FOREMOST, I WOULD LIKE TO EXPRESS MY SINCERE GRATITUDE TO MY MENTOR MR. JAROSLAV PEKAR FOR THE CONTINUOUS SUPPORT OF MY MASTER THESIS RESEARCH, FOR HIS PATIENCE, MOTIVATION, ENTHUSIASM, AND IMMENSE KNOWLEDGE. HIS GUIDANCE HELPED ME IN ALL THE TIME OF RESEARCH AND WRITING OF THIS THESIS.

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<td>Supervisor</td>
<td>Prof. Ing. Jan Macek</td>
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**Abstract**: Turbochargers have been one of the most innovative discoveries in the past century (especially for Automobile Industry). It provided us with a method to re-utilize the wasted exhaust and produce more power. With advancement in technology and stringent emission norms, the need for refinement in turbo technology has also increased. Advanced turbochargers like e-Charger, not only minimize the limitations of a conventional turbochargers but can also re-utilize excess boost to recharge their power source. Primary aim of the thesis was to develop a Simulink model of an e-Charger, which can work in combination of a waste-gated turbocharger. In addition to it, sensitivity study was carried out for degradation of turbocharger flow rate and efficiency. Such a study provides us with the trends, that can be correlated to actual failures. These variations in flow rate and efficiency can be considered to be indicators, for prognostic analysis. This approach can help in identifying the failure, before it can actually happen and with the advent of autonomous vehicle, such an approach of failure analysis would be vital.

**Keywords**: Turbocharger, e-Charger, Simulink, Prognostic, Autonomous Vehicles
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**ABBREVIATIONS AND SYMBOLS**

### Abbreviations

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<tr>
<td>VGT</td>
<td>Variable Geometry turbocharger</td>
</tr>
<tr>
<td>E-Charger</td>
<td>Electric assisted turbocharger</td>
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<tr>
<td>TS</td>
<td>Honeywell Transportation System</td>
</tr>
<tr>
<td>OEM</td>
<td>Original equipment manufacturer</td>
</tr>
<tr>
<td>LQR</td>
<td>Linear Quadratic Regulator</td>
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<tr>
<td>ICE</td>
<td>Internal combustion engine</td>
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<tr>
<td>EGR</td>
<td>Exhaust gas recirculation</td>
</tr>
<tr>
<td>HP</td>
<td>High pressure</td>
</tr>
<tr>
<td>LP</td>
<td>Low pressure</td>
</tr>
<tr>
<td>FGT</td>
<td>Fixed geometry turbine</td>
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<tr>
<td>ECU</td>
<td>Electronic Control Unit</td>
</tr>
<tr>
<td>MEP</td>
<td>Mean effective pressure</td>
</tr>
<tr>
<td>PID</td>
<td>Proportional integral derivative controller</td>
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<tr>
<td>WHTC</td>
<td>World Harmonized Transient Cycle</td>
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### Symbols

<table>
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<tbody>
<tr>
<td>$\psi$</td>
<td>Dimensionless head parameter</td>
</tr>
<tr>
<td>$T_{in}$</td>
<td>Inlet temperature</td>
</tr>
<tr>
<td>$U_c$</td>
<td>Rotor tip blade speed</td>
</tr>
<tr>
<td>$\phi$</td>
<td>Normalized flow rate</td>
</tr>
<tr>
<td>$M$</td>
<td>Mach number</td>
</tr>
<tr>
<td>$b_i$</td>
<td>Coefficients of polynomial</td>
</tr>
<tr>
<td>$P$</td>
<td>Power</td>
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<tr>
<td>$h$</td>
<td>Enthalpy</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>Slip factor</td>
</tr>
<tr>
<td>$Ntc$</td>
<td>Turbocharger speed</td>
</tr>
<tr>
<td>$\overline{N}_{TC}$</td>
<td>Compressor rotational speed parameter</td>
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$m_c$  Air mass
$A$  Area
$l$  Length
$m_{c,\text{asp}}$  Maximum mass flow
$\Pi$  Pressure ratio
$\Pi_c$  Compressor pressure ratio
$At$  Effective turbine area
$\eta_c$  Compressor efficiency
$P_c$  Compressor power
$D_c$  Mass flow through compressor
$Pt$  Turbine power
$N_{c,\text{corr}}$  Corrected Compressor speed
$U_{\text{VGT}}$  VGT actuator position
$m_{\text{asp}}$  Aspirated flow
$\beta_{\text{gas}}$  Mass flow through the compressor and turbine
$\eta_f$  Fuel conversion efficiency
$\gamma$  Ratio of specific heat for air
$V_d$  Displaced volume
$\eta_v$  Volumetric efficiency
$Q_{\text{HV}}$  Heating value of fuel
$N$  Revolution per minute
$\rho_{a,i}$  Inlet air density
$F/A$  Heating value of fuel
$\dot{Q}$  Heat transfer rate into the control volume
$\dot{W}$  Shaft work transfer rate out of control volume
$\dot{m}$  Mass flow rate
$h$  Specific enthalpy
$ho$  Total enthalpy
$C^2/2$  Specific kinetic energy
$gz$  Specific potential energy
$Wt$  Turbine work
$W_c$  \hspace{1cm} \text{Compressor work}  \\
$\eta_t$  \hspace{1cm} \text{Turbine efficiency}  \\
$\eta_c$  \hspace{1cm} \text{Compressor efficiency}  \\
$\tau_c$  \hspace{1cm} \text{Compressor torque}  \\
$\bar{\omega}_c$  \hspace{1cm} \text{Corrected compressor speed}  \\
$\dot{m}_{c,\text{cor}}$  \hspace{1cm} \text{Corrected compressor mass flow}  \\
$\Pi_t$  \hspace{1cm} \text{Turbine expansion ratio}  \\
$\bar{\omega}_t$  \hspace{1cm} \text{Corrected turbine speed}  \\
$\dot{m}_{t,\text{cor}}$  \hspace{1cm} \text{Corrected turbine mass flow}  \\
$\eta_t$  \hspace{1cm} \text{Turbine efficiency}  \\
$\tau_t$  \hspace{1cm} \text{Turbine torque}  \\
$I_{TD}$  \hspace{1cm} \text{Turbocharger rotational inertia}  \\
$P_t$  \hspace{1cm} \text{Turbine Power}  \\
$P_c$  \hspace{1cm} \text{Compressor power}  \\
$N_{sp}$  \hspace{1cm} \text{Speed parameter}  \\
$\Delta p$  \hspace{1cm} \text{Pressure difference}  \\
$A$  \hspace{1cm} \text{Resistance to the exhaust}  \\
$T_{2i}$  \hspace{1cm} \text{Intercooler temperature}  \\
$N e'$  \hspace{1cm} \text{Engine RPM}  \\
$mA'$  \hspace{1cm} \text{Inlet mass flow of air}  \\
$mF'$  \hspace{1cm} \text{Mass flow of fuel}  \\
$P2m$  \hspace{1cm} \text{Intake manifold out. Pressure}  \\
$\eta_{\text{int}}$  \hspace{1cm} \text{Intercooler efficiency}  \\
$m_{\text{A,\text{int}}}$  \hspace{1cm} \text{Flow rate through intercooler}  \\
$Cd$  \hspace{1cm} \text{Discharge coefficient}  \\
$\rho_{\text{int}}$  \hspace{1cm} \text{Intercooler density}  \\
$A_w$  \hspace{1cm} \text{Waste-gate area}  \\
$P_m$  \hspace{1cm} \text{Manifold outlet pressure}  \\
$P_2$  \hspace{1cm} \text{Compressor outlet Pressure}
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1. INTRODUCTION

Since the past century, the world has seen a huge surge in industrialization. This surge has led to the development of technologies which enabled humans to have a more convenient life. Due to this immense development cycle, the life of every individual has changed drastically. However, this rapid industrialization required the burning of fossil fuels to energize its growth, which led to the birth of a new menace known to the people of the twenty-first century as pollution. As industrialization grew, it, on the one hand, provided humans with a tool to advance further in the field of science and technology, yet on the other, armed them with an instrument to destroy mother nature. Emissions from different industries and automobiles are thus providing the menace of the pollution with a chance to grow immensely in its destructive power.

One may ask what exactly is the “emission”, the entire world is so concerned about. It is a deadly mixture of harmful gases like Carbon Monoxide (CO), Non-Methane Hydrocarbons, Nitrogen Oxides (or more popularly known as NOx), Particulate matter, Greenhouse gases (CO₂, N₂O, CH₄) and Volatile organic compound fuel vapor. The automobile industry has been one of the major contributors to emissions. Emission has not only been detrimental to the environment; it also has an adverse effect on the health of the people. As a result, in order to curb this situation, the governments across the world have opted for stricter emission norms, for example Euro norms (European Union), Bharat Stage (India), Environmental Protection Agency (America) and Central Environment Council (Japan), etc.

These norms strive towards increasing the overall fuel efficiency and reducing powertrain emission. As we reached the end of the twentieth century, these norms and fuel efficiency requirements have become more stringent. This strictness helped in reducing the emission up to a substantial level. However, these norms also offered numerous challenges to the automotive industry since the demand for power and performance from the consumers was also showing an upward trend. This led to the automotive industry developing technologies aimed at making the vehicles more efficient with minimum emission, without compromising power and performance. As a result, in order to increase the efficiency and reduce the emission, automotive engines have undergone a drastic evolution. Turbocharging became one of the major developments in the field of automotive engines in the recent past.

As compared to a naturally aspirated engine, which draws the air in at atmospheric pressure, a turbocharger forces the air into the engine’s cylinder. As a result, a smaller turbocharged engine can produce the same power as a naturally aspirated engine with larger displacement, hence resulting in the considerable reduction in fuel consumption, CO₂ emission and noise pollution, without hampering the power produced by the engine. Turbochargers utilize the wasted energy from the exhaust to drive the system, which in turn helps the piston to pull in more air into the combustion chamber, resulting in better combustion of fuel. Turbocharging provides the chance to achieve the performance of a naturally aspirated engine while producing lower emissions, through engine downsizing.
1.1. CLASSIFICATION

[1] Supercharging is the primary method used in Automotive Industry for power boosting by pressurizing the intake air prior to its entering the engine cylinder. There are several ways through which it can be achieved:

**Mechanical Supercharging:** This method of forced induction involves providing compressed air to the engine, through a compressor, powered mechanically through belt, gear, or chain connected to the engine crankshaft.

**Turbocharging:** It involves the usage of a turbine and a compressor both mounted on the same shaft. The turbine is powered by the wasted exhaust gas and in turn powers the compressor, which boosts the pressure of the inlet air.

**Pressure wave Supercharger (Wave Rotor):** This method utilizes the pressure waves produced by the intake air and exhaust gas pulses, to compress the inlet air.

Among all methods, turbocharging can be considered the most attractive one, as unlike supercharging (consumes one third of engine power), and pressure wave-charging (expensive and complex method powered by the crankshaft), it involves usage of the waste exhaust gas to boost the engine power without deteriorating the fuel efficiency. Additionally, turbocharged engine have low heat losses. While in a naturally aspirated engine higher combustion temperature will cause a decrease in volumetric efficiency, in a turbocharged engine reduction in volumetric efficiency is not as significant as higher exhaust temperature would mean increased work available for the turbine. Compared to a mechanically driven Supercharger, turbochargers are generally more efficient, but they tend to be less responsive.

1.2. HISTORY

[2] The idea of forced induction is as old as the internal combustion engine itself. In the late 19th century Gottlieb Daimler and Rudolf Diesel were also researching this concept. The technique of using a gear driven pump to force air into the engine was patented by Daimler in 1885. But the first practical turbocharger was developed and patented by Alfred Buchi in 1925. His setup used exhaust gas to drive the compressor, to increase the fuel-air mixture flow, and to enhance the performance. Initially, turbochargers were used primarily on large engines, e.g. marine engine, but later they found application in aircraft engines, as they minimized the issue of engine performance deterioration due to the declining air density in higher altitudes. Some supercharged aircraft engines found a limited application by the French Airforce during First World War. The first application of Turbochargers in automotive engines occurred in truck engines in 1938, when Swiss Machine Works Sourer developed the first turbocharged truck engine. The beginning of the Second World War saw a high growth in turbocharger technology, with the turbocharged engine (manufactured General Electronics finding application in American aviation industry. It was during this time, that the first company dedicated to turbo
technology, The Garrett Corporation, was established. Chevrolet Corvair Monza and Oldsmobile Jet fire launched in US market in 1962/63 were the first turbo-powered passenger cars. But it wasn’t until the oil crisis of 1973 that the automotive turbocharger technology evolved enough to make inroads into the commercial diesel segment. The first turbo-diesel passenger car, Mercedes Benz 300SD made its debut in 1978 and it became an instant hit due to low emissions, good efficiency and drivability similar to petrol cars. With stringent emission norms, turbocharging became more and more popular in the transport industry, with motorsport popularizing turbocharging among the public.

1.3. CURRENT TREND

Turbocharger technology has gradually evolved from being a power boosting option to an economical, low emission alternative. Because of the benefits of turbocharging, it has grown from being a niche powertrain system to a central powertrain strategy. Small cars, where restrictive engine bays make it unfeasible to install large engines, have benefited most from this technology. With the advancement in technology, the problems associated with turbochargers, such as reliability and turbo-lag have been minimized. Still, with all the progress in technology, the challenges faced by turbochargers are also growing, as the demand for more powerful and efficient engines with minimum losses has been increasing steadily. Few of these challenges are the ability to provide high pressure turbocharging at low engine speeds, high transient response and high efficiency at high pressure ratio. Different turbocharger design strategies like Variable Geometry turbocharger (VGT), Electric assisted turbocharger (E-Charger), Two-stage turbocharger and Waste gated turbocharger have been developed to accomplish the current challenges faced by turbochargers. Some of the newer technologies like ball-bearing center sections, bleeding-edge compressor and new turbine wheel designs, and surge resistant compressor covers are also being explored to allow big-power turbos to improve the boost response. There is still a rich unrevealed potential in the turbocharger technology and with the hybrid/electric vehicles still facing credibility issues such as high pricing and feasibility, it is easier to assume that clean, reliable and economical turbocharged engines bear a high chance of increasing their strong hold in the passenger car segment, at least for the next couple of decades.

*Fig. 1 Cut section of a Turbocharger (source: commons.wikipedia.org)*
1.4. THESIS BACKGROUND

This thesis has been conducted with the collaboration of Honeywell Transportation System (TS) and Czech Technical University. Honeywell has been at the forefront of turbocharger development for more than 60 years. And with the automotive industry switching towards more efficient and environment friendly technologies, TS has been striving to become a major contributor of power train strategies that will integrate electric and conventional turbochargers into air management system. The vision for TS has always been to invest in electric boosting and fuel cell technology to boost engine performance and push the performance envelope of powertrain technology forward. Honeywell has been associated with other OEMs to develop opportunities for electric turbos in the market.

To achieve these goals, the thesis involves developing a mean-value based model of a Waste-gated Turbocharger and then development of a simple control based strategy for combination of Turbocharger with an e-Charger.

1.4.1. OBJECTIVE OF THESIS

1) Reviewing of existing turbocharger technologies and modelling approaches for turbocharger and e-Charger known in literature, with focus on control oriented models.
2) Development of a control oriented model, using an existing turbocharger technology.
3) Conducting sensitivity study of key parameters related to changes of turbocharger efficiency and flow rate.
4) Proposing a simple control based strategy for combination of turbocharger with e-Charger.
5) Study effect of turbocharger degradation to the overall performance of control system.

1.4.2. THESIS DELIVERABLES

1) Details of existing turbocharger technologies. Review of different turbocharger modelling approaches, (e.g. Mean value modelling by Jensen and Kristensen) which have been incorporated into the thesis.
3) Combination of the turbocharger with e-charger and controlling of the e-charger using LQR (Linear Quadratic Regulator) controller. Sensitivity study of the model through degradation of efficiency and flow-rate.
2. TURBOCHARGING STRATEGIES

Different strategies have been employed in turbochargers to improve the transient response and performance. In addition, certain modifications have been carried out in the turbocharger setup, to make the system suitable for different applications, and to allow new technologies to easily blend into the system. Some of the strategies discussed in this section, have also been used in combination to improve the overall performance of the turbocharger.

2.1. TURBOCHARGING DESIGN VARIATIONS

2.1.1. SINGLE STAGE TURBOCHARGER

Single stage turbocharging uses the engine exhaust to power the compressor, which in turn, compresses the inlet air and provides power boost through forced induction. [4] Advantages associated with such a system are: considerable increase in power, engine downsizing, improvement in fuel economy and higher efficiency. Associated disadvantages are: the system includes turbo lag (as it requires a certain time for the turbine to spool up and provide the boost), inability to operate across wide RPM range, and large instantaneous power surge (which is achieved at boost threshold). All this can significantly affect the stability of the car.

