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Turbocharging of Diesel Engine for Improving Performance and Exhaust Emissions: A Review

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Abstract: *Turbochargers are used throughout the automotive industry to enhance the output of an internal combustion engine without increasing the cylinder capacity. The application of such a mechanical device enables automotive manufacturers to adopt smaller displacement engines, commonly known as engine downsizing. 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 automotive industry applications. The aim of this paper is to provide a review on the current techniques used in turbocharging to improve the engine efficiency and exhaust emissions as much as possible.*

Keywords: *Engine Performance, Exhaust Emission, Supercharger, Turbocharger, Volumetric Efficiency.*

I. Introduction

Turbochargers were originally known as turbosuperchargers when all forced induction devices were classified as superchargers. Nowadays the term "supercharger" is usually applied to only mechanically driven forced induction devices. The key difference between a turbocharger and a conventional supercharger is that the latter is mechanically driven by the engine, often through a belt connected to the crankshaft, whereas a turbocharger is powered by a turbine driven by the engine's exhaust gas. Compared to a mechanically driven supercharger, turbochargers tend to be more efficient. Turbochargers are commonly used on truck, car, train, aircraft, and construction equipment engines [1] [2].

1.1 Operating Principle

In normally aspirated piston engines, intake gases are pushed into the engine by atmospheric pressure filling the volumetric void caused by the downward stroke of the piston (which creates a low-pressure area), similar to drawing liquid using a syringe. The amount of air actually sucked, compared to the theoretical amount if the engine could maintain atmospheric pressure, is called volumetric efficiency. The objective of a turbocharger is to improve an engine's volumetric efficiency by increasing density of the intake gas (usually air). The turbocharger's compressor draws in ambient air and compresses it before it enters into the intake manifold at increased pressure. This results in a greater mass of air entering the cylinders on each intake stroke. The power needed to spin the centrifugal compressor is derived from the kinetic energy of the engine's exhaust gases. The pressure volume diagram shows the extra work done by turbocharging the diesel engine [1-9].

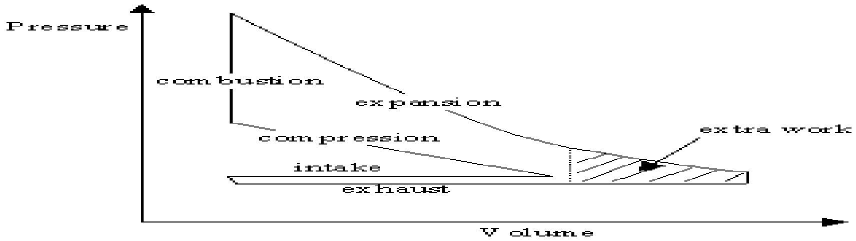

Fig 1: Pressure volume diagram of diesel engine with turbocharging [2]

II. Current Status Of Turbocharger Researches

Turbochargers are widely used in the automotive industry to enhance the volumetric efficiency and reduce the exhaust emissions. Researchers are continuously doing advancements in the turbocharging technology to improve its efficiency and reduce the exhaust emissions of automotive to meet the environmental related rules laid down by the government of different nations. A review of novel turbocharger concepts for enhancements in efficiency by many researchers is done in the following sub-headings.

