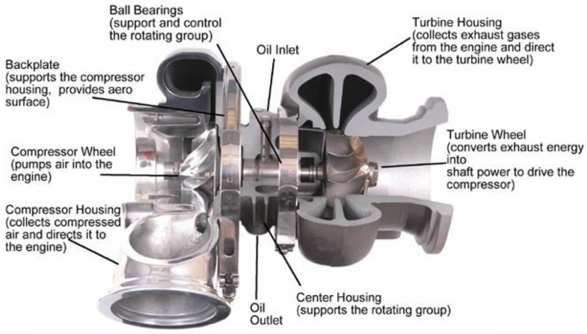


Fig. 1. Typical automotive turbocharger adopting a twin-entry turbine housing (www.garrett.com)

2.1.1. *"Cold" Exhaust Gas Test Benches*

Capobianco and Marelli [1] used a cold gas test apparatus to analyse the performance of a turbocharger. They found that certain areas of interest must be looked upon when undertaking such an experimental procedure. These include analysing the compressor and the turbine characteristics over a typical operating range, similar to those used on an internal combustion engine. They also stated that steady and unsteady feature of exhaust gas flows and their effect on engine performance are also important. Flow unsteadiness will occur due to the pulsating nature of the exhaust gases, which originates from the pumping action of the engines combustion cycle. Therefore it is necessary to replicate this effect in the test. Capobianco and Marelli have adopted two methods (see figure 2), the first being a series of rotating valves and the second an application of a cylinder head. In the former, the opening and closing of the valves create a similar pulsating nature of air supply to the turbine. This type of setup in conjunction with an electrical motor can produce a pulsating frequency similar to that of a working engine (i.e. 10 - 200 Hz). However the application of this method would be harder to calibrate and therefore maintain these values correctly. To solve this problem, the second method uses an engine cylinder head that is placed onto a plenum. The substitution of the rotating valves with the cylinder head produces a much more realistic and appropriate pulsating air flow. An additional benefit of adopting this method is that a manifold can be used to provide a more accurate testing environment. As the air reaches the internal inlet and outlet valve configurations of the cylinder head, the opening and closing conditions for the exhaust gas to enter the manifold can be easily replicated.

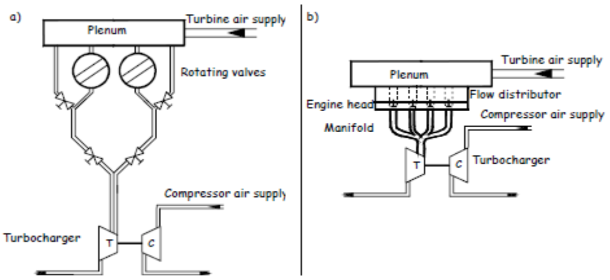


Fig. 2. Images showing two different test rigs used to perform the experimental analysis of a turbocharger unit independently from an engine [1]

Furthermore, the second experimental configuration was used to study the effect of pulsating exhaust gas flow through pressure measurements made at different locations within the turbo system. During the test, an electric fan heater was used to prevent the condensation of the inlet air. As this is a cold temperature test apparatus, the operating conditions usually found in normal driving conditions cannot be directly applied. In order to measure the pressure accurately, a series of transducers were used and coupled with high-speed sampling devices which allow for simultaneous pressure signals under transient operating conditions to be recorded. The results obtained are shown in figure 3. They indicate that the geometric shape of the exhaust gas manifold has some effects on the recorded pressure ranges. When comparing the results from the mixing volume with that of the turbine entry, one can see that the pulse period for pressure decrease and/or increase is quite similar, in terms of peak pressure pulses. It was thus deduced that the measured pressure variations in the exhaust gas manifold produce frequencies and amplitudes similar to those of the supplied air flow. Therefore a conclusion was made that the branching geometry of the manifold results in lower pressure frequencies and amplitudes at the turbine entry due to a substantial increase in flow unsteadiness. It was therefore concluded that it is important to analyse the effect of different exhaust gas manifold geometries as well.