![Fig. 2 Single Stage Turbocharger](image)

2.1.2. TWIN CHARGER

Twin charger is a compound force induction method, which makes use of a combination of mechanical supercharger and exhaust-driven turbocharger, thus cancelling out each other’s disadvantages. As discussed earlier, a normal single stage turbocharger suffers from the poor response and low rpm performance due to the time lag between the application of throttle and pressurization of manifold, while using a mechanically driven supercharger would mean an increase in fuel economy, as a portion of power produced by the engine is consumed by the compressor of the supercharger.

This system uses a combination of bypass valves in the intake and exhaust systems, and an electromagnetic clutch. The main goal of twin chargers is to achieve an acceptable response, smooth power delivery and adequate power gain.
Twin charger works in three phases depending upon the RPM.

- At low rpm pressure boost is provided by the mechanical supercharger.
- At mid rpm range boost is provided by both the systems (supercharger & turbocharger)
- At high rpm, the supercharger is bypassed or disconnected, and the boost is completely provided by the turbocharger. As a result this combination can produce linear power gains in relation to rpm of the engine.

There are various configurations of twin charging system: Parallel and Series Arrangement.

2.1.3. TWO STAGE TURBOCHARGER

It features a two-stage turbo architecture in either serial or parallel combination, where the two turbos work sequentially. Primary turbo (usually a variable geometry turbocharger), provides the boost at low engine speed (working as a single stage turbo). In this mode, the turbine valve which controls the flow of exhaust from the primary to the secondary turbo, is closed. When the engine reaches a defined load, the valve starts to open, allowing the flow of exhaust to activate the secondary turbo, which works at higher speed to provide optimum peak power. As both turbos operate sequentially, with the engine speed increasing, the turbine valve channels optimum gas flow to each of the turbos ensuring right level of boost is provided at just the right time. At the same time, the compression recirculation valve regulates pressurized air entering the combustion chamber.

Such an architecture facilitates a significant increase in power over a normal turbocharger, with the development of additional torque, hence ensuring remarkable performance even at very low rpm and significant emission reduction.
2.1.4. TURBOCOMPOUNDING

Turbo compounding involves the usage of an exhaust-driven turbine to provide additional power to the crankshaft by means of gearing or hydrodynamic coupling.

Such a setup is like a normal single-stage turbocharger, the only variation being the presence of an additional turbo, which is driven by the exhaust from the primary turbo and is connected to the engine crankshaft via a gear train, chain, belt or pulley. Another possible approach of turbo compounding is that rather being connected to the crankshaft, the second turbo can be alternatively used to drive a generator. With this approach the need for engine-driven alternators can be eliminated.

Fig. 5 Scheme of turbocompounding

Additionally, turbo compounding increases the engine output (power boost provided by compressed inlet gas + additional power provided to the crankshaft) without increasing its fuel consumption (reduction in specific fuel consumption).

Turbo compounding has advantages like improved fuel economy, lower cost, less complexity, higher exhaust gas recirculation capability and improved transient response. Additionally, variability in geometry and the rotational speed of turbines increases the efficiency of turbo compounding in different operating conditions. Disadvantage associated with this technology is the additional exhaust back-pressure, which leads to higher pumping loss in the engines.

2.1.5. ELECTRIC TURBOCHARGER

An electric turbocharger is a small electric supercharger powered by an electric motor from a high voltage (48V or higher) source based on electrical energy source (capacitor or battery), attached to a conventional exhaust-based turbine, to enhance its performance. Advantages associated with this configuration include: Improved packaging and engine downsizing, reduced length of intake ducts and increased size of the compressor wheel, tighter predictive control of in-cylinder combustion.

Fig. 6 Electronic Turbocharger [source: Wikipedia]
This system works in three phases:

a) **Acceleration phase** – At low rpm, compressor is directly powered by a power source. Resulting in compressor motor reaching full speed quickly, thus minimizing turbo-lag.

b) **Charging phase** – At high rpm, the turbine produces excess energy which can be utilized to charge the power source and power other auxiliary electric systems within the vehicle.

c) **Steady state** – In the steady state, the turbo transfers that amount of energy as required by the compressor.

### 2.1.6. WASTEGATED TURBOCHARGER

Primary motive of this configuration is to prevent the turbocharger from producing excess boost; thus, safeguarding both engine and the turbocharger. This strategy is usually used in combination with other turbocharger strategies like twin turbo, electric turbocharger, etc. This system involves regulating the speed of the turbine by diverting the exhaust gas which in turn regulates the speed of the compressor. Once excess boost pressure is detected, the excess pressure valve in the intake manifold duct gets activated, resulting in escaping of this pressure through the valve. This escaped pressure actuates the swing control valve, which opens the waste gate; thus, diverting the exhaust gas from the turbocharger.

![Waste-gated turbocharger](image)

As the boost pressure reaches its normal value, this circuit is broken (both the excess pressure valve and the waste gate close) and again all the exhaust reaches the turbine. Different Schemes of waste gated turbocharger are: - Internal waste gate and External waste-gate.
2.1.7. TWIN SCROLL TURBOCHARGER

Cylinder of an ICE fire in a sequence, so the exhaust leaves the combustion chamber in irregular pulses. In a conventional turbocharger, all the cylinders have a common exhaust outlet, due to which these irregular pulses collide and interfere with each other, resulting in the reduction of the strength of flow. In case of twin scroll, the exhaust manifold is divided into two scrolls or channels, each of which gathers the exhaust from pairs of cylinders in alternating sequence. Different size of channels can be used to change air ratio of the turbocharger. Larger channel will have higher capacity, while a smaller channel will have higher velocity.

Advantages associated with the twin scroll turbocharger are the increase in low-end torque, improved boost response, increased power throughout the power band, greater turbine efficiency, reduction in engine pumping losses, improved fuel economy, lower exhaust temperature and the decrease in inlet gas dilution during valve overlap.

Fig. 8 Twin scroll turbocharger [source: grabcad.com]

2.1.8. VARIABLE GEOMETRY TURBOCHARGER (VGT)

Conventional turbocharger systems have limitations, e.g. large turbo cannot work properly at low rpm, while a small turbo has good response at low rpm but run out of steam at higher rpm. Most of the modern turbine architecture uses a combination of turbochargers or external power source to deal with these limitations. But VGT uses variable vanes to deal with these drawbacks. These variable vanes direct exhaust flow into the turbine blades and are adjusted via an actuator. The vane angles vary throughout the engine RPM range for turbine behavior optimization.

Fig. 9 Cut-section of a VGT

At low rpm, the variable vanes are in a closed angle, making a narrow passage for the exhaust gas, causing the flow of the gas to accelerate towards the turbine blades (higher velocity). At high rpm the variable vanes are fully open causing high exhaust flow directed towards the turbine blades (high capacity). Additionally, Vane angle also governs the angle at which the exhaust gas hits the turbine blades. The variable geometry turbine area can also be used to increase the turbine back pressure, in turn increasing engine braking.
2.2. TURBOCHARGING MODELLING STRATEGIES

Discussed in this section are reviews of certain research papers elaborating various strategies of turbocharger modelling. These papers not only discuss the strategies, but also highlight the limitations of the modelling strategy and various methods through which these limitations can be minimized. These papers influenced the turbocharger modeling strategy used in this thesis.

A) MEAN VALUE MODELLING OF A SMALL TURBOCHARGED DIESEL ENGINE – J P JENSEN, A F KRISTENSEN - TECHNICAL UNIVERSITY OF DENMARK [1991]

The primary aim of the paper is to develop a simple and accurate model representing the transient operation of a swirl chambered turbocharged diesel engine (specifically light duty vehicles), using mean value modelling approach. Mean value modeling involves physically based models, hence deriving the model from basic physical and thermodynamic principles. Resulting in a simpler and more general relationships throughout the model, thus making it more flexible for different engine sizes and types. Time scale for mean value models is in the order of 3-5 revolutions, making it fast enough to provide accurate predictions of dynamical operations of the engine. But this modelling approach takes a bit of liberty in terms of sub-cycle events like cylinder pressure variation, throughout the engine cycle.

Model comprises of major sub-systems like: Engine, Compressor and Turbine, resulting in various coupled, non-linear algebraic and differential equations. Initial model calibration is done using manufacturer provided data and steady state experimental data. Further model improvisation is carried out by direct comparison with dynamical engine measurements. Engine model was designed on steady state engine measurements, and sub-models were developed independently, so they can have proper dependence on relevant engine variables.

Compressor – It is a simplified version of Winkler’s Engine Turbomachinery model and is expressed in terms of dimensionless head parameter $\Psi$ (where $\Psi$ is function of Pressure ratio, Inlet temperature $T_{in}$ and Blade speed ratio $U_c$) and normalized flow rate $\Phi$(where $\Phi$ is a function of compressor mass flow rate and Blade speed ratio). Head parameter and normalized flow are related to each other by the following equation.

$$\Psi = \frac{K_1 + K_2 M + K_3 \Phi + K_4 M \Phi}{K_5 + K_6 M - \Phi}$$  \hspace{1cm} (2.2.1) \hspace{1cm} where $M$ is inlet Mach number, $M=f(U_c,T_{in})$

Eq. (2.2.1) was fitted to manufacturer supplied data using no-linear statistical program. The compressor efficiency was expressed as a function of normalized flow and inlet Mach number $M$. Coefficients $A_i$, which is a second order expression of efficiency as a function of normalized flow rate, were found to have following dependence on $M$.

$$A_i = \frac{K_{1i} + K_{2i} M}{K_3 - \Phi}$$  \hspace{1cm} (2.2.2)
The accuracy of compressor model declines appreciably for speeds above 160000 rpm, but still this limitation doesn’t adversely affect the model, because for small engine waste-gate limits the compressor speed to somewhere around 130000 rpm.

Turbine - A relationship is established between turbine speed, pressure ratio, inlet temperature, efficiency and flow area. In case of radial turbine efficiency was established as function of the isentropic expansion velocity and turbine tip speed. The efficiency data was fitted as a function of this ratio using a third order polynomial. End point for high and low values of tip speed were included in the fitting method, to ensure correctness of efficiency over a wider operating range. The coefficients of polynomial $b_i$ were of the following form.

$$b_i = \frac{k_{1,i} + k_{2,i} \times N_t}{k_3 - N_t} \quad i = 1, 2, 3 \quad (2.2.3)$$

Equivalent turbine area has been used to describe the flow characteristic of a turbine. This area gives the correct turbine flow, when the turbine inlet temperature and pressure ratio are used in isentropic flow equation. The empirical formula for turbine flow area as a function of pressure ratio and turbocharger speed is given as:

$$b_i = k_{t_1} \frac{P_5}{P_6} + k_{t_2} \quad \text{where} \quad k_{t_i} = k_{1,i} \frac{N_t}{\sqrt{T_5}} + k_{2,i} \quad \text{and} \quad i = 1, 2 \quad (2.2.4)$$

Engine – Engine modelling technique is based on mean value model for a normally aspirated SI engine. It is composed of a series of sub-systems for friction, thermal efficiency, volumetric efficiency, Manifolds (both intake and exhaust) and Engine Speed. Since it is a turbocharged engine it also requires a model for temperature increase over the engine, for determining turbine inlet temperature. The engine speed model is based on the principle of conservation of energy applied to the crankshaft. Input energy is determined by the rate of fuel energy, while by using an accurate model for indicated thermal efficiency, indicated work can be calculated.

Turbocharger Speed – The difference between power produced by Turbine and power consumed by Compressor, determines the acceleration or deceleration of the turbocharger. On being applied to the turbocharger rotor (in connection with Moment of inertia), the following relationship is obtained.

$$\frac{dN_{TC}}{dt} = \frac{K_{t} \times \eta_{TC} (P_t - P_c)}{I_{TC} \times N_{TC}} \quad (2.2.5)$$

Testing of Model – The model prediction was evaluated for a series of transient runs. In each run, different test conditions were introduced. In the first condition engine speed was kept constant and the throttle was opened between two fixed positions as rapidly as possible (thus giving a good indication of turbocharger dynamics alone). In second condition torque was kept nearly constant, and it was separated in various phases: first phase being steady state operation, followed by second phase of ramp in fuel flow from part load to nearly full load and then steady state phase, followed by fast ramp decrease in fuel flow and finally end phase of steady state operation. The result of each these test conditions (predicted values) were then compared to the actual measured values, to find out the variation between these two values.

Conclusion of the research paper – The mean value model was developed to simulate both steady state and transient performance of a small turbocharged diesel engine. The prediction accuracy of the model for all significant engine variables was around ± 10% for both transient and steady state operation.
Dynamic simulation models of turbocharged engines have been increasingly used for designing and initial testing of engine control strategies. Standard practice of using turbocharger performance data in the form of look up tables, had certain limitations for control oriented engine models, because standard table interpolation routines are not continuously differentiable, extrapolation is not reliable and table representation is not compact. Primary aim of this paper is to give an overview of curve fitting methods for compressor and turbine characteristics, and overcoming limitations of standard tabular form of turbocharger performance map. Being crucial part of overall engine model, emphasis has also been laid upon compressor flow rate modelling.

When it comes to Turbocharger mapping, the idea should be extension of available data, rather than prediction/extrapolation of behavior outside the given range. Though, in case of a good model with underlying physics, it would be sufficient to obtain few data points, to characterize a large operating region and reduce experiment development time.

**Compressor** - On neglecting the heat losses, the power required to drive compressor \( P \) can be related to mass flow \( m_c \) and total enthalpy change (First law of thermodynamics) and on further calculation of isentropic efficiency, following relation can be obtained.

\[
P = \frac{m_c c_p T_{01}}{\eta_{c,ls}} \left( \frac{P_{02}}{P_{01}} \right)^{\gamma - 1} - 1 \quad (2.2.6)
\]

By considering enthalpy change across compressor, we can obtain a new form of eq.(2.2.9)

\[
\left( \frac{P_{02}}{P_{01}} \right) = \left( 1 + \frac{\Delta h}{c_p T_{01}} \right)^{\gamma - 1} \quad (2.2.7)
\]

\( \Delta h \) can be calculated from Euler’s formula for turbomachinery (considering no losses).

\[
\Delta h = U_2 C_{\theta 2} - U_1 C_{\theta 1} \quad (2.2.8) \quad \Delta h_{ideal} = U_2 C_{\theta 2} \quad (2.2.9) \quad \text{for no inlet pre-whirl}
\]

Manufacturers, conventionally specify the performance maps in terms of scaled mass flow rate parameter \( \varnothing \) and compressor rotational speed parameter \( \tilde{N}_{TC} \). These scaled parameters help in eliminating dependence of performance map on inlet condition.

\[
\varnothing = m_c \sqrt{T_{in}/P_{in}} \quad (2.2.10) \quad \tilde{N}_{TC} = N_{tc}/\sqrt{T_{in}} \quad (2.2.11)
\]

**Curve Fitting Compressor Mass Flow** – For mathematical representation flow characteristics of compressor can be modelled in following two ways.

\[
\text{Model 1: } m_c = f_1 \left( \frac{P_{out}}{P_{in}}, N_{tc} \right) \quad (2.2.12) \quad \text{Model 2: } \frac{P_{out}}{P_{in}} = f_1 (m_c, N_{tc}) \quad (2.2.13)
\]
Model 1 can be easily incorporated into mean value engine model, with ambient pressure ($p_{amb}$) as input & intake manifold pressure ($p_i$) as state variable.

$$p_{int} = \frac{RT_i}{\sqrt{V_i}} (m_c - m_e) \quad (2.2.14) \quad \text{where} \quad m_e = \text{engine pumping rate}$$

To incorporate Model 2, additional state variables need to be introduced, depending upon momentum equation for air mass $m_c$ in tube connecting compressor outlet with intake manifold. Momentum equation can be expressed as: ($A$ & $l$ are area & length of the connecting tube)

$$A(p_i - p_c) = m\dot{v} = \frac{m m_c}{\rho A} = \dot{m} c_l \quad (2.2.15) \quad \dot{m} c = \frac{A}{l} (p_i - p_c) \quad (2.2.16)$$

And depending upon intake assembly geometry, inclusion of state variable can increase model stiffness considerably. Model stiffness refers to the ratio of smallest to largest eigenvalue of the linearization around a given operating point.

