Presently in the vehicle industry full engine system simulations are performed using different one-dimensional software programs in order to assess the effect of different geometrical and part changes on the system as a whole. These simulations are usually fast and multiple parameters can be monitored and analysed.

In this report GT-Power simulations have been performed on a complete engine designed by Volvo Car Corporation. The investigation was performed in order to gain basic knowledge about the internal combustion engine and specifically about the gas exchange system and the turbocharger. A parameter study was performed and the responses on the turbine efficiency and break torque were analysed.

The trends in the simulation results follow the background theory well, i.e. increasing the turbine efficiency increases the engine efficiency and reduces the time to torque. The effect of the valve opening times and durations on the break torque and the turbine efficiency can be studied. There is an intricate relationship where the optimal configuration is dependent on the engine speed as well as the opening angles and times.

GT-Power is a very powerful tool for simulating complete engines. However care must be taken when analysing the results. The code only uses one direction and time, meaning that the flow will always be uniform in the cross sections. Whereas in many parts of the real engine the flow field is three dimensional and far from uniform in the cross-sections.

1. Introduction

During the upstart part of this PhD project a short introductory study was performed with Gamma Technologies GT-Power. This study was performed in order to gain basic understanding of the turbocharger’s turbine and its role in the internal combustion engine. General knowledge was also gained on GT-Powers strengths and limitations.

GT-Power is a one dimensional flow solver specifically tailored to simulate internal combustion engine flows, accounting for cylinder motion, combustion,
gas composition, and temperatures among others. GT-Power calculates the flow motion in time using many different models for all parts of the engine. The models are mainly based on experimental empiricism and curve fitting from tabulated data.

GT-Power has been used to study a variety of different cases. In Sellnau & Rask (2003) and Lancefield (2003) variable valves are investigated and their effect on engine performance is evaluated. Reifarth & Ångström (2010) performed GT-Power simulations of EGR systems to assess their strengths and weaknesses. Park et al. (2010) performed simulations of a light duty diesel engine and Shi & Feng (2010) were performing optimizations for a gasoline engine intake system. More advanced models are used when investigating chemistry of different fuels such as HCCI (Etheridge et al. (2009), Etheridge et al. (2009)). Even acoustics can be modelled and mufflers can be designed and investigated in GT-Power (Shaohua et al. (2003)),

GT-Power can be coupled to a large number of external full 3D computational fluid dynamics solvers, e.g. openFOAM, StarCCM+, and Ansys CFX. This is done in order to extend the validity of the results, to increase the accuracy, and also study certain components in more detail with a realistic engine system surrounding it.

The study was performed using GT-Power to visualize how different parameters affected parameters such as turbine efficiency, and break torque. Simulations were performed both at steady state conditions and during transients. The transient simulations were performed in order to study the effect the parameter variations had on the time to torque.

2. Method

GT-Power is a software developed by Gamma Technologies\(^1\) that can simulate an entire internal combustion engine. The software is used by almost all major car and truck companies including Volvo Car Corporation. The software can simulate all the parts of the engine and also be coupled with external software programs to study specific parts.

The programs main parts are the different pipes and flow splits that are used to build up the geometry. For the more specialized parts (e.g. cylinders, turbochargers, after-treatment devices etcetera) the program uses models and tables to calculate the values needed (e.g. pressure, heat release, mass flow, efficiency, etcetera).

For the transient calculations the time to torque value is being investigated and especially how the different parameter changes affect it. The time to torque is defined as the time it takes for the simulations to go from 45 Nm to 300 Nm brake torque.

\(^1\)www.gtisoft.com
2.1. Governing Equations

The governing equations for the GT-SUITE software are as follows; Continuity (1), Momentum (2) and Energy (3) (Gamma Technologies (2009)).

\[ \frac{dm}{dt} = \sum_{\text{boundaries}} \dot{m} \]  

\[ \frac{d\dot{m}}{dt} = \frac{dpA + \sum_{\text{boundaries}} (\dot{m}u) - 4C_f \frac{\rho u|u|}{2} dxA - C_p \left( \frac{1}{2} u|u| \right) A}{dx} \]  

\[ \frac{d(me)}{dt} = p \frac{dV}{dt} + \sum_{\text{boundaries}} (\dot{m}H) - hA_s(T_{\text{fluid}} - T_{\text{wall}}) \]  

With \( m \) is the mass, \( t \) is the time, \( \dot{m} \) is the mass flux, \( dp \) is the pressure differential across \( dx \), \( A \) is the area, \( u \) is the velocity at the boundary, \( C_f \) is the skin friction coefficient, \( \rho \) is the density, \( dx \) is the discretization length, \( D \) is the equivalent diameter, \( C_p \) is the pressure loss coefficient, \( e \) is the internal + kinetic energy, \( p \) is the pressure, \( V \) is the volume, \( H \) is the total enthalpy (\( H = e + \frac{1}{2} p \)), \( h \) is the heat transfer coefficient, \( A_s \) is the heat transfer surface area, \( T_{\text{fluid}} \) is the fluid temperature, \( T_{\text{wall}} \) is the wall temperature.