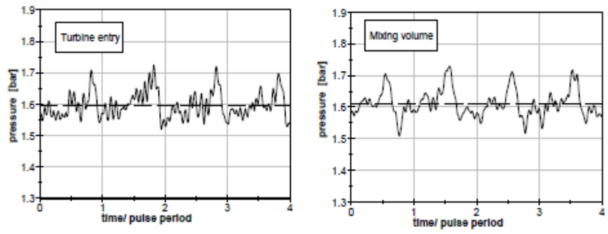


Fig. 3. Variation of pressure at different locations of a turbocharger assembly [1]

During the above experimental results, Capobianco and Marelli observed that maintaining a continuous pulse environment could produce a more uniform inflow which would further increase the performance of the turbocharger. They assumed that this phenomenon may be directly related to the turbine inlet pulse amplitude. A similar but independent experiment was carried out by Aghaali and Hajilouy-Benisi [2]. They used a testing apparatus in which three screw compressors produced the necessary mass flow rate to drive the turbine of the turbocharger. An electric heater was used to prevent the formation of condensation. However, this does not produce the same temperature levels as those found during normal engine operation. Different from the work of Capobianco and Marelli, neither a cylinder head nor a valve system were used to produce the pulsating exhaust gas. Instead, Aghaali and Hajilouy-Benisi used a twin-entry turbocharger unit that splits the air supply for each volute entry as seen in Figure 1. In order to control the amount of air supplied to each entry, a series of mass flow measuring and individually controlled devices were used. This was done to supply each separate entry of the turbine with varying mass flow rates. Even though the air supply is varying, it still does not replicate the pumping action of the exhaust gas. That is why this type of unit was used to analyse the effect of a twin-entry volute on turbocharger performance. The parameters that were measured throughout the testing include total to static isentropic efficiency, pressure ratio and the ratio of mass flow rate between the two entry ports.

Measurements were taken at different rotational speeds of the turbine, of which the results are shown in figure 4.

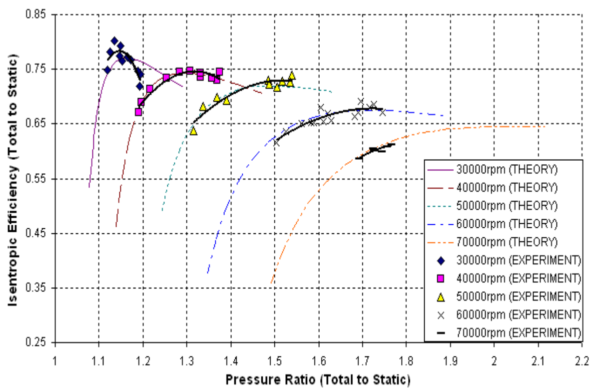
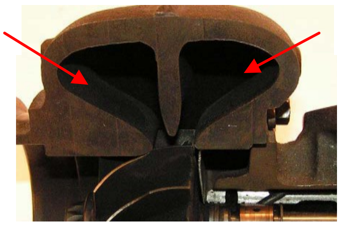


Fig. 4. The relationship between the pressure ratio and the isentropic efficiency at varying turbine speeds [2]

Aghaali and Hajilouy-Benisi concluded that the geometry of the turbine entry has an effect on the mass flow rate and this is in agreement with those discussed in other literatures. They also noted that the energy loss near the geometry of the hub entry side of the volute will affect the local air flow direction. It is known that under partial operating conditions, an increased mass flow near the hub side could lead to a decrease in turbine efficiency. Therefore the flow will tend to follow the path of least resistance for which in the case of a twin-entry turbocharger is the shroud side. The locations of the hub and the shroud of a twin-entry turbocharger are shown in figure 5.

Shroud Side



Hub Side

Fig. 5. A cross-section of a twin-entry turbocharger showing different entry locations of the turbine volute [12]

It is important to stress that the results acquired by the above researchers were recorded using a cold gas/air source. This means that the typical engine exhaust gas temperatures are not replicated.

2.1.2. "Hot" Exhaust Gas Generation

As a pressure increase will result in an increase in temperature, the results discussed above will be different to those measured at higher temperature environments. To consider these, Venson *et al.* [3] used a turbocharger test apparatus which replicates the temperatures of exhaust gas produced under normal engine operating conditions. To achieve this, an in-house built combustion chamber was used to analyse the turbocharger independently from an operating engine. One interesting observation was that the experimental apparatus is self-sufficient. This implies that the combustion chamber provides the turbine with hot gas which can drive the compressor without any additional external power source. The compressor then provides the combustion chamber with dense exhaust gas of required mass flow rate which keeps the cycle running smoothly, eradicating the need for an externally provided air supply. Even if it is able to produce high temperatures, this type of test bench will not emulate the pulsating nature of the exhaust gas.

A different approach was used by Luján *et al.* [4] to produce the operating temperatures needed for generating accurate turbocharger flow maps. An existing compression ignition engine was adopted which is run and controlled using a screw compressor. The compressor produces the required mass flow rate of dense air for the combustion process, which is used as hot exhaust gas and an energy source to run the turbocharger. This is similar to running the engine using a dynamometer, but without the extra costs involved while running an engine test cell. An additional benefit of using an existing engine system is that the pulsating nature of the exhaust gas can be accurately and realistically produced. This has been proven by Luján *et al.* who have taken pressure measurements at the turbine inlet of a turbocharger over a varying degree of crank angles (see figure 6).