**Different Curve fitting methods**

**Jensen & Kristensen Method** - already discussed in previous section [2.2.A]

**Mueller method** - This approach developed by Martin Mueller, derives compressor model from first principles including underlying physical principles and compressor assembly geometry, in order predict compressor behavior. According to Mueller's hypothesis, head parameter ($\Psi$) has been proposed as a quadratic function of normalized flow rate ($\Phi$).

$$\Psi = A \Phi^2 + B \Phi + C \quad (2.2.17) \quad \text{where} \ A, B, \text{ and } C \text{ are speed dependent}$$

**Zero Slope Line Method** – This method is similar to Model 1, which describes compressor flow parameter ($\Phi$) as a function of pressure ratio ($\pi$) and speed parameter ($\tilde{N}_{tc}$). Curve connecting max. mass flows on each speed line (zero slope line) is characterized by a quadratic in $\tilde{N}_{tc}$.

$$\Phi_{top} = k_1 N_{tc} + k_2 N_{tc}^2 \quad (2.2.18) \quad \pi_{p,top} = k_3 + k_4 \Phi_{top}^2 \quad (2.2.19)$$

**Neural networks** - They have become increasingly popular for wide range of applications, like curve fitting, system identification, etc. And though generally, neural networks are universal function approximators, even then a lot of trial and error is required for finding correct structures and coefficients. Compressor flow maps are modelled in form of Model 2, and comprises of one layer and five neurons, which can be represented in following form.

$$y = b_2 + \sum_{i=1}^{n_i} \left[ m_2i f \left( b_{1i} + \sum_{j=1}^{n_u} m_{1i,j} u_j \right) \right] \quad (2.2.20) \quad f(x) = \frac{1}{1 + e^{-x}} \quad (2.2.21)$$

$B$ – coefficient vector $W$ – coefficient matrices $f$ – neuron transfer function

**Curve fitting of Compressor efficiency** – It can be achieved by two ways: One method would be to employ Jensen Kristensen method, where efficiency is a function of normalized compressor flow $\Phi$ and coefficient $a_i$ (which can be determined through least square fit on experimental data). Second method would require neural network, which can be augmented
with a second output (additional nodes if necessary) and can be retrained to provide an approximation for isentropic efficiency and mass flow rate.

**Turbocharger** - In turbine modelling, power is obtained from First law of thermodynamics, and to obtain its final form, isentropic efficiency needs to be defined.

\[ P = m_t c_p \eta_{t, is} T_{01} \left( 1 - \left( \frac{P_{02}}{P_{01}} \right)^{\gamma-1} \right) \]  \hspace{1cm} (2.2.22)

Turbocharger model is used during simulations to calculate power, with inlet and outlet pressure values and speed as input. Speed and inlet pressure values are state variables determined by differential equation based compressor-turbine power balance and ideal gas law. Outlet pressure is a function of flow restriction of the exhaust system, downstream of turbine. Hence it is ideal to express turbine characteristics in the same form as compressor Model 1. Scaling of mass flow parameter (\( \Phi \)) and speed parameter (\( N_{TC} \)) is carried out. In addition, the difference between static and stagnation pressures and temperatures is neglected. These corrections eliminate the dependence of performance maps on inlet conditions (\( T_{in} \) and \( P_{in} \)) and make them function of only speed parameter, pressure ratio across the turbine (in case of VGT \( \theta_{vgt} \) setting). U/c is blade speed ratio.

\[ \Phi = f_1 \left( \tilde{N}_{TC}, \frac{P_{out}}{P_{in}}, \theta_{vgt} \right) \hspace{1cm} (2.2.23) \]
\[ \eta_{t, is} = f_1 \left( \frac{U}{c}, \frac{P_{out}}{P_{in}}, \theta_{vgt} \right) \hspace{1cm} (2.2.24) \]

**Curve fitting turbine mass flow** – Turbine mass flow rate can be modelled as an adiabatic nozzle flow, where effective flow area is a function of pressure ratio and turbine speed parameter. Standard orifice flow eqs. are derived using isentropic expansion. But in case of turbine flow, expansion is not isentropic & isentropic efficiency is known. So, we could use generalized form.

\[ \Phi = \begin{cases} A_t \sqrt{\frac{2\gamma}{\gamma - 1} \left( \left( \frac{P_{out}}{P_{in}} \right)^{2/\gamma} - \left( \frac{P_{out}}{P_{in}} \right)^{\gamma+1/\gamma} \right) } & \text{for } \frac{P_{out}}{P_{in}} > P_{crit} \\ A_t \sqrt{\frac{2\gamma}{\gamma - 1} \left( P_{crit}^{2/\gamma} - P_{crit}^{\gamma+1/\gamma} \right) } & \text{for } \frac{P_{out}}{P_{in}} > P_{crit} \end{cases} \hspace{1cm} (2.2.25) \]

Effective turbine area \( A_t \) is modeled as function of \( \tilde{N}_{TC} \) and \( \pi_t \). For fixed geometry TC, a simple fit for effective turbine area was proposed by Jensen & Kristensen (which is an approximation)

\[ A_t = k_{1t} / \left( \frac{P_{out}}{P_{in}} \right) + k_{2t} \hspace{1cm} (2.2.26) \]

\( k_a \) is function of \( \tilde{N}_{TC} \): \( k_{bi} = k_{b1} \tilde{N}_{TC} + k_{b2i} \) for \( i=1,2 \)

**Curve fitting of turbine efficiency** – For fixed speed, efficiency is typically shaped as inverted parabola and modelled by a quadratic or cubic polynomial in U/c, with coefficients dependent on \( \tilde{N}_{TC} \). Correction factor at low expansion ratio is used to account for observed accelerated decrease in efficiency at those lower expansion ratios.

\[ \eta_t = b_0 + b_1 \tilde{N}_{TC} + \left( b_2 + b_3 \tilde{N}_{TC} \right) \frac{U}{c} + \left( b_4 + b_5 \tilde{N}_{TC} \right) \left( \frac{U}{c} \right)^2 \hspace{1cm} (2.2.27) \]
Recent air system architecture have become very complex due to the interaction between multiple subsystems, to make engines cleaner and efficient. The primary aim of this paper is to design a control oriented model of a turbocharger adapted to an engine architecture with two EGR loops. The primary inspiration for the control structure has been taken from the work of Grondin [2009] pertaining to strategies for control of each EGR loop and estimation of intake manifold compositions.

This turbo architecture involves a four-cylinder diesel engine, fitted with a variable geometry turbocharger, connected to two EGR loops, namely: HP and LP EGR loops

- High Pressure EGR loop - Exhaust gas derived directly from exhaust manifold, is passed through intercooler and diverted back to intake manifold, where it combines with a mixture of compressed intake air and exhaust gas from LP EGR.
- Low Pressure EGR loop - This loop takes exhaust downstream of the particulate filter and transports upstream of the compressor, where it mixes with the fresh intake air.

Main idea of the model development is to provide a basis for control strategy design, while maintaining right level of complexity. As a result, two criteria have been considered. First to minimize the number of states in the controller, the model deals with only main dynamics dealing with evolution of the system. Secondly Turbocharger evolution depends upon the condition at its boundaries - Pressure and temperature, upstream and downstream of the compressor and turbine. These are very important for the architecture with two EGR loops.

Turbocharged Engine Model – Equations governing behavior of turbocharger have been taken from the publication of Mooral and Kolmanovsky [1999], Sorenson [2005] and Eriksson [2006]. And the goal is to further simplify the model and control strategy. Turbocharger model comprises of a Turbine powered by exhaust gases, connected via a shaft to a compressor, which compresses inlet air. Rotational speed \( N_t \) of turbocharger can be derived from balance between Turbine and Compressor power.

\[
\frac{d(0.5 \cdot J_T \cdot N_t^2)}{dt} = P_T - P_c \tag{2.2.28}
\]

Compressor – First law of thermodynamics has been applied, to derive compressor power equation. So according to it, compressor power \( P_c \) is related to mass flow through compressor \( m_c \) and total change of enthalpy (on neglecting heat losses). While compressor efficiency \( \eta_c \) is defined as ratio between isentropic and actual compressor power.

\[
P_c = m_c \cdot C_p \cdot T_{uc} \left( \frac{\pi_c^{\gamma - 1}}{\gamma - 1} \right) \tag{2.2.29}
\]
Compressor speed, flow, pressure ratio and efficiency are interlinked with each other and pressure ratio and efficiency have been mapped against mass flow and speed. These maps are extrapolated from data measured during characterization test. Several extrapolation methods have been proposed, for e.g. Jensen Kristensen method [1991]. All the variables like mass flow and speed corrected in order to consider the variation of upstream compressor condition.

Turbine – Like the compressor model, turbine power is also related to turbine mass flow and total enthalpy change. Thus, Turbine power (Pt) eq. takes the following form –

\[ P_t = m_t \cdot C_p \cdot T_{ut} \cdot \eta_t \left( 1 - \frac{\pi_t}{1 - \gamma/\gamma} \right) \] (2.2.30)

In case of turbine, Mass flow and Speed have also been corrected. Corrected mass flow and isentropic efficiency (\( \eta_t \)) are mapped versus pressure ratio, corrected speed (\( N_{corr} \)) and VGT actuator position (\( u_{vgt} \)). Different methods can be used to obtain the maps, for e.g. Mooral and Kolmanovsky [1999]. So the mass flow eq. can be arranged in the following form.

\[ m_t = \Phi_{Dt} \left( \pi_t, N_{corr}, u_{vgt} \right) \pi_t \frac{P_{Dt}}{\sqrt{V_{ut}}} \] (2.2.31)

Engine Model- Conventionally aspirated flow \( m_{asp} \) can be computed as:

\[ m_{asp} = \eta_v \pi_c \Psi \] (2.2.32) \quad where \quad \Psi = \frac{\nu e^{\Psi + N_c}}{120R \cdot \Psi_{int}}

In engine modelling, exhaust and intake manifolds are considered as fixed volumes, having homogenous thermodynamic states (pressure, temperature and composition). Mass balance in the intake and exhaust manifold can be explained as:

\[
\begin{align*}
\frac{dP_{ut}}{dt} &= \frac{RT_{ut}}{V_{ut}} \left( m_{asp} + m_f - m_t - m_{egr, hp} \right) \\
\frac{dP_{uc}}{dt} &= \frac{RT_{uc}}{V_{uc}} \left( m_{in} + m_{egr, lp} - m_c \right)
\end{align*}
\] (2.2.34)

Model Reduction – There are two kinds of assumption: First related to dynamics and second related steady state dependencies. Aim is to keep only relevant system dynamics and parameters, which can be measured or estimated.

**Dynamic Simplification: Model Simplification by singular perturbation** – Fifth order non linear system describes the system dynamics. However, turbocharger speed is much smaller than pressure dynamics. Typical values of \( V_{ut} / RT_{ut} \approx 5e-9\) , \( V_{dc} / RT_{dc} \approx 5e-8\) and \( J_t = 3e-5\). Which suggests to simplify these dynamics with a singular perturbation method Khalil[1992].

\[
\frac{d(J_t \cdot N_t^2)}{2 \cdot dt} = \Phi_{Dt} \left( \pi_t, N_{corr}, u_{vgt} \right) C_p \frac{P_{Dt}}{\sqrt{V_{ut}}} T_{ut} \eta_t (\pi_t) - (\eta_v \pi_c - m_{egr, hp}) \frac{T_{uc}}{\pi_c} \Phi_c (\pi_c) \\
\eta_v \pi_c \Psi = \Phi_{Dt} \left( \pi_t, N_{corr}, u_{vgt} \right) \frac{P_{Dt}}{\sqrt{V_{ut}}} - m_f + m_{egr, hp}
\] (2.2.35)
These are simplified first order non-linear dynamics with algebraic equation, steady state solution of intake and exhaust dynamics.

**Steady State Assumptions: High Pressure Flow Simplification** - Contrary to the turbocharger HP EGR loop has direct control of the flow (through EGR valves). Hence simplification is required to substitute HP EGR flow by reference value link to the intake pressure, which is done by introduction of variable $\delta_{HP}$ which can help in characterization of EGR circuit.

$$m_{egr, \text{hp}} = \frac{X_{\text{int} \delta \text{HP}}}{X_{\text{exh}}} \eta_p \pi_c \Psi$$  \hspace{1cm} (2.2.36)  \hspace{1cm} $X_{\text{int}}/X_{\text{exh}}$ – intake to burned gas ratio

**Steady State Assumptions: Turbine Flow Simplification** – We can simply assume turbine as a restriction to the exhaust flow, but equation for compressible flow across an orifice cannot applied to it. So a modification in the equation, like neglecting the effect of turbine speed, needs to be carried out to match with the experimental results.

$$m_t = \frac{P_{\text{turb}}}{\sqrt{T_{\text{ut}}}} \Psi_{\text{turb}}(\pi_t) \Psi(u_{\text{vgt}})$$  \hspace{1cm} (2.2.37)  \hspace{1cm} $\Psi(u_{\text{vgt}})$ is the effective area

**Correlation between TC speed and intake pressure** – For engine operating conditions TC speed and intake pressure are quite correlated. So by combination of equations, we get a relation, which showcases the dependency of $m_{c, \text{corr}}$ on $\pi_c$, engine speed and operating conditions. In addition, square of TC speed has a linear relationship w.r.t Compressor Pressure ratio.

$$\pi_c \Psi = \phi_{\pi_c} (\eta_p (\pi_c P_{\text{dt}}, \pi_c \Psi_{\text{vgt}}, N_t / \sqrt{T_{\text{uc}}})$$  \hspace{1cm} (2.2.38)  \hspace{1cm} $N_t^2 = a \pi_c + b$  \hspace{1cm} (2.2.39)

**Reference System** – the system can be further simplified by a linear co-relation between $\pi_c$ and turbocharger kinetic energy. $\pi_c$ can be considered as state variable and $N_{tc}$ can be neglected.

$$\pi_c = \alpha_1 (\beta \pi_c + m_f) \Psi_t (\pi_t) - \alpha_2 \beta \Psi_t (\pi_c)$$  \hspace{1cm} (2.2.40)

$$\pi_c = \alpha_3 (\phi_{\text{turb}} (\pi_t) \Psi_{\text{vgt}} (u_{\text{vgt}}) - \alpha_4)$$

$$\beta = \left(1 - \frac{X_{\text{int} \delta \text{HP}}}{X_{\text{exh}}} \right) \eta_p \Psi \alpha_1 = \frac{C_p T_{\text{ut}} \eta_t}{J_{e+\alpha}^2} \text{and} \alpha_2 = \frac{C_p T_{\text{uc}}}{J_{e+\alpha} \eta_c}$$

$$\alpha_3 = \frac{P_{\text{dt}}}{\beta \sqrt{T_{\text{ut}}}} \text{ and } \alpha_4 = \frac{m_f \sqrt{T_{\text{ut}}}}{P_{\text{dt}}}$$

$\{\alpha_i\}_{i \in [1,4]}$ depend on engine operating conditions

Following eqs. represents balance between compressor and turbine mechanical (providing dynamics of the system) and mass conservation in exhaust manifold (neglecting the dynamics). Functions $\phi_{\text{turb}}, \Psi_{\text{vgt}}, \Psi_c$ and $\Psi_t$ are nonlinear, but invertible (property to be used when designing control law). Variable $\beta$ represents gas mass flow through the compressor and turbine (depends upon EGR loop choice) and coefficients $\alpha_1$ can be computed from sensors fitted on engine.
3. LIMITATION OF CONVENTIONAL TURBOCHARGER SYSTEM

Fundamentally turbocharger is a device that harnesses energy from waste exhaust gases, to increase the air flow inside the engine cylinder. It increases the air flow by pressurizing the incoming air. As a result, turbochargers can harness higher power output from a relatively small engine. This high power can be obtained at a relatively lower RPM and no extra energy is required to drive the turbocharger (as in the case of the supercharger). Though turbocharger technology has lots of advantages, there are also certain disadvantages associated with it that limit its usage.

[7] Low Speed Boost – For a turbocharged engine, attaining the maximum torque at low rpm can be challenging. For a typical fixed geometry turbocharger without a waste gate, turbine mass flow vs. the expansion ratio graph indicates that for typical low-speed operations of low-mass flows, the expansion ratio across the turbine is quite low, consequently resulting in lower power generated by the turbine (since expansion ratio strongly influences the power) and the lower boost pressure by the compressor, hence limiting the airflow into the engine. In diesel engines, this airflow limitation places a limit on engine torque due to smoke emission considerations while in stoichiometric gasoline engines, the torque would be limited from charge flow considerations.

The expansion ratio at low engine speed can be increased by selecting a lower flow capacity turbine and using a waste gate. Additionally, using a variable geometry turbine, gives one the flexibility in operation and higher expansion ratio at lower exhaust flows.

Fig. 10 Effect of various TC methods on low speed engine torque [source: dieselnet.com]
**Turbo lag** – is the time required to change power output, with respect to change in throttle position. It can be identified as the slowed throttled response when accelerating as compared to a naturally aspirated engine. This delay comes from the time required for the exhaust system and turbocharger to generate the necessary boost to spin the turbo and pump. This lag is maximum at low-engine rpm and low-load cruising situation. The main contributors to the turbo lag are Inertia, Friction and Compressor load.