2.1 Impact of Turbocharger Non-Adiabatic Operation on Engine Volumetric Efficiency and Turbo Lag

Shaaban et al [10] studied that turbocharger performance significantly affects the thermodynamic properties of the working fluid at engine boundaries and hence engine performance. Heat transfer takes place under all circumstances during turbocharger operation. This heat transfer affects the power produced by the turbine, the power consumed by the compressor and the engine volumetric efficiency. Therefore, non-adiabatic turbocharger performance can restrict the engine charging process and hence engine performance. Author's research work investigated the effect of turbocharger non-adiabatic performance on the engine charging process and turbo lag. Two passenger car turbochargers were experimentally and theoretically investigated. The effect of turbine casing insulation was also explored. The research investigation shows that thermal energy is transferred to the compressor under all circumstances. At high rotational speeds, thermal energy is first transferred to the compressor and latter from the compressor to the ambient. Therefore, the compressor appears to be adiabatic at high rotational speeds despite the complex heat transfer processes inside the compressor. A tangible effect of turbocharger non-adiabatic performance on the charging process is identified at turbocharger part load operation. The turbine power is the most affected operating parameter, followed by the engine volumetric efficiency. Insulating the turbine is recommended for reducing the turbine size and the turbo lag. Turbocharger performance significantly affects the overall performance of turbocharged engines. Turbocharger operation involves heat transfer under all circumstances. Even if the turbocharger casing is well insulated, heat transfer takes place from the turbine to the lubricating oil [11-13] or from the oil to the compressor at low rotational speeds. Malobabic et al [14] reported that the turbocharger will operate at a considerably lower speed due to non-adiabatic operation which in turn influences the charging process. Non-adiabatic turbocharger operation can also increase the turbo lag because the time required to accelerate the turbocharger from angular velocity ω_1 to ω_2 is given by

$$\Delta t_{\text{acceleration}} = I_{\text{rotor}} \int_{\omega_1}^{\omega_2} \frac{\omega \, d\omega}{\dot{W}_T - \dot{W}_C} \quad \text{………………..} (1)$$

Turbo lag increases if the actual non-adiabatic turbocharger operation involves a decrease in the turbine power and an increase of the compressor power. Moreover, the turbo lag decreases if the turbine can produce the same power at smaller size (smaller rotor inertia). Rakopoulos et al [15] reported that turbocharger lag is the most notable off-design feature of diesel engine transient operation that significantly differentiates the torque pattern from the respective steady state conditions. It is difficult to measure the non-adiabatic turbine and compressor actual power due to heat transfer between the turbocharger components as well as between the turbocharger and the ambient. The high exhaust gas temperature, the very high rotational speed and the shaft motion associated with the use of the sliding hydraulic bearing are some factors that increase the difficulty of measuring the compressor torque under non-adiabatic operating conditions. Therefore, non-adiabatic turbocharger operation is investigated using either thermodynamic models or CFD simulation.

Rautenberg and Kammer [16] modeled the non-adiabatic compressor performance by decomposing the amount of thermal energy transfer to the compressor into three portions. The first portion takes place before the impeller, the second portion takes place during the compression process in the impeller and the third portion takes place after the impeller. Hagelstein et al [17] simplified the model of Rautenberg and Kammer [16] and decomposed the amount of thermal energy transfer to the compressor into two portions only. The first portion takes place before the compressor impeller, while the second portion takes place after the compressor. They considered the compression process in the impeller to be adiabatic. Cormerais et al [18] experimentally and analytically investigated the process of heat transfer inside the turbocharger. Galindo et al. [19] presented an analytical study of a two stage turbocharging with inter and after cooler. They considered the amount of thermal energy transfer in the turbine side before gas expansion in the turbine. Bohn et al. [20–21] performed 3D conjugate calculation for a passenger car turbocharger. Eriksson et al. [22] modeled a spark ignition turbocharged engine with intercooler. They neglect the effect of heat transfer in the turbocharger. Serrano et al. [23] presented a model of turbocharger radial turbines by assuming that the process undergone by the gas in the turbine is adiabatic but irreversible. Most of the previous publications concern with the amount of thermal energy exchange between the turbocharger components or even assume the turbocharger to be adiabatic. These investigations are important for engine modeling programs. Shaaban et al [10] investigated the probable effect of actual turbocharger non-adiabatic operation on engine volumetric efficiency and turbo lag. They modeled and estimated the actual turbine and compressor power under real non-adiabatic operating conditions. They also explored the increase in compressed air temperature due to thermal energy transfer to the compressor and estimated its subsequent effect on engine volumetric efficiency. Experimental investigations were performed on the small combustion chamber test rig of the University of Hanover as shown in figure 2.