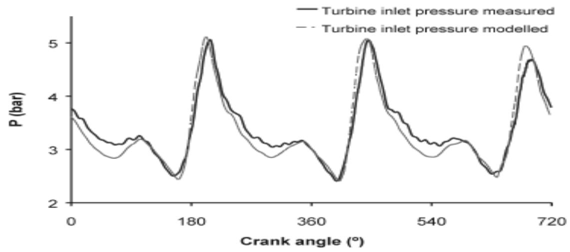


Fig. 6. Measured turbine inlet pressure of the turbocharger showing the pulsating nature of the exhaust gases [4]

These measurements are important as they provide the raw data used to produce the flow maps needed to quantify the performance of the turbocharger. The compressor flow and efficiency maps which display a series of speed lines are therefore produced using measured values of air mass flow rate and pressure ratios. It is worthwhile to mention that using this type of test bench enables the user to take measurements from the turbocharger at low engine loads which is crucial for compression ignition engines. The results can then be used to tailor the turbochargers to low engine speed conditions that turbocharger manufacturers normally do not consider when testing their units.

The turbocharger efficiency graphs shown in figure 7 are plotted using recorded pressure and temperature values. In general, a turbocharger should exhibit its maximum efficiency typically near the middle of the constant speed lines as seen in the graphs.

In contrast to the above described experimental apparatus, Vávra *et al.* [5] adopted a spark ignition high power density engine coupled to a direct current dynamometer to perform the turbocharger analysis. Natural gas was used to provide the fuel source for the combustion. Similar to Luján *et al.* [4], Vávra *et al.* [5] recorded a series of parameters which include exhaust gas temperature, mass flow rate, turbine shaft speed and pressure, thus providing sufficient information for the turbocharger efficiency and engine performance calculations (see figure 8). While comparing these results to those of figure 6, consistencies between the two sets of measured turbocharger intake pressures are apparent. In agreement with those previously described, the pulsating nature of the exhaust gases are clearly visible, and this further confirms that using on-engine turbocharger testing methods would provide more accurate and reliable results. Figure 8 also shows that the cylinder pressure (purple line) during the exhaust stroke falls below the turbine inlet pressure (red line). It is known from literature reviews that this observation is typical when using in-cylinder un-cooled pressure sensors. Even though the problem was identified, the results were still deemed viable for validation application, due to the fact that the pressure variation pattern would be more reliable at high cylinder pressures.

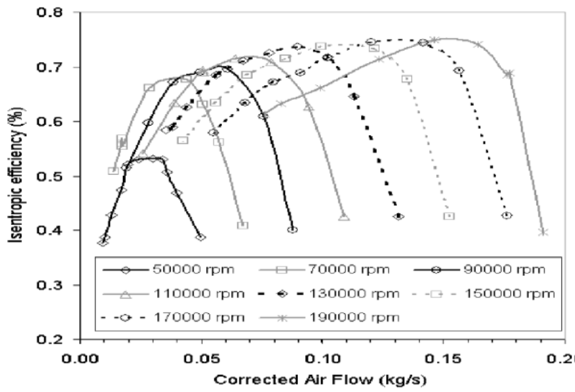


Fig. 7. The calculated isentropic efficiency of the turbocharger from measured pressure and temperature values over a range of rotating speeds [4]

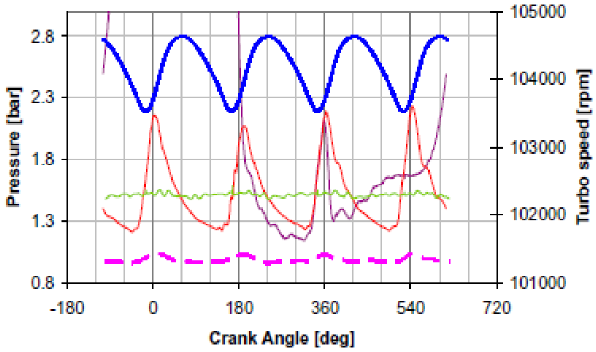


Fig. 8. Variation of pressure and turbo speed against crank angle [5]

Based on the literature review and discussions, a growing trend of cross-validation between experimental measurements and numerical analysis is apparent. The latter usually consists of either a simple and fast 1D system modelling or a sophisticated and time-consuming 3D numerical simulation. With the advancement of accurate numerical techniques and powerful computer systems, CAD software in conjunction with 3D simulation packages are becoming more and more popular nowadays. Some examples of popular 3D simulation packages are highlighted by Takei *et al.* [6]. It was concluded that the use of such software will drastically reduce the turn-around time of conventional turbocharger testing and manufacturing procedures. In addition, complex flow features can be captured more accurately by simulation. Under a CAD environment, the designer could largely reduce the time of a design cycle, as well as the cost of the manufacturing process. It therefore becomes clear that the adoption of simulation and modelling software is highly beneficial. The work described below shows the importance of using this approach.