The issue of turbo lag can be resolved by using a small turbo, which can produce a significant boost at low rpm, but such a small turbo will overspeed and malfunction in full throttle conditions. While a bigger turbo will work efficiently at peak power, it will not produce any boost until well into the engine’s powerband.

Turbocharger lag is a major problem in applications that require a rapid change in power output (such as automotive) and there is no single solution to eliminate it, though there are some strategies which can be implemented to reduce the lag.

- a) Lowering rotational inertia of the turbocharger by using lower radius parts and lighter materials.
- b) Changing the aspect ratio of turbine
- c) Reducing bearing frictional losses
- d) Using variable-nozzle turbochargers
- e) Increasing upper-deck air pressure (compressor discharge), with improved waste gate response
- f) Use of multiple turbochargers, either sequentially or in parallel

**Additional heat in the engine bay** – due to the recirculation of exhaust gas, the heat is transferred from the exhaust gas to the engine bay area (turbines operate in extreme heat, sometimes in excess for 1000 K for both diesel and gasoline engines) hence requiring better cooling upgrades and thermal shielding around the turbine and exhaust components to protect sensitive items. This additional heat in the engine owing to the turbocharger can also heat up the oil in the engine. More heat would mean the breakdown of the oil, hence, requiring more frequent oil change.

**Increased engine complexity and additional weight** – By installing the turbocharger system, additional components are added to both, intake and exhaust, thereby complicating the existing setup, occupying more space in the engine bay area, and increasing the overall weight. More components would mean higher chances of part failure; thus, the increased requirement for better maintenance. High level of accuracy is required while designing and manufacturing turbo components. Additionally, ECU needs to be tuned to take the full advantage of turbo-system. Incorrect tuning can lead to bad drive ability and engine failure.
4. TURBOCHARGER MODELLING

4.1. PURPOSE OF TURBOCHARGING

The maximum power an engine generates depends upon the amount of fuel which can be efficiently burned in the combustion chamber and that depends upon the quantity of air that can be introduced into each of the cylinder. If the intake air can be made more denser, than the maximum achievable power of an engine with fixed displacement, can be increased.

Following equations showcase the relationship between engine power, torque, mean effective pressure and inlet air density.

Power: \[ P = \eta_f \eta_v N V_d Q_{HV} \rho_{a,i} \left( \frac{F}{A} \right) \] \hspace{1cm} (4.1.1)

Torque: \[ T = \frac{\eta_f \eta_v N V_d Q_{HV} \rho_{a,i} \left( \frac{F}{A} \right)}{4\pi} \] \hspace{1cm} (4.1.2)

Mean effective pressure: \[ mep = \eta_f \eta_v V_d Q_{HV} \rho_{a,i} \left( \frac{F}{A} \right) \] \hspace{1cm} (4.1.3)

Where:
- \( \eta_f \) = fuel conversion efficiency
- \( V_d \) = displaced volume
- \( \eta_v \) = volumetric efficiency
- \( N \) = rpm
- \( Q_{HV} \) = Heating value of fuel
- \( F/A \) = Heating value of fuel
- \( \rho_{a,i} \) = inlet air density

Eq. (4.1) and (4.2) shows inlet air density directly affects Engine Power and Torque, and in turbocharging we utilize exhaust to pressurize the inlet air and thus increase the inlet density.

4.2. WORK-ENERGY RELATIONSHIP FOR TURBOMACHINERY

First and second law of thermodynamics has been utilized to obtain the basic equation for the work required to drive a compressor and the work produced by the compressor. We utilize the first law by applying the steady state energy equation on a control volume around a turbomachinery component.

\[ \dot{Q} - \dot{W} = \dot{m} \left[ \left( h + \frac{C^2}{2} + gz \right)_{out} - \left( h + \frac{C^2}{2} + gz \right)_{in} \right] \] \hspace{1cm} (4.2.1)

Where:
- \( \dot{Q} \) = heat transfer rate into the control volume
- \( \dot{W} \) = shaft work transfer rate out of control volume
- \( \dot{m} \) = mass flow
- \( h \) = specific enthalpy
- \( C^2/2 \) = specific kinetic energy
- \( gz \) = specific potential energy
Defining total enthalpy as \( h_o = h + \frac{c^2}{2} \) and neglecting specific potential energy and also treating the process as adiabatic, we can simplify the above equation.

\[
-W = \dot{m}(h_{o,\text{out}} - h_{o,\text{in}}) \tag{4.2.2}
\]

[10] Working of a compressor and turbine can be very clearly indicated by Temperature Entropy (\( T \)-\( s \)) plot, as compared to conventional Pressure volume (\( P \)-\( V \)) diagram of both Diesel and Otto Cycle.

An ideal compressor is isentropic, since it is both adiabatic and reversible. This property is depicted by a vertical line 1-2s in the \( T \)-\( s \) diagram. But the actual compressor is never isentropic (as they associated with an increase in entropy), hence they are depicted by a dashed line 1-2 in the diagram. Similarly an ideal turbine is also considered isentropic (depicted by line 3-4) but an actual is never isentropic (depicted by dashed line 3-4s). Pressure lines P1, P2, P3 and P4 represent Atmospheric pressure, Compressed Air pressure in the intake manifold, Exhaust gas pressure in the exhaust manifold and Exhaust gas pressure in the exhaust pipe, respectively.

Hence from the graph and equation (5) can be denoted as

Turbine work:
\[
W_t = h_3 - h_4 \tag{4.2.3}
\]

Compressor work:
\[
W_c = h_2 - h_1 \tag{4.2.4}
\]
Further derivation of the equations (4.2.3) and (4.2.4), using the concept that enthalpy is a function of temperature for semi-perfect gases, followings of work can be obtained.

Turbine work: \[ W_t = C_p (T_3 - T_4) \] (4.2.5)

Compressor work: \[ W_c = C_p (T_2 - T_1) \] (4.2.6)

Isentropic efficiency (comparison of ideal work to actual work) of both compressor and Turbine:

Compressor \[ \Pi_c = \frac{(h_2 - h_1)}{(h_2 - h_1)} = \frac{(T_{2s} - T_1)}{(T_2 - T_1)} \] (4.2.7)

Turbine \[ \Pi_t = \frac{(h_3 - h_4)}{(h_3 - h_{4s})} = \frac{(T_3 - T_4)}{(T_3 - T_{4s})} \] (4.2.8)

Isentropic temperature ratio can be obtained from pressure ratio

\[ \frac{T_{2s}}{T_1} = (\Pi_c)^{(\gamma - 1)/\gamma} \] (4.2.9) \[ \frac{T_3}{T_{4s}} = (\Pi_t)^{(\gamma - 1)/\gamma} \] (4.2.10)

where \( \frac{P_2}{P_1} = \Pi_c \) (compressor pressure ratio)

Obtaining the value of \( T_{2s} \) from equation (7.1) and then substituting it to equations (6.1), we get the following expression for \( T_2 \).

\[ T_2 = T_1 \left[ 1 + \left( \frac{\Pi_c}{\eta_c} \right)^{\gamma - 1} \right] \] (4.2.11)

On substituting these values of \( T_2 \) in equation (5.3)

\[ W_c = C_p T_1 \left( \frac{\Pi_c}{\eta_c} \right)^{\gamma - 1} \] (4.2.12)

So the power required by the compressor can be calculated

\[ P_c = \dot{m} C_p T_1 \left( \frac{\Pi_c}{\eta_c} \right)^{\gamma - 1} \] (4.2.13)

Torque absorbed by the compressor can be calculated by

\[ \tau_c = \frac{P_c}{\omega_c} \] (4.2.14)
4.3. TURBOCHARGER MAPS

Performance of a turbo can usually be presented using corrected performance variable, because without the correction, these maps would only be effective for the conditions under which they were measured. These corrections are made using dimensional analysis and the correction equations evaluates such performance variables, depending upon the instantaneous inlet pressure and temperature.

4.3.1. COMPRESSOR MAP

![Fluid Dynamic Compressor map with normalized mass flows & efficiencies](source: Introduction to modelling & Control ICE Systems, Lino Guzzella & Christopher H Onder)

[111] Predominantly there are four performance variable for compressor map:

- Compressor pressure ratio  \( \Pi_c = P_2 / P_1 \)
- Corrected compressor speed  \( \tilde{\omega}_c = \omega_c * \sqrt{T_{1,0}/T_1} \) where \( T_{1,0} \) is the compressor inlet temperature at which the map was measured originally.
- Corrected compressor mass flow  \( \dot{m}_{c,cor} = (P_{1,0}/P_1) * \dot{m}_c * \sqrt{T_1/T_{1,0}} \) where \( P_{1,0} \) is ambient pressure at which map was measured originally and \( \dot{m}_c \) compressor mass flow.
- Adiabatic efficiency  \( \eta_c = \left( \frac{\gamma - 1}{\gamma} \right) \) where \( \gamma = \) ratio of specific heat for air
Graph 1: Predicted and measured compressor mass flow vs pressure ratio map

Graph 2: Predicted and measured compressor efficiency vs mass flow map
4.3.2. TURBINE MAP

![Turbine Map Diagram](image)

Fig. 13 Turbine map [source: Compressor Modelling for Control of Automotive two Stage Turbochargers, Oskar Leufven]

[Turbo map also utilizes four performance variable:

- **Turbine expansion ratio** $\Pi_t = \frac{P_3}{P_4}$
- **Corrected turbine speed** $\tilde{\omega}_t = \omega_t \cdot \sqrt{\frac{T_{3,0}^3}{T_3}}$ where $T_{3,0}$ is the turbine inlet temperature at which the map was measured originally.
- **Corrected turbine mass flow** $\tilde{m}_{t, cor} = \left(\frac{P_{3,0}}{P_3}\right) \cdot \tilde{m}_t \cdot \sqrt{\frac{T_3}{T_{3,0}}}$ where $P_{3,0}$ is turbine inlet pressure at which map was measured originally and $\tilde{m}_t$ compressor mass flow.
- **Adiabatic efficiency** $\eta_t = \left(1 - \frac{T_4}{T_3} \frac{1}{1 - \Pi_t \gamma} \right)$ where $\gamma$ = ratio of specific heat for air

So the power required by the compressor can be calculated

$$P_t = \tilde{m}_t \cdot C_p \cdot T_3 \cdot \eta_t \left(1 - \Pi_t \gamma \right) \quad (4.3.2.1)$$

Torque absorbed by the compressor can be calculated by

$$\tau_t = \frac{P_t}{\omega_t} \quad (4.3.2.2)$$
Graph 3  Predicted and measured turbine mass flow vs pressure ratio map

Graph 4  Predicted and measured turbine efficiency vs pressure ratio map
4.3.3. COMBINED TURBOCHARGER EQUATION

The combined equation which describes the dynamics of a turbocharger rotor is described by the following equation.

\[
\frac{d}{dt} N_{TD} = \frac{900}{\pi^2 I_{TD} N_{TD}} \left[ P_t(t) - P_c(t) - P_{zm}(t) \right]
\]

(4.3.3.1)

Where:

- \( I_{TD} = \) turbocharger rotational inertia
- \( P_t = \) Turbine power
- \( P_c = \) Compressor Power
- \( P_{zm} = \) Losses

![Fig. 14 e-Charger model layout](image-url)
4.4. COMPRESSOR MODELLING – PRESSURE RATIO AND EFFICIENCY

Compressor is based on the model proposed by Jenson & Kristensen [13] described in terms of a dimensionless Head parameter $\psi$, which is defined by below equation.

$$
\psi = \frac{C_p T_{in} \left( \frac{Y-1}{\Pi_c^Y - 1} \right) \frac{1}{2} U_c^2}{\frac{1}{2} U_c^2}
$$  \hfill (4.4.1)

Normalized flow rate $\Phi$ defined in eq. (16), is used to express the mass flow rate $\dot{m}$.

$$
\Phi = \frac{\dot{m}}{\pi \rho U_c D_c^2} \hfill (4.4.2)
$$

where $U_c = \frac{\pi}{60} D_c N_c$

Head parameter can be described as a function of normalized flow rate and inlet Mach number.

$$
\psi = \frac{K_1 + K_2 \Phi}{K_3 - \Phi} \hfill (4.4.3)
$$

where $K_i = K_{i1} + K_{i2} M$

Where M is inlet Mach number at the ring orifice of the compressor

$$
M = \frac{U_c}{\sqrt{\gamma R T_{in}}}
$$

The coefficient of the expression $K_i$ can be determined by using least square fitting algorithm.

$$
PR (\pi_c) = \left( \frac{\psi \pi_{0.5} U_c}{C_p T_{in} - \Phi} + 1 \right) \frac{Y}{Y - 1} \hfill (4.4.4)
$$

Compressor efficiency has been derived from normalized flow rate $\Phi$ and polynomial $A_i$

$$
\eta_c = A_1 \Phi^2 + A_2 \Phi + A_3 M + A_4 \hfill (4.4.5)
$$

The coefficients of the polynomial $A_i$ can be derived from the least square fitting algorithm.

Compressor outlet temperature is a function of pressure ratio, efficiency and inlet temperature (Tin), and can be described by the formulae

$$
T_2 = \left( \frac{\pi_c \frac{Y-1}{\eta_c} + 1}{\eta_c} \right) \times T_{in} \hfill (4.4.6)
$$

Compressor Power has been expressed as a function a mass flow of air ($mA$), Pressure ratio, efficiency and inlet air temperature (Tin)

$$
P_c = \left( \frac{\dot{m} C_p T_{in} \pi_c \frac{Y-1}{\eta_c}}{\eta_c} \right) \hfill (4.4.7)
$$

For speeds below 160000 rpm, the agreement between the measured and predicted values was good but for higher speeds the variation between both the values was slightly more.
4.5. TURBINE MODELLING – MASS FLOW AND EFFICIENCY

Turbine model is quite similar to that of a compressor. The relationship is established between turbine speed, pressure ratio, inlet temperature, efficiency and flow rate.

Parameters $K_p$ (normalized flow rate) and $A_t$ (turbine flow area), which are function of turbine speed $N_t$, can be determined using elements of symmetric matrix $A = [a_{11}, a_{12}; a_{21}, a_{22}]$

$$K_p = e^{(a_{11} * N_t) + a_{12}} \quad (4.5.1)$$

$$A_t = e^{(a_{21} * N_t) + a_{22}} \quad (4.5.2)$$

Turbine mass flow is derived from $K_p$, $A_t$ and Pressure ratio, thus making mass flow a function of turbine speed and pressure ratio.

$$m_t = A_t * \sqrt{1 - \Pi_t^{-K_p}} \quad (4.5.3)$$

For deriving efficiency, rotor blade tip speed $U_t = \frac{\pi}{60} D_t N_t$ (which is a function of Pressure ratio and turbine speed), gas velocity at isentropic expansion ($A$-which is a function of pressure ratio) $A = \sqrt{\left(1 - \Pi_t^{1-Y/Y} \right)}$ and Speed parameter $N_{sp} = \frac{N_t}{\sqrt{T_{in}}}$ are calculated.

With the help of $U_t$, $T_3$ (engine out. temperature) and $A$, blade speed ratio is determined ($U$).

$$U = U_t / \left(2 * C_p * A * T_3 \right) \quad (4.5.4)$$

Further on turbine efficiency is determined from Blade speed ratio, Speed parameter and Polynomial $K_i$.

$$\eta_t = K_1 + K_2 N_{sp} + (K_3 + K_4 N_{sp}) U + (K_5 + K_6 N_{sp}) U^2 \quad (4.5.5)$$

Where the values of Polynomial $K_i$ is obtained from least square fitting algorithm.

Like the compressor model, Turbine outlet temperature is also a function efficiency, Pressure ratio and Inlet temperature (Engine outlet temperature $T_3$).

$$T_4 = \frac{T_3}{\left(1 - \eta_t * (1 - \Pi_t^{1-Y/Y}) \right)} \quad (4.5.6)$$

And Turbine Power ($P_t$) likewise is computed by the formula:

$$P_t = m_t * C_p * T_3 * \eta_t * \left(1 - \Pi_t^{1-Y/Y} \right) \quad (4.5.7)$$
4.6. ENGINE MODEL

The engine model was provided by Honeywell. It was a mid-size turbocharged gasoline passenger car engine.