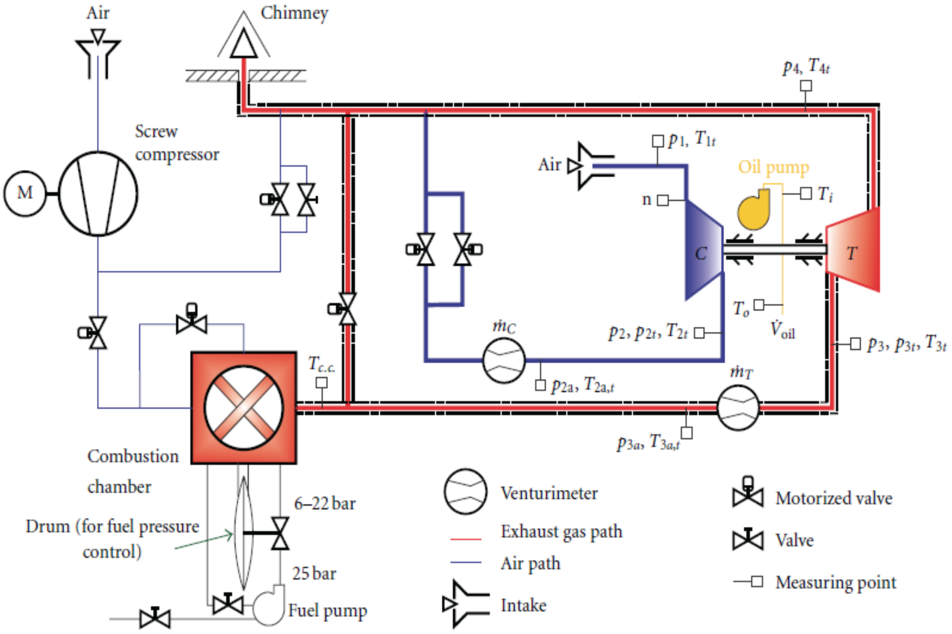

Fig 2: Schematic of the small combustion chamber test rig [10]

The research investigated the effect of non-adiabatic turbocharger performance on engine volumetric efficiency and turbo lag. Thermostat significant effect of turbocharger non-adiabatic performance on turbo lag is identified at the turbine. Experimental data show 55% decrease of the turbine actual power at 60000 rpm due to thermal energy transfer from the turbine. Experimental data also show that insulating the turbine significantly improves the non-adiabatic turbine performance. It is therefore recommended to insulate the turbine and provide compressor cooling in order to improve the turbocharger non-adiabatic performance and hence the engine performance. Experimental data also show that insulating the turbine results in 2.4% increase of the exhaust gas temperature at turbine exit.

2.2 Effect of Variable Geometry Turbocharger

Cheong et al [24] studied the effect of variable geometry turbocharger on HSDI diesel engine. Power boosting technology of a High Speed Direct Injection (HSDI) Diesel engine without increasing the engine size has been developed along with the evolution of a fuel injection system and turbocharger. Most of the turbochargers used on HSDI Diesel engines have been a waste-gated type. Recently, the Variable Geometry Turbocharger (VGT) with adjustable nozzle vanes is increasingly used, especially for a passenger car in European market. Cheong et al described the first part of the experimental investigation that has been undertaken on the use of VGT, in order to improve full load performance of a prototype 2.5 liter DI Diesel engine, equipped with a common rail system and 4 valves per cylinder. The full load performance result with VGT was compared with the case of a mechanically controlled waste-gated turbocharger, so that the potential for a higher Brake Mean Effective Pressure (BMEP) is confirmed. Within the same limitation of a maximum cylinder pressure and exhaust smoke level, the low speed torque could be enhanced by about 44% at maximum.