2.2. Computational Fluid Dynamics

Margot *et al.* [7] used a computational fluid dynamics (CFD) tool to perform an analysis of compressor flow near surge conditions. This is an important area of interest as a compressor blade working in such an environment is close to stalling. When the compressor is operating near the surge line, a reversal of flow is possible due to a unsteady motion of air flow brought upon by a pressure difference within the compressor housing. In order to perform the analysis, Margot *et al.* have modelled the complete compressor unit of the turbocharger including the impeller and the housing. Due to the complexity of the geometry, the solid model was split into four different zones and meshed independently using semi-structured hexahedral meshes for the smaller domains, i.e. the impeller and tetrahedral meshes for the larger domains such as the volute. The connections between the zones were made using interfaces which link two adjacent domains with common faces. In order to perform the simulation, operating conditions had to be defined so that they represent exactly the same conditions as those of experimental test. For example, the mass flow rate and temperature boundary conditions have been defined at the compressor inlet, and a static pressure at the outlet. The rotating speed of the impeller was also adapted to cover the conditions up to the compressor surge.

In order to consider the interactions between rotating and stationary domains, special care needed to be taken to ensure that the rotating impeller affected the near-by fluids. In the study of Margot et al., a moving mesh zone comprising of the rotating impeller blades was adopted. The domain was then joined with the surrounding stationary domain using arbitrary sliding interfaces. The surge conditions were then considered in a transient calculation where the mass flow rate at the inlet boundary reduces over certain time period according to the test condition. Using a similar technique to Luján *et al*. [4], Margot et al. also performed an experimental investigation in order to characterise the surge conditions during experimental testing. The pressure upstream and downstream of the compressor blades were also measured as this is the most accurate way to identify oncoming surge conditions. In the meantime thermocouples were placed upstream of the impeller on the shroud. Hence, any temperature gradient changes would indicate the onset of surge conditions. Using the experimental results, a compressor flow map was therefore created as shown in figure 9. The blue line represents the CFD prediction that agrees fairly well with the test data. However, there is still a slight difference near choke conditions. This is probably due to the transonic flow in this area which presents strong nonlinear features which is difficult for CFD to capture accurately. Similar turbocharger flow studies were performed by Hajilouy-Benisi *et al.* [8] who used a twin-entry turbocharger as a baseline in their CFD simulations.

In contrast to the work of Margot *et al*., Hajilouy-Benisi *et al.* used an unstructured tetrahedral mesh over the entire flow domain. The interfaces were also defined between the stationary and rotating domains. The boundary conditions at the turbine inlet were total pressure and temperature, whereas at the outlet a mass flow rate boundary condition was used. Adiabatic wall conditions for the blade and other wall surfaces were also applied. The experimental data was then validated using the CFD predictions. To make a reliable comparison, the velocity profiles and the flow angles at the exit of the turbocharger were measured and used for the simulations.

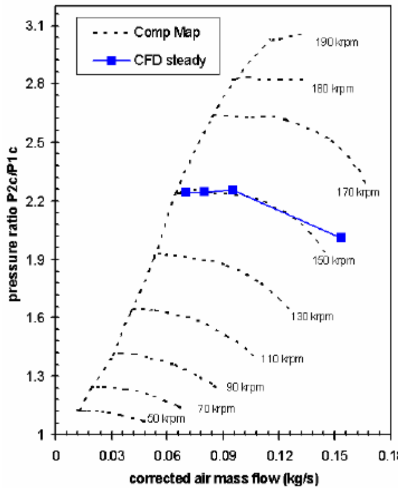


Fig. 9. Compressor flow map created by experimental data [7]

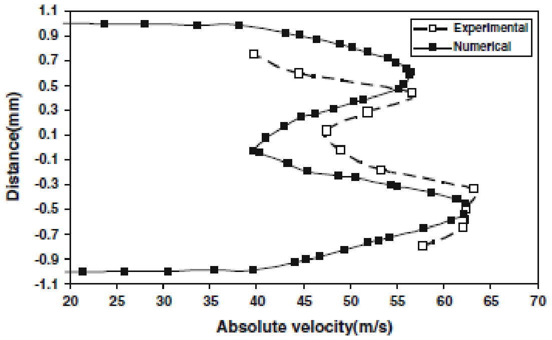


Fig. 10. A comparison between experimental measurement and CFD simulation results [8]