4.6.1. PRESSURE MODEL

The pressure difference ($\Delta p$), is a function of engine outlet mass flow ($m_E = m_T + m_{WG}$, which is the sum of turbine mass flow and mass flow through the waste-gate (if the waste-gate is open)), density of exhaust air ($d$) and resistance to the exhaust ($A$). Turbine outlet pressure is determined through delta $p$, and consequently Engine Outlet pressure ($P_3$) is derived. Hence making $P_3$ a function of turbine mass flow ($m_T$) as well.

$$P_3 = P_4 \times \pi_t$$
$$P_4 = P_{atm} + \Delta p$$  \hspace{1cm} (4.6.1) \hspace{1cm} where $\Delta p = f(m_E, A, d)$

4.6.2. TEMPERATURE MODEL

$$T_3' = f(m_A, m_F, T_{2i}, N_e)$$ \hspace{1cm} (4.6.2)

where $T_{2i} = T_{2i}/100$ (100K) $T_{2i}$ = intercooler temperature[K]
$N_e = N_e'/1000$ $N_e'$ = engine RPM
$m_A = 10 \times m_A'$ $m_A'$ = inlet mass flow of air [kg/sec]
$m_F = 10 \times m_F'$ $m_F'$ = Mass flow of fuel [kg/sec]

4.6.3. MASS FLOW MODEL

Mass flow into the engine was determined from the volumetric efficiency of the engine.

$$m_A = f(T_{2i}, P_{2m}, N_e)$$ \hspace{1cm} (4.6.3)

where $P_{2m}$ = Intake manifold out. pressure
4.7. OTHER SUBSYSTEMS OF MODEL

Various subsystems have been incorporated to make the turbocharger model more accurate. These subsystems facilitate to make the Simulink model more detailed and more similar to a physical model.

4.7.1. INTERCOOLER

The intercooler is placed downstream of the compressor. The primary purpose of the intercooler is to cool down the pressurized inlet air from compressor, thus lowering the danger of pre-detonation of the fuel-air mixture, prior to ignition and also makes the incoming charge denser. So primary work of intercooler model is temperature change.

4.7.1.1 TEMPERATURE MODEL

Intercooler efficiency ($\eta_{\text{int}}$) is the primary factor, which influences the ability of the intercooler to reduce the temperature of compressed air.

$$T_{2t} = T_2 \ast (1 - \eta_{\text{int}}) + (\eta_{\text{int}} \ast T_{\text{cool}}) \quad (4.7.1.1)$$

For simplification purposes, intercooler efficiency is considered around 65-70%[15] and $T_{\text{cool}}$ is approximated to be equal to ambient temperature (298 K)

4.7.1.2 PRESSURE MODEL

In this model we, considering the intercooler as a static flow restriction. Hence by applying Bernoulli’s law for mass flow for an incompressible fluid, we can derive the following eq.

$$\frac{dp}{dt} = \frac{A \ast mA_{\text{int}}^2}{2 \ast C_d \ast \rho_{\text{int}}} \quad (4.7.1.2)$$

$mA_{\text{int}}$ = mass flow rate through intercooler $A$ - Area of intercooler $C_d$ - Discharge coefficient

$\rho_{\text{int}}$ = Intercooler density

Since the pressure drop across the intercooler is not substantial, so the intercooler pressure model is of less relevance for the complete turbocharger model

4.1.1.1 Density Model

Density model was used to employed to determine intercooler pressure & manifold flow rate

$$\rho_{\text{int}} = \frac{2 \ast P_2 \ast (1 + 6)}{R \ast T_2} \quad (4.7.1.3)$$
4.7.2. INTAKE MANIFOLD
The intake manifold is placed upstream of the engine. This model governs the airflow into the engine cylinders, and is a critical factor for the performance of the engine, as the airflow inducted into the cylinder, greatly influences the amount of fuel inducted and hence the engine power. It can be roughly considered to be a reservoir, where flow from the intercooler enters the reservoir and flow into the cylinder, leaves the reservoir.

4.7.2.1 MASS FLOW MODEL
Through the mass flow model, inlet mass flow to the intake manifold is determined (which is also considered to be equal to the mass flow through the intercooler, as the pressure drop across the intercooler is relatively small and hence neglected, while the outlet flow is determined from the engine model (with the help of engine speed, volumetric efficiency of engine and manifold pressure).

\[
m_{A_{\text{int}}} = 2 \cdot \rho_{\text{int}} \cdot dP \cdot A \quad (4.7.2.1)
\]

\(dP\) - Pressure difference across the manifold \hspace{1cm} A - Manifold Area

4.7.2.2 PRESSURE MODEL
Pressure model is based on the mass flow difference across the intake manifold. Pressure is the most important aspect of the intake manifold model as there is an appreciable decline in pressure in the manifold.

\[
P_{m} = (Y \cdot T_{2i} \cdot dmA \cdot R) / V \quad (4.7.2.2)
\]

\(dmA\) - mass flow difference across the manifold \hspace{1cm} V - Manifold volume

4.7.3. WASTE-GATE
The dynamics of the waste-gate model are assumed to be quite rapid as compared to other dynamic processes in the engine. Hence a quasi-steady model for waste-gate flow is used.

The waste-gate flow is determined from the Pressure difference between the manifold and compressor pressure (\(P_{2}-P_{m}\)). This method is not the most accurate, but it is being used as it was more practical to use such a model, keeping the scope of thesis in mind (as our intention is to make a model to mimic the behavior of the waste-gate, rather than getting an accurate value).

\[
m_{W} = A_{w} \cdot k \cdot \sqrt{dP} \quad (4.7.2.3)
\]

\(dP\) - pressure difference across the manifold \hspace{1cm} A_{w} - Waste-gate area (position) \hspace{1cm} k - resistance to flow
5. TURBOCHARGER SIMULINK MODEL

Fig. 15 Scheme of Turbocharger Simulink model


5.1. MODEL DESCRIPTION

Above model tries to replicate a waste-gated turbocharger for a gasoline engine. Turbocharger model has predominantly two main components, namely Compressor and Turbine connected to an engine model on either side. The compressor model which is based on Jensen and Kristensen [13] model for the compressor, provides inputs like compressor pressure ($P_2$ from Pressure ratio) and Temperature ($T_2$), to the engine and power ($P_c$) for Turbocharger speed model. While turbine model, which is influenced from Mooral and Kolmanovsky [14] model, receives input from the engine like Temperature ($T_3$) and Pressure ($P_3$), to provide Turbine mass flow rate ($mT$), turbine temperature, pressure and Power (for Turbocharger speed model).

In between compressor and engine, there are some sub-systems like Intercooler and Intake manifold. These sub-systems evaluate degradation of temperature and pressure in the compressed air mass flow, from the compressor to the engine. Additionally, there is a separate subsystem, for calculation of Turbine outlet Pressure ($P_4$) from the ambient pressure, using the pressure gradient ($\Delta p$). There is also a function within the turbine model for mass flow through the waste-gate $mW$ (for total engine outlet mass flow $mE$).
In this section, the aim is to test the turbocharger model under various test conditions. Each of the test conditions tries to simulate different on-road scenarios that an actual turbocharger has to go through. These test conditions might range from normal road conditions to extreme conditions, which would help to test the functionality as well as the durability of the turbocharger.

In addition to the test conditions, some of the critical inputs like mass flow and efficiency are altered (special care has been taken to ensure, that these variations of values are within the scope of the reference turbo-map, so it would be possible to get a reasonable result from the turbocharger). By degrading these values, the effort is to replicate various turbocharger failures. For example, degradation of inlet air mass flow to the compressor can cause lowering of engine power, and the root cause of this failure can be connected to clogging of the air filter, presence of some foreign material in the intake system or collapsed intake pipe. Similarly, degradation of the compressor efficiency can be linked to the lack of lubricating oil or foreign material in the intake system, which might lead to the failure of Journal bearings.

The idea of conducting sensitivity is to correlate the critical turbo parameters with actual field failures. Fig. 17 illustrates this correlation between the turbocharger mass flow and the efficiency with the end failures of turbochargers and the symptoms associated with those failures. Such analysis can help in understanding the conditions leading to each of these failures.

This variation in mass flow and efficiency (both compressor and turbine) is achieved by degrading the flow map and the efficiency map of both compressor and turbine. Due to degradation of the map, certain values of coefficients of Polynomial A_i and K_i are obtained. These values are then subsequently used in turbocharger model, to obtain the desired degradation of the model. Degradation of only mass flow (both compressor and turbine) and efficiency (both compressor and turbine) is being considered because these are the most critical parameters for a turbocharger, and their degradation can be related to most the turbocharger failures. If the failure root cause chart is carefully studied, then it can be analyzed, that major cause of any turbocharger related failure – Foreign object in air path, lack of lubrication, clogging of air filter, will cause fluctuation of flow rate and efficiency. These can actually be identified as an indicator of the upcoming failure. And these indicators can be vital for not only diagnosis of a failure, that has already occurred but can also be used for prognosis purposes as well. Thus, identifying the failure before it can even occur and providing crucial time for rectification measures to be carried out and prevent failure.

With the Automotive industry, shifting the focus more on Autonomous or Driverless vehicles, such an approach can be crucial in vehicle health monitoring system, which are deployed in such vehicles.
<table>
<thead>
<tr>
<th>Causes</th>
<th>Symptoms</th>
<th>Failure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Decrease in Turbocharger efficiency</td>
<td>Lack of Lubricating Oil</td>
<td>Blueing or discoloration of bearings of the turbine shaft</td>
</tr>
<tr>
<td></td>
<td>Delay in delivery of Oil</td>
<td>Excessive temperature</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Failure of Journal Bearings</td>
</tr>
<tr>
<td>Decrease in Turbocharger efficiency</td>
<td>Oil cracking or break-down due to excessive temperature</td>
<td>Production of carbonaceous(sludge) material</td>
</tr>
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<td></td>
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<td>Production of highly corrosive organic acids</td>
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<td></td>
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</tr>
<tr>
<td>Decrease in Turbocharger efficiency</td>
<td>Foreign material like dust and dirt entering through the intake system</td>
<td>Journal Bearing wear</td>
</tr>
<tr>
<td></td>
<td>Introduction of natural by-product of combustion- ash, soot, water, unburnt fuel, into the lubrication system</td>
<td>Bearing-housing bore wear</td>
</tr>
<tr>
<td></td>
<td>Tiny metal fragments produced by the wear and tear in the engine, enters the turbo through lubrication oil</td>
<td>Blocking of internal oil passages</td>
</tr>
<tr>
<td>Decrease in Compressor Mass flow</td>
<td>Entry of foreign material through intake or exhaust system</td>
<td>Noise</td>
</tr>
<tr>
<td></td>
<td>Engine lacks power</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Metal fragments in the intake air due to damage of compressor wheel</td>
<td>Excessive Lubrication oil consumption</td>
</tr>
<tr>
<td>Decrease in Compressor Mass flow</td>
<td>Exhaust Gas leakage</td>
<td>Noise</td>
</tr>
<tr>
<td></td>
<td>Dirty Air filter</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Restricted or Collapsed pipe from turbo to intake manifold</td>
<td>Engine lacks power</td>
</tr>
<tr>
<td>Too fast acceleration at initial start</td>
<td>Excessive rotating assembly play</td>
<td>Damaged compressor wheel</td>
</tr>
</tbody>
</table>
6.1. SIMULATION TEST MANEUVERS

6.1.1. CONSTANT RPM WITH CHANGE IN THE BOOST SETPOINT

This test scenario tries to replicate a vehicle trying to go up the hill. RPM is set constant around 2500 rpm and boost pressure points are set between 1.4 – 1.8 bar. In simple terms RPM is constant but the torque is increasing, as per boost requirement. The test duration is 100 sec and in the initial 40 sec; the turbocharger is given time to reach lower pressure point in order to ensure that the turbocharger is in a steady state before the start of upward pressure boost ramp.

Additionally, by keeping rpm constant, the effort is to simplify engine dynamic operation, hence providing good idea about TC dynamics.

![Fig. 17 Car going up-hill](image)

**Graph 5 (a)**

**Graph 5 (b)**

**Graph 5 (c)**

**Graph (d)**
Graph 5 (d) and Graph 5 (e)

**Graph 5 Illustration of turbocharger plots, for constant RPM and changing pressure boost**

a) Comparison between intake manifold (orange) pressure and pressure boost set-point (blue). Boost is constant in the first 40 sec and then steps up to 1.8 bar. Manifold pressure tries to match boost pressure, when boost sets in and the pressure rises gradually (transient phase of the manifold plot illustrates turbo-lag) to achieve boost pressure.

b) Engine Outlet Pressure — similar to the manifold plot, engine outlet pressure rising gradually when the boost pressure sets in.

c) Compressor outlet pressure — it mimics the intake manifold graph, but it has higher pressure values.

d) Turbine outlet pressure — it more or less follows the trend of waste-gate position and the pressure value increases and decreases depending upon opening or closing of waste-gate, only difference being the pressure rise in the second transient phase being more substantial.

e) Waste-gate position — it opens up when there is excess pressure (thereby increasing exhaust pressure value) and closes (goes to minimum value) whenever pressure boost is required.

f) Turbocharger — Similar to intake manifold, turbocharger speed also increases rapidly in case of transient phase and remains constant during steady state.

### 6.1.2. CHANGING RPM WITH CHANGE IN THE BOOST PRESSURE SETPOINT

In this test condition, RPM changes according to boost requirement. In simple terms, this condition is trying to replicate a rapid car acceleration from a steady state. Range of RPM change is 2500-4500 RPM and boost pressure points are set between 1.4 – 1.8 bar (RPM rise is a function of mass flow of fuel). Test duration is 100 sec, out of which, in initial 40 secs, the turbocharger is given time to reach lower boost pressure point, to ensure that turbocharger is in a steady state, before start of upward pressure boost ramp.
Graph 6 Plots for turbocharger, for changing RPM and changing pressure boost

a) Engine RPM – RPM increases gradually from 2500-4500 RPM, depending upon the engine load
b) Pressure Set point – Ramps up from 1.4 to 1.8 bar after 40 sec  c) Comparison between intake manifold (orange) pressure and pressure boost set-point (blue).  b) Engine Outlet Pressure – it increases steadily in the transient phase and once waste-gate opens up, pressure rise becomes more slower  c) Compressor outlet pressure – in the initial part of transient phases it mimics the manifold graph, but once the waste-gate opens up completely, the pressure rise becomes very slow and like engine pressure, it reaches steady state as the waste-gate hits the steady state  d) Turbine outlet pressure – It more or less follows the trend of waste-gate position and the pressure value increases and decreases depending upon opening or closing of waste-gate  e) Waste-gate position – initially it opens (slightly) to maintain the initial steady state of manifold pressure and when there is a step up of boost pressure, it closes a little bit and then opens up completely first steadily and later rapidly to control the excess pressure  d) Turbocharger – Similar to compressor outlet pressure, RPM increases rapidly once boost pressure sets-in and then, while in the transient phase, it increases steadily and after waste-gate rapidly opens, becomes steady.

6.1.3. CHANGING RPM WITH UPWARD AND DOWNWARD BOOST PRESSURE RAMP

In this test scenario, similar to the previous case, RPM changes according to pressure boost requirement. As the boost pressure set-point moves from 1.4 to 1.8 bar, engine RPM also shows a rapid upsurge, which is followed by a steady state (constant RPM) at higher boost pressure point (1.8 bar). Then there is a step-down in boost pressure (1.8-1.4 bar) and engine RPM shows a rapid down surge. This maneuver is, again, followed by a steady state. This situation closely replicates the lane change maneuver on a highway (for overtaking another car).

Fig. 19 Lane Change Maneuver
Graph 7 (a)

Graph 7 (b)

Graph 7 (c)

Graph 7 (d)

Graph 7 (e)

Graph 7 (f)
Graph 7 Plots for turbocharger, for changing RPM with upward and downward boost ramp

a) Engine RPM  b) Pressure Boost Set-point  c) Intake manifold pressure  d) Compressor Outlet Pressure  
e) Engine Outlet Pressure  f) Turbine Outlet pressure  g) Waste-gate operation  h) Turbocharger Speed

6.1.4. CHANGING RPM WITH MULTIPLE UPWARD AND DOWNWARD BOOST RAMP

This case is very similar to the previous case, with RPM changing according to pressure boost requirement, the only difference being the presence of multiple (in the given case only two) pressure boost ramps. RPM range is 2500 – 4500 while pressure set-point range is 1.4 – 1.8 bar.

This maneuver is, again, preceded and followed by a steady state. This situation closely replicates the double lane change maneuver on highway (for overtaking multiple cars).
Graph 8 Plots for turbocharger, for changing RPM with multiple upward and downward boost ramp

a) Engine RPM  b) Pressure Boost Set-point  c) Intake manifold pressure  d) Compressor Outlet Pressure  
e) Engine Outlet Pressure  f) Turbine Outlet pressure  e) Waste-gate operation  f) Turbocharger Speed
6.1.5. CONSTANT RPM WITH UPWARD AND DOWNWARD PRESSURE BOOST RAMP

This test case follows a trend of the increase in boost pressure (1.4-1.8 bar) followed by a steady state and then a downward pressure boost ramp (1.8-1.4 bar), with the RPM being constant (2500 RPM) throughout. This test case is trying to simulate a vehicle on the cruise control, going up and down a gradient. Test duration is 100 sec, out of which, the initial 20 secs are used up in getting the turbocharger to the lower boost pressure point before the start of the upward pressure boost ramp.