In power boosting of engines, the application of conventional turbochargers could realize only a limited improvement because it is effective in a narrow flow range. Charging effect of a conventional turbocharger is too poor in a low flow range below the matching point to realize a high power output at a low engine speed region.The waste-gated turbochargers that bypass some portion of an exhaust gas were generally used for boosting high speed Diesel engines. But, recently, VGT (Variable GeometryTurbocharger) is increasingly used in HSDI Diesel engines, which makes it possible to raise the boost pressure even at lower engine speeds, together with the reduction of pumping losses at higher engine speeds, compared with a waste-gated turbocharger. In his study, aVGT was applied to an HSDI Diesel engine, and the improvement of a full load performance over the case with a mechanically controlled waste-gated turbocharger was confirmed. The test engine was a prototype 2.5 liter direct injection diesel engine, equipped with a common rail fuel injection system with a maximum rail pressure of 1350 bar and 4 valves per cylinder. The VGT tested in this study was a Variable Nozzle Turbine (VNT) type, and the vane angle of the turbine nozzle can be varied, as shown in Fig. 3.

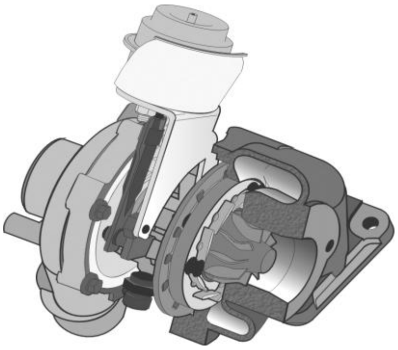

Fig 3: Schematic diagram of VGT [24]

Cheong et al found that with the use of the VGT, it was possible to increase the charge air mass by about 10 ~ 20 % at a low speed range. As a result of this, the exhaust smoke was reduced and the fuel consumption was improved with the same fuel delivery and start timing of injection. At low speed, over 40 % of additional torque increase was observed within the same exhaust smoke, the cylinder pressure, and the exhaust gas temperature limit, by adjusting the boost pressure and fuel delivery with the VGT. In the medium engine speed range, there was a marginal gain in the fuel consumption for the VGT, with the same fuel delivery. When the boost pressure and fuel delivery were increased, more torque could be achieved with the expense of the deterioration in fuel consumption. This is because the injection timing should be retarded not to exceed the maximum cylinder pressure limit. At high engine speed, with the same fuel delivery, the rated power can be enhanced by 3.5 %, mainly caused by the reduction of pumping loss. However, within the same boundary conditions, the power increase for the VGT could reach about 7.9 %. Cheong concluded that the application of VGT could provide HSDI Diesel engines with a great potential for full load performance, especially at low engine speed.

2.3 Availability analysis of a turbocharged diesel engine operating under transient load conditions

Rakopoulos and Giakoumis [25] had done the availability analysis of a turbocharged diesel engine operating under transient load conditions. A computer analysis was developed for studying the energy and availability performance of a turbocharged diesel engine, operating under transient load conditions. The model incorporates many novel features for the simulation of transient operation, such as detailed analysis of mechanical friction, separate consideration for the processes of each cylinder during a cycle (multi-cylinder model) and mathematical modeling of the fuel pump. This model was validated against experimental data taken from a turbocharged diesel engine, located at the authors' laboratory and operated under transient conditions. The availability terms for the diesel engine and its subsystems were analyzed, i.e. cylinder for both, the open and closed parts of the cycle, inlet and exhaust manifolds, turbocharger and aftercooler. The analysis revealed how the availability properties of the diesel engine and its subsystems develop during the evolution of the engine cycles, assessing the importance of each property. In particular the irreversibilities term, which was absent from any analysis based solely on the first-law of thermodynamics, was given in detail as regards transient response as well as the rate and cumulative terms during a cycle, revealing the magnitude of contribution of all the subsystems to the total availability destruction.