Figure 10 shows the comparisons. The graph displays that the simulated velocity profile correlates to the experimental data very well. Jiao *et al.* [9] also analysed the effect of using a twin-entry turbocharger of a completely different design. In contrast to Hajilouy-Benisi *et al.,* a dual volute design is applied to the compressor side of the turbocharger as shown in figure 11. Jiao et al. applied a very fine mesh with over 2.5 million elements distributed over the entire model. Similar to Margot *et al.* [7], Jiao *et al.* [9] adopted a multi-domain approach with different types of mesh elements, i.e. hexahedral meshes for the impeller whereas tetrahedral meshes for the two separate volute domains. This type of approach would be most suitable for such complex geometry. A cross-sectional plot in both volutes is produced to identify the differences in velocity magnitude between volute 2 and volute 1 as seen in figure 11. The flow velocity magnitude is found higher in volute 2 because the maximum radial velocity is located at the lower section of the impeller exit which is closest to that of volute 2. Some interesting observations were also made between a single and a dual volute compressor in terms of efficiency. The results are shown in figure 12. From the graph, it is clearly visible that the single entry vaneless volute turbocharger exhibits a smaller operating range (0.24 – 0.34 Kg/s) when compared to the dual vaneless volute (0.22 – 0.35 Kg/s). This indicates that the overall efficiency of the dual volute design would be better than the single-entry type.

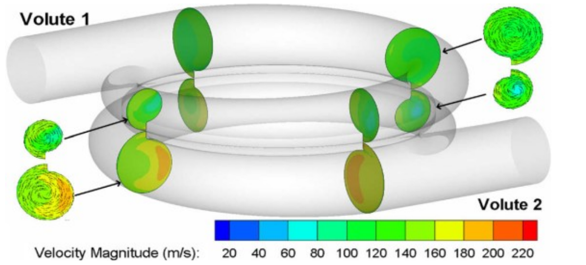


Fig. 11. The velocity magnitudes in volute 1 and volute 2, showing that an increased tangential flow in volute 1 will produce a strong air flow interruption [9]

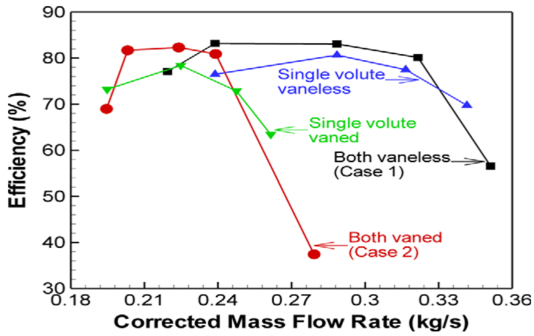


Fig. 12. Efficiency comparisons of the vaneless and vaned single and dual volute designs [9]

2.3. 1D Simulation Method

The 1D simulation method is often used in industry design to provide simple and fast solutions. The results are fairly accurate and can be used as a viable source of data validation as discussed by Fanelli *et al.* [10]. A commercially available 1D simulation package known as AVL Boost was used. The software allows the user to simulate engine dynamometer testing on a desktop PC. This allows turbocharger efficiency under exhaust gas recirculation (EGR) conditions to be studied. A valve is used to control the onset and the quantity of EGR and consequently the gas flow through the turbine is measured. To begin with Fanelli et al. used the simulation data to compare crucial compressor operation points. These are then compared to the experimental data supplied by the turbocharger manufacturer of which a close match was found. This provided a first indication that the software calibration and the set-up are indeed producing correct operating conditions. The effect of the EGR valve is analysed by performing a simple test using the calibrated 1D simulation software. This involves gradually changing the opening of the valve during a testing cycle whilst measuring different parameters of the turbocharger. The results are shown in figure 13. One can see that the experimental measured mass flow rate (in blue colour) and the 1D simulation results (in yellow/red colour) have virtually no discrepancies.

Both the experimental data and 1D simulation results show that at higher EGR valve opening angles, the mass flow rate through the turbocharger decreases. When taking the effect of EGR into consideration it becomes apparent that during low load driving conditions when the throttle is only partly opened, the maximum mass flow rate could be equivalent to that of the minimum mass flow rate when measured during full open throttle. These results clearly show the advantage of using 1D simulation software to analyse these effects.