3. Case set-ups

The tested parameter space can be seen in table 1 for the steady state cases and in table 2 for the transient cases.

For the steady state cases 13 different engine speeds were used (1000 - 6000 RPM). Between 3 and 18 different simulations were performed per parameter and engine speed, resulting in approximately 900 different steady state cases.

For the transient cases 3 different engine speeds were chosen, 1250, 1500 and 1750 RPM. The transient cases also had fewer tested parameters, the steady state simulations showed no effect on the tested variables and were then removed from the test matrix. In total approximately 140 simulations were performed for the transient cases.
Table 1. The parameters tested in the steady state simulations. For the heat conduction the value was increased by 7 from 7 to 70, and then by 70 up until 700. VVT is the variable valve timing.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Min</th>
<th>Max</th>
<th>Increment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Exhaust manifold angle</td>
<td>0°</td>
<td>45°</td>
<td>15°</td>
</tr>
<tr>
<td>Blade speed ratio</td>
<td>1.4</td>
<td>2.2</td>
<td>0.1</td>
</tr>
<tr>
<td>Heat conduction</td>
<td>7</td>
<td>700</td>
<td>7, 70</td>
</tr>
<tr>
<td>Lift duration intake</td>
<td>80%</td>
<td>120%</td>
<td>5%</td>
</tr>
<tr>
<td>Lift duration exhaust</td>
<td>80%</td>
<td>120%</td>
<td>5%</td>
</tr>
<tr>
<td>Turbine efficiency multi</td>
<td>80%</td>
<td>120%</td>
<td>5%</td>
</tr>
<tr>
<td>VVT exhaust</td>
<td>-20</td>
<td>20</td>
<td>$\frac{20}{T}$</td>
</tr>
<tr>
<td>VVT inlet</td>
<td>5</td>
<td>55</td>
<td>12.5</td>
</tr>
</tbody>
</table>

Table 2. The parameters tested in the transient simulations.

<table>
<thead>
<tr>
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<th>Increment</th>
</tr>
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<tbody>
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<td>0.1</td>
</tr>
<tr>
<td>Heat conduction</td>
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<td>1200</td>
<td>100</td>
</tr>
<tr>
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<td>80%</td>
<td>120%</td>
<td>5%</td>
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<td>12.5</td>
</tr>
</tbody>
</table>

The exhaust manifold angle is the angle at which the exhaust pipes from the cylinder are entering the turbine volute.

The blade speed ratio (BSR) is defined according to equation (4), with $U$ and $C_s$ defined according to equations (5) and (6).

\[
BSR = \frac{U}{C_s} \tag{4}
\]

\[
U = \frac{2\pi N}{60} \left( \frac{D}{2} \right) \tag{5}
\]

\[
C_s = \left[ 2C_p T_{in} \left( 1 - PR^{\frac{\gamma - 1}{\gamma}} \right) \right]^{1/2} \tag{6}
\]
With $N$ being the turbine speed in RPM, $D$ is the turbine diameter, $C_p$ is the specific heat at constant temperature, $PR$ is the pressure ratio across the turbine and $\gamma$ is the ratio of specific heats.

The heat conduction parameter changes the external convection coefficient value of the exhaust manifold, due to small effects for the steady cases the parameter was largely increased for the transient cases.

The turbine efficiency multiplier changes the turbine efficiency by multiplying it with the specified value. The unchanged efficiency value is found in the turbine map supplied to the model, see figure 1.

![Figure 1. Typical turbomap from the manufacturer. The colors show the efficiency in %.

3.1. Variable Valves

For the cases when the valve lift duration is changed the valve lift curves can be seen in figure 2. The variations in valve opening angles (VVT) is shown in figure 3.

3.2. Response Parameters

In all simulations the tested parameters effect on turbine efficiency and break torque was investigated. The turbine efficiency is read from the table corresponding to figure 1 and it is dependent on the pressure ratio and mass flow through the turbine. The break torque is calculated from the engine simulation data.
Figure 2. For the valve lift duration: $-$ is the original exhaust valve lift curve, $-$ is the 20\% increase curve, and $-$ is the 20\% decrease lift curve. $-$ is the original intake valve lift curve, $-$ is the 20\% increase curve, and $-$ is the 20\% decrease lift curve.

Figure 3. For the valve timing: $-$ is the original exhaust valve lift curve, $-$ is the 20 CAD earlier curve, and $-$ is the 20 CAD later lift curve. $-$ is the original intake valve lift curve, $-$ is the 5 CAD delayed curve, and $-$ is the 55 CAD delayed lift curve.
4. Results - Steady State

After the steady state results had been analysed the exhaust manifold angle was found to have a very low effect on the tested parameter responses and were then chosen not to be further tested in transient runs.

4.1. Exhaust Manifold Angle

When changing the exhaust manifold angle no significant change in the results are found (see figure 4). The plot lines are virtually placed on top of each other, except for the 5000 RPM case where a dip is seen at the end of the runs, but all cases still end up at the same end-value. The dip is coming from when one of the monitored variables is reaching a maximum or minimum limit set in the model. The variable value is then changed and the simulation converges.