The experimental investigation was carried out on a 6-cylinder, IDI (indirect injection), turbocharged and aftercooled, medium-high speed diesel engine of marine duty coupled to a hydraulic brake, located at the authors' laboratory. A high-speed data acquisition system was setup for measuring engine and turbocharger variables performance, under both steady-state and transient operation. The transient behavior of the engine was predicted adequately by the developed code, despite the long non-linear brake loading times and the IDI nature of the engine. From the experimental data it was concluded that the availability term for the heat loss to the cylinder walls increases substantially during the transient event (increased potential for work recovery), but the reduced term returns to the initial value after a peak in the middle of the transient event. The availability of the exhaust gases from the cylinder increase significantly after an increase in load (increased potential for work recovery). Cylinder irreversibilities decrease, proportionally, after a ramp increase in load due to the subsequent increase in fueling, while combustion irreversibilities account for at least 95% of the total cylinder ones. Every operating parameter that can decrease the amount of combustion irreversibilities (e.g. greater cylinder wall temperature) was favorable according to second-law and can lead to increased piston work. Exhaust manifold irreversibilities increase significantly during a load increase, reaching as high as 15% of the total ones, highlighting another process which needs to be studied for possible efficiency improvement. This increased amount of irreversibilities arises mainly from the greater pressures and temperatures due to turbocharging, which have already lowered the reduced magnitude of combustion irreversibilities. The inlet manifold irreversibilities, on the other hand, were of lesser and decreasing importance during the transient event. Turbocharger irreversibilities, though only a fraction of the (dominant) combustion ones, not negligible, while the intercooler irreversibilities steadily remain of lesser importance (less than 0.5% of the total ones) during a load change.