*Fig. 21 Cruise control on gradient*
Graph 9 Plots for turbocharger, for constant RPM with upward and downward boost ramp

a) Intake manifold outlet pressure with boost set-point   b) Compressor Outlet Pressure

c) Engine Outlet Pressure   d) Turbine Outlet pressure   e) Waste-gate operation   f) Turbocharger Speed

6.1.6. COMPARISON TO WORLD HARMONIZED TRANSIENT CYCLE

In this section, turbocharger model data is being compared to the actual engine data. Total duration for WHT Cycle is 1800 sec. But in this test maneuver a small portion of the cycle is being considered. Duration of test is 200 sec and RPM range is 2830 – 2910. The initial 20 sec of the cycle have been used to bring the pressure values to a stable point.
Graph 10 Plots for turbocharger, comparison with WHTC cycle

a) Engine RPM    b) Pressure Boost Set-point    c) Intake manifold pressure    d) Compressor Out. Pressure
 e) Engine Out. Pressure    f) Turbine Out. pressure    g) Waste-gate operation    h) Turbocharger Speed
6.2. DEGRADATION IN TURBOCHARGER MAP

Predominantly, there are two kinds of mass flowing through the turbocharger system. The first is the compressor mass flow (which is, basically, the fresh inlet air entering the compressor) and the second – the turbine mass flow (containing the products of combustion occurring in the engine). This mass flow is the driving force for any turbocharged system, and if there is a certain degradation in it, then consequently the performance of the turbocharger also gets degraded, and it might lead to the turbocharger’s failure. There can be many reasons for the degradation in mass flow (compressor and turbine) and it can be attributed to the failure of components, obstruction of the intake system, presence of foreign material, etc.

In this section, the aim is to simulate such scenarios and then relate them to actual turbocharger failure. As explained in the beginning of the Sensitivity section, the degradation in mass flow is obtained by degrading the flow map. Three sets of values are taken – Normal value (with no degradation), 5% degradation and 10% degradation. For each set, certain values of coefficients of Polynomial $A_i$ and $K_i$ are obtained from the map. These polynomials are then substituted into the Simulink model. Degradation is carried out in terms of Compressor Mass Flow, Turbine Mass Flow, Compressor Efficiency and Turbine Efficiency. For simplicity, only two test maneuvers are being considered in the sensitivity study – Constant speed and changing speed.

6.2.1. DEGRADATION IN FLOW MAP (COMPRESSOR)

6.2.1.1 Constant RPM with change in the boost pressure set-point

Graph 10(a): Intake Manifold Pressure  
Graph 11(b): Engine Outlet Pressure
Graph 11 (c): Compressor Outlet Pressure

Graph 11(d): Turbine Outlet Pressure

Graph 11(e): Turbo-Speed

Graph 11(f): Waste-gate

Graph 11 Illustration of compressor mass flow variation plots for constant RPM and changing boost

a) Intake manifold pressure variation - Variation is occurring only in the transient phase and it is very small. Compressor Pressure ($P_2$) is a function compressor mass flow and manifold pressure ($P_{m}$) is also dependent upon compressor pressure. Hence $P_{m}$ degrades with mass flow degradation. b) Engine outlet pressure variation - pressure variation is very small (occurring during steady state phase) and engine outlet pressure increases with degradation of compressor mass flow. It happens due to increase of TC speed during steady state. c) Compressor Outlet Pressure - Variation is occurring only in the transient phase and it is very small. $P_2$ is directly depends upon mass flow. Hence it degrades with mass flow. d) Turbine Outlet Pressure - Pressure plot follows waste-gate plot, variation occurs in the steady state phase, with the pressure increasing (very slightly) with degradation in mass flow. e) Turbo-Speed - Prominent speed variation during steady state phase, speed increases with mass flow degradation. Speed increases to compensate for the loss of mass flow and maintain pressure ratio. f) Waste-gate - Waste gate position plot also shows variation during steady state, with the waste-opening more and more with mass flow degradation (controller function)
6.2.1.2 Changing RPM with change in the boost pressure set-point

Graph 12 (a): Intake Manifold Pressure

Graph 12 (b): Engine Outlet Pressure

Graph 12 (c): Compressor Outlet Pressure

Graph 12 (d): Turbine Outlet Pressure

Graph 12 (e): Turbo-Speed

Graph 12 (f): Waste-gate
Graph 12 Illustration of compressor mass flow variation plots for changing RPM and changing boost

a) Intake manifold pressure variation - variation is occurring in the transient phase and it is very small. Compressor Pressure ($P_2$) is a function of compressor mass flow and manifold pressure ($P_m$) is also dependent upon compressor pressure. Hence $P_m$ degrades with mass flow degradation.  
b) Engine outlet pressure variation - shows variation during boost pressure steady state region. Engine outlet pressure increases with mass flow degradation, due to increase of speed during steady state phase  
c) Compressor Outlet Pressure - variation is occurring in the transient phase and it is very small. Decrease in flow rate leads to decrease in compressor pressure ratio, which in turn affects compressor pressure.  
d) Turbine Outlet Pressure - variation occurs in the steady state phase, with the pressure increasing (very slightly) with degradation in mass flow. As waste-gates open more with mass flow degradation Turbine outlet pressure also increases  
e) Turbine Speed – Prominent speed variation during steady state phase, speed increases with mass flow degradation. With mass flow degrading TC speeds up to maintain the pressure ratio.  
f) Waste-gate – Waste gate position plot also shows variation during steady state, with the waste-opening more and more with mass flow degradation (controller function)

6.2.2. DEGRADATION IN FLOW MAP (TURBINE)

6.2.2.1 Constant RPM with change in the boost pressure set-point

Graph 13 (a): Intake Manifold Pressure

Graph 13 (b): Engine Outlet Pressure

Graph 13 (c): Compressor Outlet Pressure

Graph 13(d): Turbine Outlet Pressure
Graph 13 Illustration of turbine mass flow variation plots for constant RPM and changing boost

a) Intake manifold pressure variation - variation is occurring in the transient phase. Degradation of mT leads to decrease in Turbine Power Pt. As result turbo-speed gets degraded, which in turn affects manifold pressure, leading to decrease in pressure values  
b) Engine outlet pressure variation - Engine outlet pressure variation – variation in pressure value in the boost pressure steady state region. Due to closing of waste-gate (with degraded mass flow) increased resistance to pressure leads to significant increase in engine outlet pressure  
c) Compressor Outlet Pressure - variation is occurring during transient phase. Degradation of mT leads to decrease in Turbine Power Pt, which in turn degrades compressor pressure  
d) Turbine Outlet Pressure – $P_4$ is directly dependent on waste-gate position, and with waste-gate closing, pressure also decreases with turbine mass flow degradation  
e) Turbine Speed – Very small speed variation during transient phase, speed decreases with mass flow degradation (as turbine power gets degraded)  
f) Waste-gate – Waste gate position plot also shows variation during boost pressure steady state region, with waste-gate closing more with mass flow degradation (controller function)

6.2.2.2 Changing RPM with change in the boost pressure set-point

Graph 14 (a): Intake Manifold Pressure  
Graph 14 (b): Engine Outlet Pressure
Graph 14 (c): Compressor Outlet Pressure  
Graph 14 (d): Turbine Outlet Pressure  
Graph 14 (e): Turbo-speed  
Graph 14 (f): Waste-gate

Graph 14 Illustration turbine mass flow variation plots for changing RPM and changing boost

a) Intake manifold pressure variation - variation is occurring in the transient phase. Degradation of $m_T$ leads to decrease in Turbine Power $P_t$. As result turbo-speed gets degraded, which in turn affects manifold pressure, leading to decrease in pressure values.  
b) Engine outlet pressure variation - Engine outlet pressure variation – variation in pressure value in the boost pressure steady state region. Due to closing of waste-gate (with degraded mass flow) increased resistance to pressure leads to significant increase in engine outlet pressure.  
c) Compressor Outlet Pressure - variation is occurring during transient phase. Degradation of $m_T$ leads to decrease in Turbine Power $P_t$, which in turn degrades compressor pressure.  
d) Turbine Outlet Pressure – $P_4$ is directly dependent on waste-gate position, and with waste-gate closing, pressure also decreases with turbine mass flow degradation.  
e) Turbine Speed – Very small speed variation during transient phase, speed decreases with mass flow degradation (as turbine power gets degraded).  
f) Waste-gate – Waste gate position plot also shows variation during boost pressure steady state region, with waste-gate closing more with mass flow degradation (controller function).
6.2.3. DEGRADATION IN EFFICIENCY MAP (COMPRESSOR)

6.2.3.1 Constant RPM with change in the boost pressure set-point

Graph 15 (a): Intake Manifold Pressure

Graph 15 (b): Engine Outlet Pressure

Graph 15 (c): Compressor Outlet Pressure

Graph 15 (d): Turbine Outlet Pressure

Graph 15 (e): Turbo-speed

Graph 15 (f): Waste-gate
Graph 15 Illustration of compressor efficiency variation plots for constant RPM and changing boost.

a) Intake manifold pressure variation - variation is occurring only in the transient phase and it is very small. Degradation of efficiency leads to increase in Compressor Power \( P_c \) (or work). As result turbo-speed gets degraded, which in turn affects manifold pressure, leading to a slight decrease in manifold pressure values, in the boost region.  
b) Engine outlet pressure variation. Variation in very prominent (occurring during steady state). Since waste-gate closes (due to controller action) more with compressor efficiency degradation, thus providing more resistance to engine outlet pressure  
c) Compressor Outlet Pressure - Degradation of compressor efficiency directly affects Turbo-speed, which in turn affects compressor pressure  
d) Turbine Outlet Pressure - Variation occurs in the steady state phase, with pressure decreasing with degradation in efficiency. Turbine pressure gets directly influenced with waste gate position  
e) Turbine Speed – Very small speed variation during steady phase (higher boost point), speed decreases with efficiency degradation  
f) Waste-gate – waste gate position plot also shows prominent variation during both steady state phases, with waste-gate closing more with efficiency degradation during steady state (owing to controller action)

6.2.3.2 Changing RPM with change in the boost pressure set-point

Graph 16 (a): Intake Manifold Pressure  
Graph 16 (b): Engine Outlet Pressure  
Graph 16 (c): Compressor Outlet Pressure  
Graph 16 (d): Turbine Outlet Pressure
Graph 16 Illustration of compressor efficiency variation plots for changing RPM and changing boost

a) Intake manifold pressure variation (Pm) – very small pressure degradation with efficiency degradation
b) Engine outlet pressure variation - prominent increase in $P_3$(occurring during boost pressure steady state) with degradation of compressor efficiency, due to more closing of waste-gate cause the resistance to $P_3$ to increase. c) Compressor Outlet Pressure – Similar to Pm, compressor pressure decreases with efficiency degradation  d) Turbine Outlet Pressure -Variation occurs in the steady state phase, with pressure decreasing with degradation in efficiency. Turbine pressure gets directly influenced with waste gate position  e) Turbine Speed – Decrease in efficiency leads to increase in Compressor work, leading to degradation of speed as well f) Waste-gate – Waste gate position plot also shows prominent variation during steady state phases, with waste-gate closing more with efficiency degradation

6.2.4. DEGRADATION IN EFFICIENCY MAP (TURBINE)

6.2.4.1 Constant RPM with change in the boost pressure set-point

Graph 17 (a): Intake Manifold Pressure
Graph 17 (b): Engine Outlet Pressure
Graph 17 (c): Compressor Outlet Pressure

Graph 17 (d): Turbine Outlet Pressure

Graph 17 (e): Turbo-speed

Graph 17 (f): Waste-gate

Graph 17 Illustration of turbine efficiency variation plots for constant RPM and changing boost

a) Intake manifold pressure variation is occurring only in the transient phase with pressure decreasing with efficiency degradation  
b) Engine outlet pressure variation in very prominent (occurring during boost pressure steady state) and engine outlet pressure increases with degradation of turbine efficiency.  
c) Compressor Outlet Pressure variation is occurring only in the transient phase with pressure decreasing with efficiency degradation  
d) Turbine Outlet Pressure variation occurs in transient phase, with pressure decreasing with degradation in efficiency  
e) Turbine Speed variation during transient phase, speed decreases with efficiency degradation  
f) Waste-gate – Waste gate position plot shows prominent variation during both steady state phases, with waste-gate closing more with efficiency degradation
6.2.3.2 Changing RPM with upward and downward boost ramp

Graph 18 (a): Intake Manifold Pressure

Graph 18 (b): Engine Outlet Pressure

Graph 18 (c): Compressor Outlet Pressure

Graph 18 (d): Turbine Outlet Pressure

Graph 18 (e): Turbo-speed

Graph 18 (f): Waste-gate
Degradation of parameters like flow and efficiency are characterized by deterioration of turbocharger performance. A clear trend of reduction in compressor and manifold pressure during boost pressure transient region with deterioration of flow map and efficiency map is quite visible. Flow characteristics of compressor model have been modelled in such a way that the pressure ratio has been expressed as a function of mass flow rate and speed. Degradation of mass flow affects the pressure ratio and with the ambient temperature ($P_1$) being considered constant, compressor outlet pressure ($P_2$) gets directly affected with the decrease in pressure ratio. Manifold pressure ($P_m$) is directly linked to $P_2$, so with it, $P_m$ also decreases. Degradation of compressor efficiency leads to an increase of Compressor work, which again adversely affects the pressure ratio model. The reason for the pressure degradation occurring only in the transient phase, can be explained by the fact that during the steady phase, influenced by the controller function either the Turbocharger speeds up or the waste-gate closes (or sometimes opens), with degradation of mass flow or efficiency, to maintain the pressure ratio. But in the transient phase due to the surge in boost pressure requirement, the controller can’t perform its function properly.

In case of turbine model, the trend of $P_2$ and $P_m$ decrease with degradation of mass flow and efficiency continues. Since these degradations clearly affect the turbine power, so the degraded turbine power would lead to degradation of $P_m$. In the sensitivity of turbine mass flow and efficiency, the waste-gate plays an important part. Influenced by the controller, it actively tries to negate the effect of degradation during the steady state by closing more, but in the transient phase fails due to surge in boost demand. The turbine outlet pressure ($P_4$) is directly influenced by the waste-gate and as can be seen by $P_4$ plots, gets manipulated by waste-gate position. Another important observation of the sensitivity study is the prominent increase in engine outlet pressure ($P_3$) during the steady state (in the higher boost pressure region), with degradation of mass flow and efficiency. It is predominantly due to closing of the waste-gate in the steady state region (to negate the effects of degradation), which causes an increase in the resistance to the engine outlet flow. Consequently increase in resistance leads to increase in $P_3$. 

6.3. TURBOCHARGER DEGRADATION SUMMARY
7. SIMULINK MODEL - ECHARGER

![Simulink Model Diagram]

Fig. 23 Scheme of e-Charger Simulink model
7.1. MODEL DESCRIPTION

E-charger model is almost identical to the existing TC model, the only visible difference being the inclusion of an additional compressor and controller. This additional compressor is placed downstream of the main compressor and receives all its output \((P_2 \text{ and } T_2)\) and provides pressure and temperature \((P_{2,e} \text{ and } T_{2,e})\) inputs to the intercooler and manifold. This compressor is controlled by an LQR controller, which provides the compressor with RPM as input. Like the TC model e-Charger behaves like a waste-gated turbocharger, the only difference being the inclusion of an additional compressor, which gets activated by the controller, whenever there is a pressure boost requirement.

7.2. LQR CONTROLLER

For the specific control of the additional compressor, Linear Quadratic Regulator was developed.\(^{[14]}\) LQR controller is based on the theory of optimal control for dynamical system. The main aim of this model is to provide a linear controller to minimize the quadratic cost function and the regulator keeps the system stabilized.

Specifically, in the e-charger model LQR, provides rpm input to the additional compressor.

![LQR controller layout](image)

The LQR model predominantly consists of three systems:

Kalman filter- it receives inputs like Intake manifold pressure \((P_m)\), Waste-gate position. Its primary functions are the estimation of the internal state for Linear Quadratic Controller, Removal of Noise (Noise was not considered in the thesis) and provision of integral action.

Linear Quadratic function – it receives pressure set-point values and internal state estimation and integral action. Its primary output includes waste-gate and e-charger input.

PID controller - PID receives e-charger input from LG function and converts it into RPM, to be provided as input to e-Charger model.
8. SENSITIVITY STUDY – E-CHARGER

8.1. SIMULATION TESTING MANEUVERS

8.1.1. CONSTANT RPM WITH CHANGE IN THE BOOST PRESSURE SETPOINT

As discussed in the earlier sections, this test simulates the condition of a vehicle going up the hill. Initial 40 secs, have used to bring intake manifold pressure to a stable point.