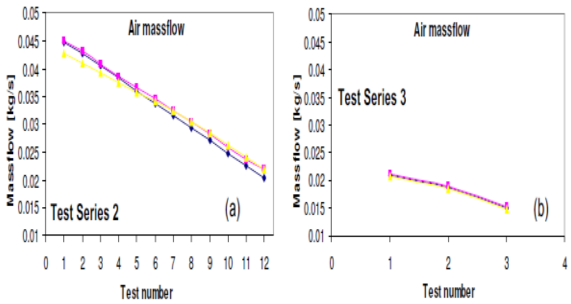


Fig. 13. Comparison between 1D simulation and experimental data from a test. (a) performed with the EGR and throttle valve fully open at 1500 RPM and (b) EGR valve fully open but throttle valve partly open respectively [10]

Chen *et al.* [11] have also used AVL Boost 1D simulation models to investigate the turbochargers performance. They proposed to use a turbocharger through-flow model in conjunction with the AVL Boost software. The through-flow model uses mass, momentum and energy conservation equations for turbocharger simulation. The model measures key variables needed to determine the operating performance of the turbocharger. In order to make it more accurate (i.e. replicating the pressure loses of the engine system into consideration), semi-empirical correlations were developed using detailed geometry of the turbocharger such as impeller exducer and inducer diameters. Due to the increased geometry complexity, certain calibration techniques needed to also be applied to further improve the accuracy of the results. To do so, a known experimentally acquired compressor mass flow rate was defined as an expected result within the 1D model. The results from the model were then compared to pre-defined experimental value. After the comparison base calculations within the 1D model were then subsequently tailored to achieve a close correlation between the two values. An example of the calibration results are shown in figure 14. The results clearly indicate that the simulation results are in good agreement with those acquired experimentally. An advantage of using the through-flow model is that a user can then examine the efficiency of the turbocharger using the geometry variables of the device, rather than relying on the manufacturers' flow maps which are not always publicly accessible. It is due to this reason that the 1D simulation models are efficient and cost effective means of studying the performance of a turbocharger. The same process can be applied to a twin-entry turbocharger system as discussed by Winkler *et al.* [12]. The 1D simulation software is also developed using GT-Power suite software to validate certain variables measured from an on-engine turbocharger test bench. The model proposed by Winkler *et al.* is based on equations governing the fluid motions. These include averaging equations of continuity, momentum and energy across the flow direction. Due to the complexity of the twin-entry volute geometry certain precautionary steps need to be taken, e.g. an orifice, which represents the two separate volute entries used within the 1D model in order to correctly represent the geometry.

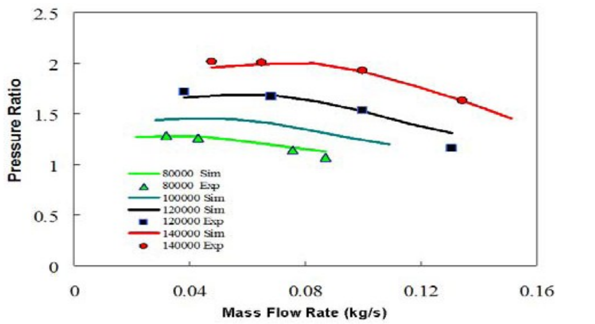


Fig. 14. Comparison between simulated and measured mass flow rates at varying turbocharger speeds using through-flow model [11]

In order to verify the experimental results, a series of turbine shaft accelerations are calculated. Using the above results, it was concluded that during engine operation, heat and mass transfers between the two different turbine entries (i.e. hub and shroud side) are a regular occurrence which affects the performance of the impeller. This is shown by the additional troths seen from the measured results during different cylinder is blow down cycles. One can see from figure 15 that the troughs are present over the whole combustion cycle. In order to characterize these differences, a series of further measurements were taken. The graph in figure 16 shows that at certain crank angles (75°, 315°), the mass flow rate just before the twin-entry geometry of the turbines volute slows down due to the heat transfer between the outer and the inner flow passages. This sudden decrease of mass flow rate affects the rotating speed of the turbine shaft as displayed by the highlighted troughs in figure 16. This is probably the reason for the differences in the simulated and measured results.

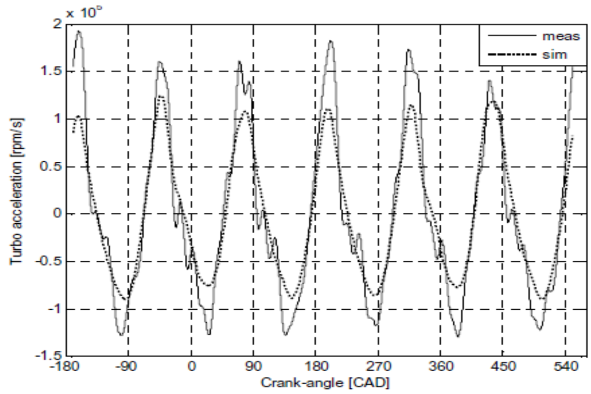


Fig. 15. Comparison between simulated and measured turbo acceleration at different crank angle increments [12]