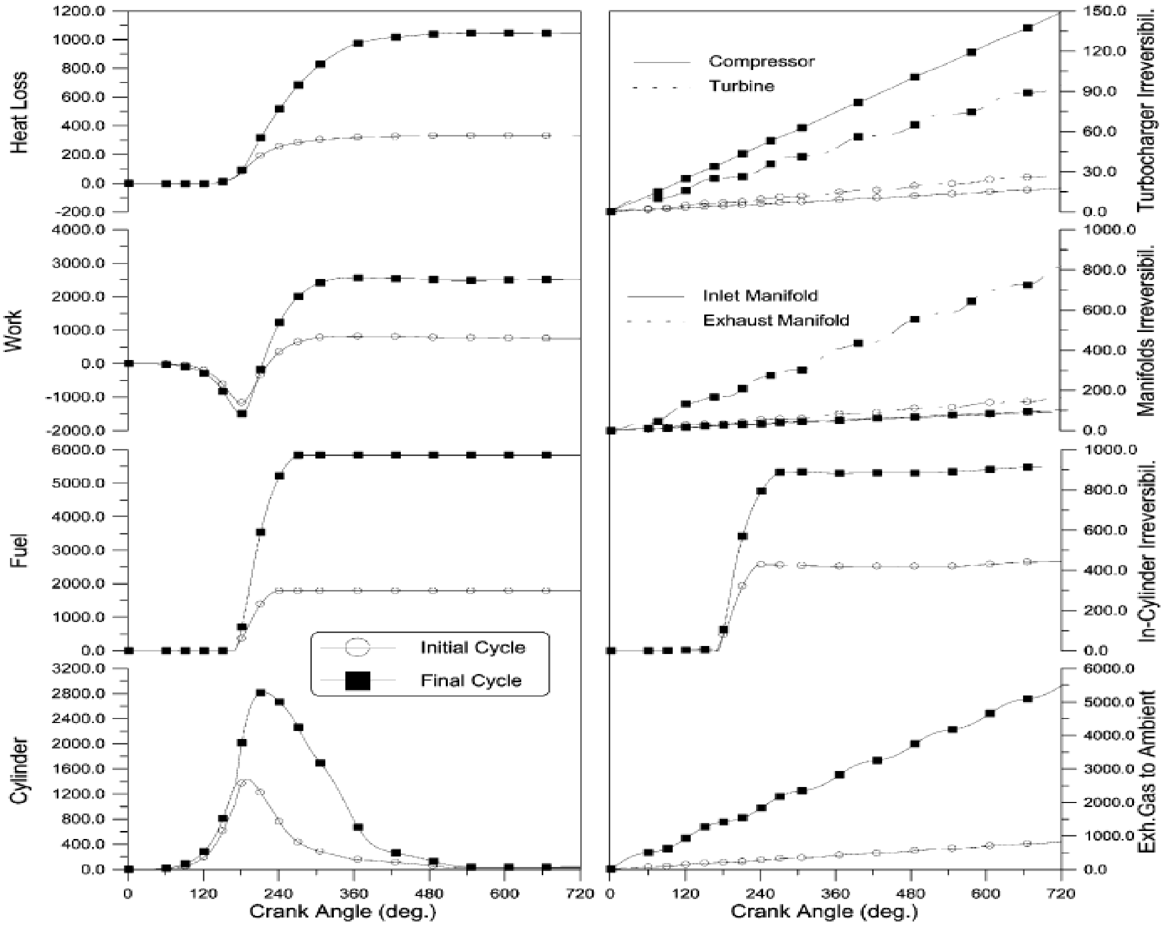

Fig 4: Development in the cumulative (J) availability terms of diesel engine and its subsystems, at the initial and final steady-state conditions. [25]

2.4 Effect Of Intercooler On Turbocharged Diesel Engine Performance

Increased air pressure outlet compressor can result in an excessively hot intake charge, significantly reducing the performance gains of turbo charging due to decreased density. Passing charge through an intercooler reduces its temperature, allowing a greater volume of air to be admitted to an engine. Intercoolers have a key role in controlling the cylinder combustion temperature in a turbocharged engine. Naser et al [26] through their own worked out programmed code in MATLAB presented the effect of intercooler (as a heat exchange device air-to-liquid with three different sizes and overall heat transfer coefficient and one base) at a multi-cylinder engine performance for operation at a constant speed of 1600 RPM. They presented the simulation predictions of temperature and pressure in cylinder for three types of intercoolers. Also they presented the pressure and temperature in intake, exhaust manifold and other performance. From the experimental data, the authors concluded that the maximal temperature in engine cylinder was decreasing from 1665.6 K at SU =1000 to 1659.2 K at SU=1600. Also intercooler performance was increased by increasing the design parameters. Intercooler efficiency was 0.92% at SU =1000 and 0.98% at SU=1600.

Canli et al [27] also studied the intercooling effect on power output of an internal combustion engine. In his study, a diesel engine was considered and it was evaluated whether it was equipped with either a turbocharger or both a turbocharger and a super intercooler. Using thermodynamics laws and expressions, the power output of the engine was analytically examined by changing intercooling features such as pressure drop values and engine revolution at full load. Results were presented and interpreted as power (kW) and downsizing of the engine volume values (m^3). In this study Canli et al concluded that engine power can be increased to 154% by ideal intercooler while single turbocharger without intercooler can only increase 65% engine power output. The power output of engine at different RPM is shown in the graph below.