Graph 19 (a) $P_m$

Graph 19 (b) $P_2$ vs $P_3$

Graph 19 (c) waste-gate

Graph 19 (d) $P_4$
Graph 19 (e) Pc vs Pech

Graph 19 (f) Turbo-speed

Graph 19: Plots for an e-charger, for constant RPM and changing pressure boost

a) Comparison between intake manifold (blue) pressure and pressure boost set-point. Boost is constant in the first 40 sec and then steps up to 1.8 bar. Manifold pressure in the initial steady state condition becomes equal to initial boost point and after pressure boost occurs, tries to achieve final boost set-point and after achieving it, it becomes steady at 1.8 bar. b) Engine Outlet Pressure and compressor outlet pressure – both the plots mimic the intake manifold, but have higher values. c) Waste-gate position – it opens up when there is excess pressure (thereby increasing exhaust pressure value) and closes (goes to minimum value) whenever pressure boost is required. d) Turbine outlet pressure – It more or less follows the waste-gate position and the pressure value increases and decreases depending upon opening or closing of waste-gate. e) e-Charger vs compressor power – plot shows e-charger powering the system during the transient phase. f) Turbocharger – Speed also increases rapidly in case of transient phase and remains constant during steady state.

8.1.2. CHANGING RPM WITH CHANGE IN THE BOOST SETPOINT

Following test maneuver simulates, a sudden acceleration (maybe to overtake another car). Vehicle is at constant speed and then suddenly accelerates when the pressure boost sets in.
a) Comparison between intake manifold (blue) pressure and pressure boost set-point. Boost is constant in the first 40 sec and then steps up to 1.8 bar. Manifold pressure in the initial steady state condition becomes equal to initial boost point and after pressure boost occurs, it increases rapidly to achieve the upper boost point

b) Engine Outlet Pressure – it increases steadily in the transient phase once the boost pressure sets-in and once it reaches peak pressure it becomes steady, while Compressor outlet pressure – in the initial phases it mimics the manifold graph (reaches steady position and increases rapidly once boost pressure sets-in) and then, while in the transient phase, it increases steadily and after waste-gate rapidly opens, pressure becomes steady

c) Waste-gate position – initially it opens (slightly) to maintain the initial steady state of manifold pressure and when there is a step up of boost pressure, it closes a little bit and then opens up completely first steadily and later rapidly to control the excess pressure

d) Turbine outlet pressure (P4) – It more or less follows the waste-gate position and the pressure value increases and decreases depending upon opening or closing of waste-gate

e) e-Charger power vs Compressor power

d) Turbocharger – Similar to compressor outlet pressure, RPM increases rapidly once boost pressure sets-in and then, while in the transient phase, it increases steadily and after waste-gate rapidly opens, becomes steady
8.1.3. CHANGING RPM WITH UPWARD AND DOWNWARD BOOST RAMP

Following test simulates lane change maneuver on a highway. RPM changes according to the boost pressure requirement.

Graph 21 (a) $P_m$

Graph 21(b) $P_2$ vs $P_3$

Graph 21 (c) Waste-gate

Graph 21(d) $P_4$

Graph 21 (e) $P_c$ vs $P_{ech}$

Graph 21(f) Turbo-speed
Graph 21 Plots for an e-charger, for changing RPM with upward and downward boost pressure ramp

a) Comparison between intake manifold (blue) pressure and pressure boost set-point. Boost is constant in the first 35 sec and then steps up to max pressure, maintains steady boost for next 35 sec and then declines to lower boost point. Manifold pressure tries to mimic the boost pressure (increase rapidly to achieve boost pressure, but decreases slowly to achieve lower boost point, as waste-gate take extra time to get rid of excess boost)

b) Engine Outlet Pressure – it increases steadily in the transient phase once the boost pressure sets-in and once it reaches peak pressure it declines when waste-gate opens completely. Compressor outlet pressure – it mimics the manifold graph (in the first transient phase, it increases steadily and in second transient phase it declines rapidly)

c) Waste-gate position – initially it opens (slightly) to maintain the initial steady state of manifold pressure and when there is a step up of boost pressure, it closes a little bit and when manifold achieves boost pressure, opens up first steadily and then almost instantaneously to facilitate step-down in boost pressure

d) Turbine outlet pressure – It more or less follows the waste-gate position and the pressure value increases and decreases depending upon opening or closing of waste-gate

e) e-Charger power vs Compressor power

f) Turbocharger - RPM, increases steadily and then declines steadily, following boost pressure set-points

8.1.4. CHANGING RPM WITH MULTIPLE UPWARD AND DOWNWARD BOOST RAMP

Following test simulates double lane change maneuver on a highway. RPM changes according to the boost pressure requirement, with multiple boost pressure ramps.

Graph 22 (a) Prm

Graph 22(b) P2 vs P3

Graph 22 (c) Waste-gate

Graph 22(d) P4
Graph 22 (e)  

Graph 22 (f)  

Graph 22  Plots for an e-charger, for changing RPM with multiple upward and downward boost pressure ramp  

a) Comparison between intake manifold (blue) pressure and pressure boost set-point. Manifold pressure tries to mimic both boost pressure ramps (increase rapidly to achieve boost pressure, but decreases slowly to achieve lower boost point, as waste-gate take extra time to get rid of excess boost)  
b) Engine Outlet Pressure – it increases steadily in the transient phase once the boost pressure sets-in and once it reaches peak pressure it declines when waste-gate opens completely. Compressor outlet pressure – it mimics the manifold graph (in the first transient phase, it increases steadily and in second transient phase it declines rapidly)  
c) Waste-gate position – initially it opens (slightly) to maintain the initial steady state of manifold pressure and when there is a step up of boost pressure, it closes a little bit and when manifold achieves boost pressure, opens up first steadily and then almost instantaneously to facilitate step-down in boost pressure  
d) Turbine outlet pressure – It more or less follows the waste-gate position and the pressure value increases and decreases depending upon opening or closing of waste-gate  
e) e-Charger power vs Compressor power  
f) Turbocharger – Turbocharger RPM, increases steadily and then declines steadily, following boost pressure set-points.

8.1.5  CONSTANT RPM WITH UPWARD AND DOWNWARD PRESSURE BOOST RAMP

Following test simulates the condition of a vehicle going up and down a gradient, under cruise control. RPM is set at 2500.
Graph 23 Plots for an e-charger, for constant RPM with multiple upward and downward boost pressure ramp

a) Comparison between intake manifold (blue) pressure and pressure boost set-point. Manifold Pressure matches pretty closely with boost pressure set-points (with rapid increase during boost step-up phase and a slightly steady decline during boost step-down phase) b) Engine Outlet Pressure – it increases steadily in the transient phase once the boost pressure sets-in and once it reaches peak pressure it declines when waste-gate opens completely. Compressor outlet pressure – it mimics the manifold graph (though, it gradually reaches the peak pressure value as compared to intake manifold) c) Waste-gate position – initially it opens (slightly) to maintain the initial steady state of manifold pressure and when there is a step up of boost pressure, it closes a little bit and when manifold achieves boost pressure, opens up first steadily and then almost instantaneously to facilitate step-down in boost pressure d) Turbine outlet pressure – It more or less follows the waste-gate position and the pressure value increases and decreases depending upon opening or closing of waste-gate e) e-Charger power vs Compressor power f) Turbocharger – Turbocharger RPM (similar to compressor outline pressure), increases steadily and then declines steadily, following boost pressure set-points.
8.1.6. COMPARISON TO WORLD HARMONIZED TRANSIENT CYCLE

Graph 24 (a) Intake Manifold Pressure

Graph 24 (b) Engine Outlet Pressure

Graph 24 (c) Compressor Outlet Pressure

Graph 24 (d) Turbine Outlet Pressure

Graph 24 (e) Waste-gate position

Graph 24 (f) Turbo-speed

Graph 24 Plots for an e-Charger, Comparison with WHTC
8.2. DEGRADATION IN E-CHARGER MAP

8.2.1. DEGRADATION IN FLOW MAP (COMPRESSOR)

Similar to the turbocharger sensitivity study, polynomial coefficient obtained from the turbo-map is manipulated and used in the Simulink model to obtain degradation (in this case, the degradation of Compressor flow). In the sensitivity study, only two of the testing maneuvers are being considered: Constant Speed and Changing Speed. Three set of values are being considered – Normal value (with no degradation), 5% degradation and 10% degradation.

8.2.1.1 Constant RPM with change in the boost pressure set-point

Graph 25 (a): Intake Manifold Pressure

Graph 25 (b): Engine Outlet Pressure

Graph 25 (c): Compressor Outlet Pressure

Graph 25 (d): Turbine Outlet Pressure
Graph 25 Illustration of compressor mass flow $m_A$ variation plots for constant RPM and changing boost

a) Intake manifold pressure – exhibits pressure variation during boost pressure transient region. Compressor pressure ratio is a function of compressor mass flow and with degradation of mass flow, Manifold pressure also gets degraded  
b) Engine Outlet Pressure – shows variation during boost pressure steady state region. It increases very slightly with mass flow degradation, due to increase of speed during steady state phase  
c) Compressor Outlet Pressure – Shows variation during boost pressure transient phase. Decrease in flow rate leads to decrease in compressor pressure ratio, which in turn affects compressor pressure  
d) Turbine Outlet pressure – shows variation during boost steady state phase. Turbine pressure is greatly influenced by waste-gate position. Pressure increases with mass flow degradation.  
e) Turbo-speed – it shows variation during steady state and increases with mass flow degradation. During steady state speed increases to compensate for the loss of mass flow and maintain pressure ratio.  
f) Waste-gates- it opens more during boost pressure steady state region, with mass flow degradation (controller function)

8.2.1.2 Changing RPM with change in the boost pressure set-point
Graph 26 (c): Compressor Pressure  
Graph 26 (d): Turbine Pressure  
Graph 26 (e): Turbo-Speed  
Graph 26 (f): Waste-gate position  

Graph 26 Illustration of compressor mass flow mA variation plots for changing RPM and changing boost pressure

a) Intake manifold pressure – exhibits pressure variation during boost pressure transient region. Compressor pressure ratio is a function of compressor mass flow and with degradation of mass flow. Manifold pressure also gets degraded  
b) Engine Outlet Pressure – shows variation during boost pressure steady state region. Engine outlet pressure increases with mass flow degradation, due to increase of speed during steady state phase  
c) Compressor Outlet Pressure – shows variation during boost pressure transient phase. Decrease in flow rate leads to decrease in compressor pressure ratio, which in turn affects compressor pressure  
d) Turbine Outlet pressure – shows variation during boost steady state phase. Turbine pressure is greatly influenced by waste-gate position. Pressure increases with mass flow degradation.  
e) Turbo-speed – it shows variation during steady state and increases with mass flow degradation. During steady state speed increases to compensate for the loss of mass flow and maintain pressure ratio  
f) Waste-gates- it opens more during boost pressure steady state region, with mass flow degradation (controller function)
8.2.2. DEGRADATION IN FLOW MAP (TURBINE)

8.2.2.1 Constant RPM with change in the boost pressure set-point

**Graph 27 (a): Intake Manifold Pressure**

**Graph 27 (b): Engine Outlet Pressure**

**Graph 27 (c): Compressor Pressure**

**Graph 27 (d): Turbine Pressure**

**Graph 27 (e): Turbo-Speed**

**Graph 27 (f): Waste-gate position**
Graph 27 Illustration of turbine mass flow mT variation plots for constant RPM and changing boost pressure

a) Intake manifold pressure - variation is occurring during transient phase. Degradation of mT leads to decrease in Turbine Power Pt. As result turbo-speed gets degraded, which in turn affects manifold pressure, leading to decrease in pressure values.  
b) Engine outlet pressure variation – variation in pressure value in the boost pressure steady state region. Due to closing of waste-gate (with degraded mass flow) increased resistance to pressure leads to significant increase in engine outlet pressure.  
c) Compressor Outlet Pressure - variation is occurring during transient phase. Degradation of mT leads to decrease in Turbine Power Pt, which in turn degrades compressor pressure.  
d) Turbine Outlet Pressure - variation occurring in boost pressure transient region, with the Turbine pressure decreasing due to closing of waste gate.  
e) Turbine Speed – speed degradation can be observed during transient state. Since mass flow degradation decreases Turbine power, leading to decrease in speed.  
f) Waste-gate – It closes more with turbine mass flow degradation in the boost pressure steady state region (owing to controller function).

8.2.2.2 Changing RPM with change in the boost pressure set-point

Graph 28 (a): Intake Manifold Pressure  
Graph 28 (b): Engine Outlet Pressure  
Graph 28 (c): Compressor Pressure  
Graph 28 (d): Turbine Pressure
Graph 28 Illustration of turbine mass flow \( m_T \) variation plots for changing RPM and changing boost pressure

a) Intake manifold pressure - variation (very small) is occurring during transient phase. Degradation of \( m_T \) leads to decrease in Turbine Power \( P_t \). As result turbo-speed gets degraded, which in turn affects manifold pressure, leading to decrease in pressure values.  
b) Engine outlet pressure variation – prominent variation in pressure value in the boost pressure steady state region. Due to closing of waste-gate (with degraded mass flow) increased resistance to pressure leads to significant increase in engine outlet pressure  
c) Compressor Outlet Pressure - variation (very small) is occurring during transient phase. Degradation of \( m_T \) leads to decrease in Turbine Power \( P_t \), which in turn degrades compressor pressure  
d) Turbine Outlet Pressure variation occurring in boost pressure transient region, with the Turbine pressure decreasing due to closing of waste gate.  
e) Turbine Speed – speed degradation (very small) can be observed during transient state. Since mass flow degradation decreases Turbine power, leading to decrease in speed  
f) Waste-gate – It closes more with turbine mass flow degradation in the boost pressure steady state region (owing to controller function).

8.2.3. DEGRADATION IN EFFICIENCY MAP (COMPRESSOR)

8.2.3.1 Constant RPM with change in the boost pressure set-point

Graph 29 (a): Intake Manifold Pressure  
Graph 29 (b): Engine Outlet Pressure
**Graph 29 Illustration compressor efficiency variation plots for constant RPM and changing boost pressure**

a) Intake manifold pressure - Variation is occurring at the end of transient region (very small variation). Degradation of efficiency leads to increase in Compressor Power $P_c$ (or work). As result turbo-speed gets degraded, which in turn affects manifold pressure, leading to a slight decrease in pressure values, in the boost region.  

b) Engine outlet pressure variation - Variation is very prominent (occurring during steady states) and engine outlet pressure increases with degradation of turbine efficiency, because waste-gate closes, thus providing more resistance to engine outlet pressure.  

c) Compressor Outlet Pressure - Degradation of compressor efficiency directly affects Turbo-speed, which in turn affects compressor pressure, leading to very slight decreases in pressure values, in the boost region.  

d) Turbine Outlet Pressure - Variation occurs in the boost pressure steady state region, with the pressure decreasing with degradation in compressor efficiency. Turbine pressure gets directly influenced with waste gate position.  

e) Turbine Speed – Prominent speed degradation can be observed during steady state region. Since efficiency degradation increase Compressor work, hence speed gets adversely affected  

f) Waste-gate – It closes more with compressor efficiency degradation in the steady state region (controller function)
8.2.3.2 Changing RPM with change in the boost pressure set-point

Graph 30 (a): Intake Manifold Pressure

Graph 30 (b): Engine Outlet Pressure

Graph 30 (c): Compressor Pressure

Graph 30 (d): Turbine Pressure

Graph 30(e): Turbo-Speed

Graph 30 (f): Waste-gate position
8.2.4. DEGRADATION IN EFFICIENCY MAP (TURBINE)

8.2.3.1 Constant RPM with change in the boost pressure set-point

Graph 31 (a): Intake Manifold Pressure

Graph 31 (b): Engine Outlet Pressure

Graph 31 (c): Compressor Pressure

Graph 31 (d): Turbine Pressure
Graph 31 Illustration of turbine efficiency variation plots for constant RPM and changing boost

a) Intake manifold pressure variation - Variation is occurring during transient phase (very small variation). Degradation of turbine efficiency directly affects Turbine power, speed and compressor pressure, which in turn affects manifold pressure because it is a function of compressor pressure, leading to a slight decrease in pressure values, in the boost region.  
b) Engine outlet pressure variation – variation in pressure value in the boost pressure steady state region. Due to closing of waste-gate (with degraded mass flow) increased resistance to pressure leads to significant increase in engine outlet pressure.  
c) Compressor Outlet Pressure - Degradation of turbine efficiency directly affects Turbine power and speed, which in turn affects compressor pressure, leading to very small decreases in pressure values, in the boost region.  
d) Turbine Outlet Pressure - Variation occurs in boost pressure steady state region, with the pressure decreasing with degradation in turbine efficiency. Turbine pressure gets directly influenced with waste gate position.  
e) Turbine Speed – Very small speed degradation can be observed during transient phase. Since efficiency degradation negatively affects Turbine power (Pt) and Turbine speed is a function of Pt.  
f) Waste-gate – Waste gate closes more with turbine efficiency degradation especially in the higher boost steady state region (controller function).