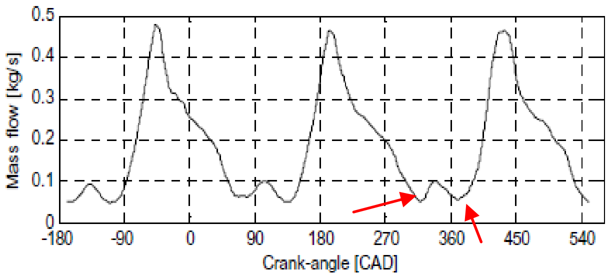


Fig. 16. Variation of simulated mass flow rate in the exhaust manifold in relation to varying crank-angle [12]

An important aspect which is apparent from the above studies is that the engine size used for simulation and experimental testing usually consists of a larger 2L six cylinder engine. However, a current trend in industry is to apply "engine downsizing". This means that manufacturers are more favourable to develop smaller capacity engines. By adopting suitable turbochargers, smaller engines can actually produce similar work rates as naturally aspirated larger engines. For example, Stan and Taeubert [13] have shown that the popularity of hybrid propulsion systems can be matched by turbocharging and downsizing. They believed that using such engine configurations could provide increased torque and torque response while reducing fuel consumption and certain exhaust gas emissions. During downsizing, these traits are achieved due to the increase in mean thermal efficiency of the engine by using a high effective mean pressure. The increase in torque and pressure is provided by pressure boosting the engine i.e. turbocharging. It was also stated that using a turbocharger provides the engine with sufficient brake mean effective pressure for urban driving conditions where high load and speeds are necessary. In order to provide a comparison between engines of different configurations, the fuel consumption results from 1D simulation are shown in figure 17. It can be seen from the graph that at lower engine speeds (1000 - 1500 RPM), both highlighted engines produce similar fuel consumption trends.

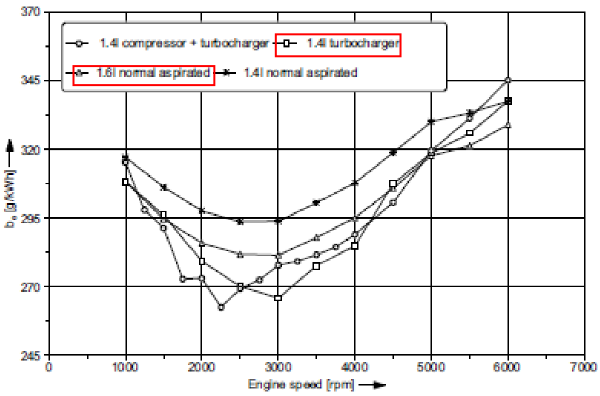


Fig. 17. Comparison of fuel consumption between four different engine configurations [13]

It is only at middle range engine speeds (1500 - 4500 RPM) where the effect of the turbocharger becomes apparent. The fuel consumption clearly drops in this speed range, but rises again at higher engine speeds of 4500 to 6000 RPM. This indicates that during "normal" driving conditions i.e. mid-range engine speeds, the smaller displacement 1.4L turbocharged engine exhibits a better fuel consumption. Therefore it has shown why engine downsizing is a positive trend.

3. Ongoing Activities and Future Trends

3.1. VanDyne SuperTurbocharger (SuperTurbo)

Chadwell and Walls [14] argued that smaller down-sized turbocharged engines are less efficient at lower engine speeds than normally-aspirated (NA) engines. This might be true due to the function of the turbocharger unit. At lower engine speeds, the transient response of the turbocharger is limited because of the inertia force of the turbine impeller which is directly affected by the energetic exhaust gases. To overcome this problem, Chadwell and Walls suggested a new technology known as SuperTurbo. This type of turbocharger can be coupled to a continuously variable transmission (CVT) which is directly run via the engines crankshaft. This type of gearbox allows the turbocharger to act as a supercharger boosting device even at lower engine speeds. It does this by utilising the energy of the crankshaft to accelerate the turbine shaft, when the exhaust gas energy is not sufficient, i.e. at low engine loads and speeds. The results acquired by Chadwell and Walls are recorded using GT-Power 1D simulation software. In order to compare to a larger naturally-aspirated engine with a down-sized SuperTurbo engine, the following results are presented and discussed.

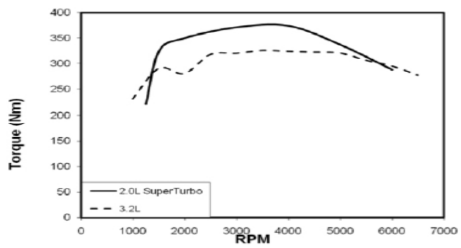


Fig. 18. Comparison of torque between a SI 3.2L NA engine and a 2L SuperTurbo engine over a range of speeds [14]

In figure 18, the smaller SuperTurbo equipped engine clearly shows better torque characteristics over a wide range of engine speeds and it again proves that turbocharged down-sized spark ignition engines could provide better performance. Petitjean *et al.* [15] also studied the trend of engine downsizing using turbochargers. They compared different naturally-aspirated (NA) and turbo-charged (TC) vehicle models of the year 2002-2003. They concluded that smaller turbo-charged engines produce similar power outputs as the larger naturally-aspirated engines but at considerably lower fuel consumption rates.