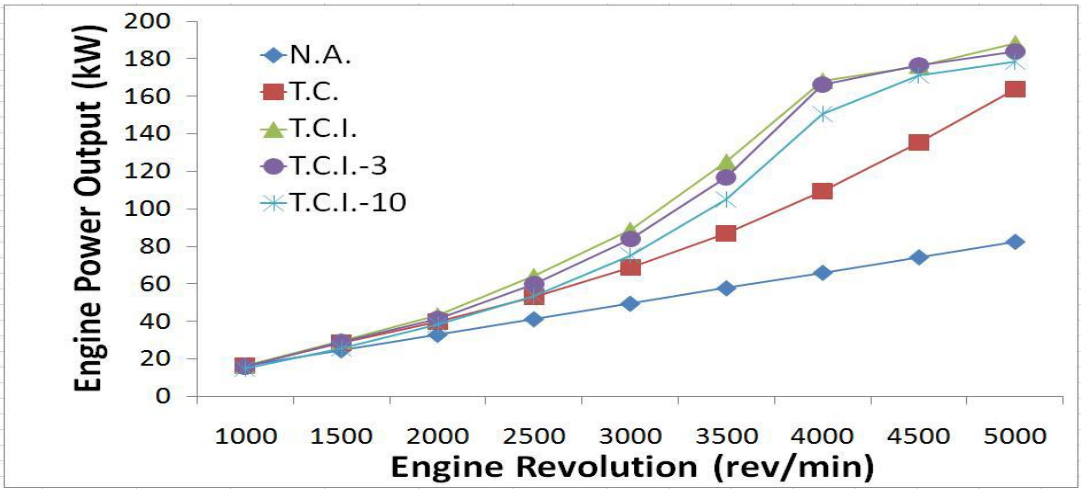

Fig 5: Power output values of the engine due to supercharging, N.A. Naturally aspirated engine, T.C. Engine with turbocharger and without intercooler, T.C.I. Engine with turbocharger and intercooler, T.C.I.3 Engine with turbocharger and intercooler and 3 percent pressure drop, T.C.I.10 Engine with turbocharger and intercooler and 10 percent pressure drop [27]

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

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 [28] and Hawley [29] 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 [30] suggested a new technology known as a Super Turbo 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 [31] and Petitjean et al [32]. Two-stage turbocharging as discussed by Watel et al [33] 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 6) 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 ofthe 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 [34]. 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 studied by Kusztelan et al [35].

Fig 6: Turbocharger cut-away highlighting the twin-entry volute geometry, allowing differently pulsed exhaust gases to remain separate [35]

Kusztelan et al, through the AVL Boost engine simulation code,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 amaximum 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 was more beneficial during low engine speeds as the turbocharger delay time would be reduced making the engine more responsive to driver input. The"drivability" of the vehicle has therefore also improved. Figure 7 shows the increment in the engine power output of a 2.0L CI engine using a twin-entry turbocharger.

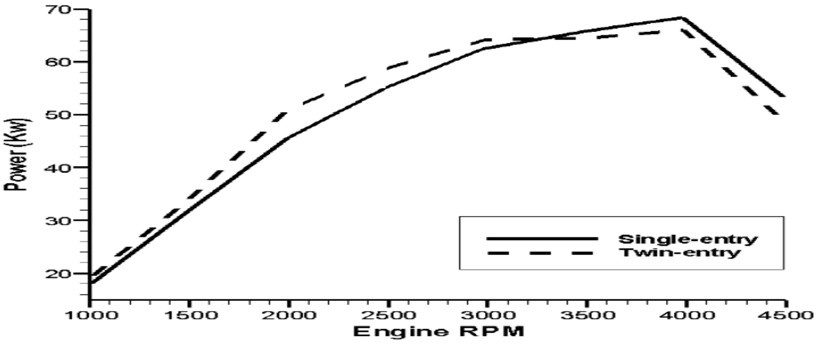

Fig 7: Increased engine power output of a 2.0L CI engine using a twin-entry Turbocharger [35]

III. Conclusion

The literature review study presented in this paper provides a general outline of the advancements in the turbocharging technology to enhance the engine performance. In last two decades various new advancements are done to improve the power output of an engine and to reduce its emissions by making some changes and installing some additional accessories like intercooler in the turbocharging technology. This will carry on in the future because in coming days there will be an increment in the demand of fuel efficient engines with more power and minimum emissions and this is possible with continuous advancements in turbocharging technology.

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