8.2.3.2 Changing RPM with change in the boost pressure set-point
Graph 32 (a): Intake Manifold Pressure

Graph 32 (b): Engine Outlet Pressure

Graph 32 (e): Turbo-Speed

Graph 32 (f): Waste-gate position

Graph 32  Illustration of turbine efficiency variation plots for changing RPM and changing boost

a) Intake manifold pressure variation - Variation is occurring during transient phase (very small variation). Degradation of turbine efficiency directly affects Turbine power, speed and compressor pressure, which in turn affects manifold pressure because it is a function of compressor pressure, leading to a slight decrease in pressure values, in the boost region.  
b) Engine outlet pressure variation – variation in pressure value in the boost pressure steady state region. Due to closing of waste-gate (with degraded mass flow) increased resistance to pressure leads to significant increase in engine outlet pressure.  
c) Compressor Outlet Pressure - Degradation of turbine efficiency directly affects Turbine power and speed, which in turn affects compressor pressure, leading to very small decreases in pressure values, in the boost region.  
d) Turbine Outlet Pressure - Variation occurs in boost pressure steady state region, with the pressure decreasing with degradation in turbine efficiency. Turbine pressure gets directly influenced with waste gate position and with waste-gate closing for efficiency degradation, Turbine pressure also decreases.  
e) Turbine Speed – Very small speed degradation can be observed during transient phase. Since efficiency degradation negatively affects Turbine power (Pt) and Turbine speed is a function of Pt.  
f) Waste-gate – Waste gate closes more with turbine efficiency degradation especially in the higher boost steady state region (controller function).
8.3. DEGRADATION SUMMARY – E-CHARGER

E-Charger sensitivity study follows a similar trend like the turbocharger. Manifold Pressure ($P_m$) and Compressor Pressure ($P_2$) depreciate in the transient phase, while Engine outlet pressure increases ($P_3$) and Turbine outlet pressure ($P_4$) decreases in the steady phase due to closing of the waste-gate. Waste-gate is closing in the steady state to negate the effect of flow rate and efficiency degradation. Turbo-speed also experiences deterioration but the region of deterioration differs in compressor and turbine, e.g. deterioration in speed takes place in the transient region during turbine flow and efficiency degradation while it happens in the steady state region for compressor flow and efficiency degradation.

Sensitivity study of the flow and efficiency parameter, help us in identifying the critical regions of e-Charger operation, where the system is susceptible to failure. In addition to this it also helps us in identifying severity of the failure. In general, most of the turbocharger related failures can attributed to the below mentioned reasons.

- Foreign object damage (FOD)
- Contaminated Lube oil
- Lack of lubrication (leading to sludge buildup)
- High exhaust temperatures

Most of these common reasons can be directly linked to the parameters related to changes of turbocharger efficiency and flow rate.

For example if we consider the degradation of turbine flow map, the probable cause of such an issue can be blockage of exhaust path, presence of foreign material in the exhaust stream or contaminated lube oil (which can lead to degradation of turbine blade). Now one of the probable outcome of such a condition can be decline in manifold pressure, which can lead to power loss while accelerating. Another peculiarity which can be observed in such a situation, is the prominent rise in engine outlet pressure due to closing of waste-gate (waste-gate is trying to negate the degradation effect), this increased exhaust pressure can cause problems with efficiencies, leakage of oil, carbonization of oil within the turbo and exhaust gas leaks from the turbo.

In certain plots, over speeding of turbocharger has also been observed (by over speeding the turbo is trying to compensate for mass flow degradation to maintain pressure ratio). It can induce cyclic stress and can cause failure of turbine blade.

Another case for turbocharger failure can be correlated with Compressor flow degradation. Flow degradation can be direct result of clogged air filter, presence of foreign object in the air path, which can lead to power loss, over heating of compressor and damage to compressor wheel. With the aid of the sensitivity study trend of different kinds of failure and their root cause can be identified. It can prove vital, while designing failure prediction models.
9. COMPARISON OF TURBOCHARGER AND E-CHARGER

The primary aim of the thesis was to analyze a conventional turbocharger system, to identify its drawbacks and to develop a new strategy to eliminate or minimize the shortcomings of the conventional turbocharger. The main motive of implementing the e-charger has been to pursue this goal. In the below-mentioned graphs, a vis-a-vis comparison has been made in order to detect the differences and the similarities between the performance of the turbocharger and the e-Charger under the same standard test conditions (Constant RPM with changing boost pressure point). In the graph 25, intake manifold outlet pressure of both systems has been compared. The improvement in the response time (thus minimizing turbo-lag) is visible in the case of e-Charger. When it comes to the turbo-speed comparison (graph 26), better response time and higher turbo-speed values can, again, be observed. Graph 27 the illustrates the waste position of the two systems – in case of e-Charger higher waste-position is visible, indicating the presence of excess boost during the steady state position. If this excess boost can be efficiently harvested, then the e-Charger performance can be further enhanced.

Graph 33  Intake Manifold Outlet Pressure Comparison between e-Charger and Turbocharger

Graph 34  Turbo speed and Waste-gat position comparison between Turbocharger & e-Charger
CONCLUSION

In the thesis, a turbocharger model for a gasoline engine has been developed on MATLAB Simulink using mean value modelling approach. Initial turbocharger data was provided by Honeywell, which was developed into a Turbo-map (compressor and turbine map) using MATLAB code. These maps formed the basis for the Simulink model. Additionally to a greater extent, the model has been influenced by Jensen and Kristensens method for mean value modeling. The turbocharger model has been validated under various simulated test conditions, to understand it’s characteristics and identify its limitations. The results of such maneuvers have been satisfactory, since the model matches the behavior of actual turbocharger up to a certain extent. Few limitations of the model have been identified, e.g. model is quite unstable at very high turbocharger rpm, the model shows unusual behavior if it goes beyond the scope of the turbocharger map, on which the model is based.

In the later part of the thesis a new model, with combination of turbocharger and e-charger has been proposed. In this model, e-charger is controlled by a Linear quadratic regulator, which feeds rpm as an input to the e-charger, whenever a requirement of a pressure boost arises. The new model has been able to minimize limitations of the previous model, like slow transient response and turbo lag. But it still can be considered a work in progress, since as observed by the waste-gate position comparison, there is a big portion of the engine outlet pressure which gets by-passed by the waste-gate. If this excess boost can be harvested to recharge the battery powering the e-charger, then the model can be completely self-sufficient. In the e-Charger model a system to monitor the battery charging values (kWh) has been included and with the implementation of an additional control strategy, the recharging functionality can be included in the existing model. The boost response of the e-charger can be further enhanced by the further calibration LQR controller parameters (e.g e-charger slope, waste-gate slope, charge pressure tracking and e-charger zero penalty) but they have not been intentionally tuned to achieve feasible e-Charger boost values.

One of the objective of the thesis was to conduct sensitivity study of the key parameters related to changes of turbocharger flow rate and efficiency. By degradation of flow rate and efficiency, overall effect on the performance and behavior of the turbocharger can be studied. Through this sensitivity study, the effort was to correlate the performance degradation of the model with actual field failures. Such correlation can be find wide range of application in prognostic approach. With the advent of autonomous driving, Automotive industry is trying to find the best solution for vehicle health management and prognostic approach meets these expectations. Once the trend of a failure is known, predictive models can be developed to look for similar trends, thus preventing failures before it can damage the turbo-machinery.
11. ANNEXURE

11.1. COMPRESSOR MAP MATLAB CODE

close all
clear all

% Data from Map

cmap = xlsread('Turb0.xlsx'); % Turbo-speed [rpm]
NC = cmap(:,1); % Turbo-speed [rpm]
WC = cmap(:,2); % Flow rate [Kg/sec]
PRC = cmap(:,3); % Pressure ratio
ETAC = cmap(:,4); % Compressor efficiency

k0 = [1 1 1 1 1]; % initial polynomial coefficient for pressure ratio
fhandle = @(k, X) compressor_PR(k, X);

X = [WC, NC];
k = lsqcurvefit(fhandle, k0, X, PRC); % curve fitting of the flow map
PRC_ = compressor_PR(k, X);

% pressure ratio vs mass flow plot

figure (1)
hold on % plot for measured values
plot(WC(1:7), PRC(1:7), '.', 'MarkerSize', 20, 'MarkerEdgeColor', [1 0 0])
plot(WC(8:14), PRC(8:14), '.', 'MarkerSize', 20, 'MarkerEdgeColor', [0 1 0])
plot(WC(15:21), PRC(15:21), '.', 'MarkerSize', 20, 'MarkerEdgeColor', [0 0 1])
plot(WC(22:28), PRC(22:28), '.', 'MarkerSize', 20, 'MarkerEdgeColor', [1 1 0] *0.8)
plot(WC(29:35), PRC(29:35), '.', 'MarkerSize', 20, 'MarkerEdgeColor', [0 1 1])
plot(WC(36:44), PRC(36:44), '.', 'MarkerSize', 20, 'MarkerEdgeColor', [1 0 1])

plot(WC(1:7), PRC_(1:7), 'Color', [1 0 0]) % plot for fitted values
plot(WC(8:14), PRC_(8:14), 'Color', [0 1 0])
plot(WC(15:21), PRC_(15:21), 'Color', [0 0 1])
plot(WC(22:28), PRC_(22:28), 'Color', [1 1 0] *0.8)
plot(WC(29:35), PRC_(29:35), 'Color', [0 1 1])
plot(WC(36:44), PRC_(36:44), 'Color', [1 0 1])

hold off % naming of y-axis
ylabel('Pressure ratio [-]')
xlabel('Mass flow [kg/s]') % naming of x-axis
grid on

a0 = [1 1 -1; 1 -0.1 1; 1 -1 0.1]; % initial polynomial coefficient for pressure ratio
X = [WC, NC];

fhandle = @(a, X) compressor_eta_new(a, X);
a = lsqcurvefit(fhandle, a0, X, ETAC); % curve fitting of the efficiency map
ETAC_ = compressor_eta_new(a, X);

figure (2) % efficiency vs mass flow plot
hold on % plot for measured values
% plot for fitted values
plot(WC(8:14), ETAC(8:14)*100, 'Color', [0 1 0])
plot(WC(15:21), ETAC(15:21)*100, 'Color', [0 0 1])
plot(WC(22:28), ETAC(22:28)*100, 'Color', [1 1 0] *0.8)
plot(WC(29:35), ETAC(29:35)*100, 'Color', [0 1 1]*1)
plot(WC(36:44), ETAC(36:44)*100, 'Color', [1 0 1])
hold off
ylabel('Efficiency [%]')
xlabel('Mass flow [kg/s]')
grid on

%% Functions for pressure ratio and efficiency calculations

function PR = compressor_PR(k, X) % pressure ratio function
    Tin = 298; % input temperature in kelvin
    Gm = 1.4; % gamma value of air
    Cp = 1005; % specific heat capacity [J/Kg K]
    d = 1.225; % density of air [kg/m3]
    R = 287; % gas constant [J/Kg K]
    Dc = 0.057; % compressor diameter [m]

    mA = X(:,1);
    Ntc = X(:,2);

    H = (Gm-1)/Gm;
    Uc = pi*Dc*Ntc/60;
    M = Uc/sqrt(Gm*R*Tin);
    K1 = k(1,1) + k(1,2)*M;
    K2 = k(2,1) + k(2,2)*M;
    K3 = k(3,1) + k(3,2)*M;
    Phi = mA/(d * (pi/4) * (Dc^2) .* Uc);
    Psi = (K1 + K2.*Phi)./(K3 - Phi);
    PR = ((Psi.*0.5.*Uc.*Uc)./(Cp*Tin) + 1).^1/(1/H);
end

function eta = compressor_eta_new(a, X) % efficiency function
    Tin = 298; % input temperature in kelvin
    Gm = 1.4; % gamma value
    d = 1.225; % density of air [kg/m3]
    R = 287; % gas constant [J/Kg K]
    Dc = 0.057; % compressor diameter [m]

    mA = X(:,1);
    Ntc = X(:,2);

    Uc = pi*Dc*Ntc/60;
    M = Uc/sqrt(Gm*R*Tin);
    Phi = mA/(d * (pi/4) * (Dc^2) .* Uc);
    eta = a(1,1).*Phi.^2 + a(1,2).*Phi + a(1,3)*M +a(2,1);
end
11.2. TURBINE MAP MATLAB CODE

```matlab
clear all
clc

% Data from Map
PR_orig = xlsread('TurboMaps.xlsx');
Mf = xlsread('TurboM.xlsx');
Nt = xlsread('TurboM.xlsx');
mETA = xlsread('TurboM.xlsx');
PR= PR_orig;

k0 = [9.979e-06; -1.494; -5.5488e-06; -0.1037];

fhandle = @(k,PR)turbine_m(k,PR,Nt);
k = lsqcurvefit(fhandle,k0,PR,Mf);
mfit = turbine_m(k,PR_orig,Nt);
figure(13)
plot(PR_orig,Mf,'ob'), hold on
plot(PR_orig,mfit,'.r','MarkerSize', 20), hold off
ylabel('Mass flow [kg/s]')
xlabel('Pressure ratio [-]')
legend('EXP','FIT')
grid on

% ETA FIT
a0 = [0.6 0; 2 -0.00001; 4.2 -0.35];

fhandle = @(a,PR)turbine_eta(a,PR,Nt);
a = lsqcurvefit(fhandle,a0,PR,mETA);
ETAfit = turbine_eta(a,PR,Nt);

% plot for Turbine measured efficiency vs PR
figure(14)
hold on
plot(PR(1:7), mETA(1:7)*100, '.', 'MarkerSize', 20, 'MarkerEdgeColor', [1 0 0])
plot(PR(8:14), mETA(8:14)*100, '.', 'MarkerSize', 20, 'MarkerEdgeColor', [0 1 0])
plot(PR(15:21), mETA(15:21)*100, '.', 'MarkerSize', 20, 'MarkerEdgeColor', [0 0 1])
plot(PR(22:28), mETA(22:28)*100, '.', 'MarkerSize', 20, 'MarkerEdgeColor', [1 1 0])
plot(PR(29:35), mETA(29:35)*100, '.', 'MarkerSize', 20, 'MarkerEdgeColor', [0 1 1])
plot(PR(36:44), mETA(36:44)*100, '.', 'MarkerSize', 20, 'MarkerEdgeColor', [1 0 1])
hold off
ylabel('Efficiency [%]')
```

xlabel('Pressure ratio [-]') ;  grid on;

%%% Functions for flow rate and efficiency calculations

% Turbine mass flow rate function

function m = turbine_m(k,PR,Nt)

Tin =1100;  
Gm = 1.32;  
Cp = 1005;  
d = 1.225;  
R = 287;  
Dt = 0.051;  
A1 = k(1);  
A2 = k(2);  
A3 = k(3);  
A4 = k(4);

Kp = exp((A1*Nt) + A2);  
At = exp((A3*Nt) + A4);

m = At. * sqrt(1-PR.^( -Kp));
% mass flow rate turbine

end

% Turbine efficiency function

function eta = turbine_eta(a,PR,Nt)

Tin =1000;  
Gm = 1.32;  
Cp = 1005;  
Dt = 0.051;  
Nsp = Nt/sqrt(Tin);

H = (1-Gm)/Gm;

U1 = pi*Dt.*Nt/60;  
U2 = ((2*Cp.*Tin.*((1-(PR).^H))).^0.5));  
Uc = U1/U2;  

eta = a(1,1) + a(1,2) * Nsp + (a(2,1) + a(2,2). * Nsp). * Uc  + (a(3,1) + a(3,2). * Nsp). * (Uc.^2);

end
11.3. SIMULINK MODEL TURBOCHARGER

Fig. 25 Simulink model of turbocharger
11.4. SIMULINK MODEL OF E-CHARGER

Fig. 26 Simulink model of e-charger
### 11.5. PRESSURE RATIO VS TIME PLOT (E-CHARGER)

**Graph 35** Pressure ratio vs time plot for constant RPM with upward pressure boost ramp

**Graph 36** Pressure ratio vs time plot for constant RPM with upward and downward boost ramp
11.6. BATTERY CHARGING STATUS (E-CHARGER)

Graph 37 Battery charge status at constant RPM with upward boost pressure

Graph 38 Battery charge status at constant RPM with upward and downward boost pressure
12. REFERENCE


[12] Prof Ing. Jan Macek, Modeling of transient modes of vehicle motors - Turbocharger dynamics, Josef Bozek Research Centre - CVUT v Praze