3.2. Variable Geometry Turbochargers

The use of SuperTurbos is a relatively new technology which is currently not being commercially used by any manufacturer. This is because similar results can be achieved using variable geometry turbochargers (VGT) which are commonly used throughout the automotive industry. Variable geometry turbochargers have conventional volute passages but the flow path of the gas is regulated using pivoted nozzle vanes. The vanes can be manipulated into different angles which determine how much exhaust gas flows into the turbine impeller i.e. how much energy is transferred from the exhaust gas to the turbine impeller. Hawley *et al.* [16] studied the emission characteristics of a compression ignition (CI) engine using VGT technology. The results obtained were used as a comparison between standard fixed geometry turbochargers (FGT) and variable geometry turbochargers (VGT) units and their effect on a 1.8L compression ignition engine manufactured by the Ford motor company.

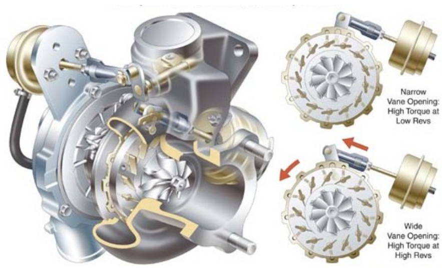


Fig. 19. Variable geometry turbocharger (www.volvo.com)

Based on the study, it was claimed that the VGT unit will decrease the quantity of NO_x in the exhaust gas emissions when compared to an FGT unit. In order to quantify this finding, the amount of NO_x produced at varying engine speeds using a VGT unit are shown in figure 20. It is apparent that the adoption of a variable geometry turbocharger increases the engine performance while reducing NO_x emissions. For example, at 2500 RPM the FGT equipped engine produces 360 g/hr whereas the VGT engine produces less than 60 grams per hour. Arnold [17] also discussed the advantage of using variable geometry turbochargers. It was mentioned that using exhaust gas recirculation (EGR) systems in conjunction with a VGT could reduce the formation of NO_x during the combustion cycle. The system implemented is known as a high-pressure EGR loop. During such a cycle, the exhaust gases are cooled down and re-circulated back into the combustion chamber.

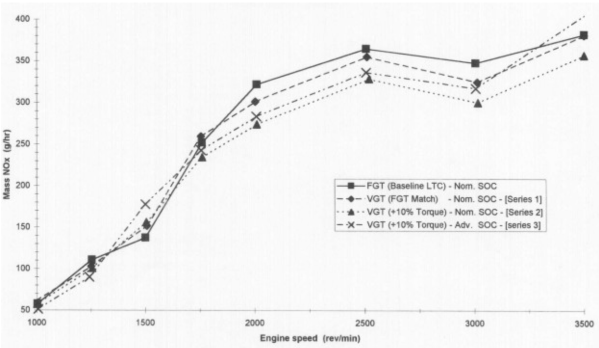


Fig. 20. The NO_x production over a range of engine speeds [16]

The cooled re-circulated gas functions as an energy absorber and therefore reduces the combustion temperatures which in turn reduce NO_x quantities. The role of the variable geometry turbocharger is very important as it acts as an additional driver for the EGR system. By closing the vanes of the turbocharger unit at low engine speeds, an increase in boost is produced which accelerates the air flow towards the turbine. The extra air flow increases the back pressure on the turbine, creating a pressure difference between the turbine and manifold. This increases the exhaust flow through the EGR loop therefore producing less NO_x emission.

4. Conclusion

The literature review study presented in this paper provides a general outline as to how engine testing techniques can be used to perform the necessary experimental analysis of the turbocharger. First, it is clear that the application of 'hot' gas technology is far more realistic as it can replicate the operating conditions of the turbocharger accurately. Second, it is also common to use the exhaust gas at low temperature (i.e. 'cold') to drive the turbine due to its simplicity in test rig setup. Using suitable down-sized engines can generally achieve better engine outputs and fuel economy. Furthermore, different turbocharger designs used in conjunction with the down-sized engines can be analysed using the experimental testing procedures outlined in this paper. The results can then be validated using efficient 1D simulation methods and/or accurate 3D CFD simulation tools. This allows decisions to be made as to which type of turbocharger better suites the economic advances and energy constrains required in the years to come.

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