Figure 4. The effect of the exhaust manifold angle on the turbine efficiency parameter. The "dip" for the 5000 RPM simulations can be seen.
4.2. Blade Speed Ratio

For the blade speed ratio parameter the plots show similar results for all engine speeds except for 5000 RPM case. As can be seen in figure 6 the break torque reaches a maximum limit and is then reduced by the software control. The effect of this can also be seen in figure 5. The blade speed ratio parameter’s largest effect is on the turbine efficiency results, which is expected (see figure 5) according to equation 4.

![Blade Speed Ratio plots](image)

(a) 6000 RPM.  
(b) 1500 RPM.  
(c) 5000 RPM.

**Figure 5.** The effect of the blade speed ratio on the turbine efficiency parameter. The "dip" for the 5000 RPM simulations can be seen.
Figure 6. The effect of the blade speed ratio on the break torque parameter.
4.3. Heat Conduction

When changing the external convection coefficient for the heat conduction object of the exhaust manifold parts no effects are observed to any of the monitored parameters (see figure 7).

![Figure 7. The effect of the external convection coefficient on the turbine efficiency parameter.](image)
4.4. Intake Lift Duration

During these simulations the lift end position (in crank angles) was kept constant and only the lift duration was changed. This means that the lift starting time is changed, in order to maintain the lift curve and allow for changes in duration. Results for turbine efficiency and break torque can be seen in figures 8 and 9.

For shorter than normal durations an increase in the turbine efficiency can be seen and for longer durations only very small changes can be seen for high engine speeds. For low engine speeds both shorter and longer lift durations gives an increase to the turbine efficiency except for very short lift durations when the engine does not get enough air. During short lifts the mass of air into the cylinder is reduced and depending on where in the turbine map you are the efficiency is changed. For long lifts the mass of air is increased and a higher mass flow and pressure ratio is achieved, meaning that the efficiency is increased.

![Image](a) 6000 RPM.  
![Image](b) 1500 RPM.  
![Image](c) 5000 RPM.

**Figure 8.** The effect of intake lift duration on the turbine efficiency parameter.

The intake lift duration only has a small or negligible effect on the break torque, except for certain engine speeds. For the 6000 RPM case very small changes are noticed and for the 1500 RPM case similar small effects can be observed, except for the shortest duration when the engine does not get enough air. For the 5000 RPM simulations the two cases with the shortest lift duration
Johan Fjällman

do not get enough air into the cylinder. This causes the break torque to reach a minimum limit and the simulation stops.

**Figure 9.** The effect of the intake lift duration on the break torque parameter.
4.5. Exhaust Lift Duration

When the exhaust lift duration was changed the exhaust valve opening time was kept constant and only the closing time was varied, keeping the lift profile constant. With a constant lift profile and varying lift duration the maximum lift decreases and a significant flow constriction is added.

For low RPMs and long lift durations the turbine efficiency is drastically lowered, the difference is up to 11% lower than the baseline case. This is most likely caused by back flow from the other cylinders which will lower the pressure ratio and mass flow.

For high RPMs the turbine efficiency is almost constant except for the short duration lifts. This is caused by an increase in pressure ratio due to the valve closing just as the blow down pulse ends.

When considering the exhaust lift duration effect on the break torque similar tendencies can be seen. For the 6000 RPM case the break torque is kept constant except for the short duration cases. Same for the low RPMs, constant break torque for lower durations and then as the lift duration increases the break torque is lowered.

For high RPMs and short lift durations the gases in the cylinders doesn’t have time to be expelled and because of that the break torque is lowered.
Figure 11. The effect of the exhaust lift duration on the break torque parameter.
4.6. **Turbine Efficiency Multiplier**

Changing the turbine efficiency multiplier from 80% to 120% in 5% steps changed the turbine efficiency by that value.

![Diagram](image)

**Figure 12.** The effect of turbine efficiency multiplier on the turbine efficiency parameter.

For high RPMs the break torque converges to the same value for all turbine efficiencies. This is the same for low RPMs as well, unless the efficiency is too low and lowers the engine power output.
4.7. Variable Valve Timing

During the VVT simulations the lift duration was kept constant and the valve opening angle was changed instead. This was done in order to change both the valve overlap (intake and exhaust valves) and the cylinder overlap (exhaust valves between cylinders).

It can generally be seen that a later exhaust valve opening (EVO) angle gives a higher turbine efficiency (see figure 14) and that a later intake valve opening angle also increases the turbine efficiency. Opening the exhaust valve later gives the cylinder more time to build up pressure and as such a higher pressure ratio is achieved, which increased turbine efficiency. Opening the intake valve later reduces the valve overlap in the cylinders which increases efficiency.

The low RPM cases showed the same trends for the break torque as they did for the turbine efficiency. An early intake valve opening angle results in a higher break torque. For the high RPM cases the trends are opposite. A late exhaust valve opening angle lowers the break torque.
Figure 14. The figures show the effect of the varied valve timing on the turbine efficiency parameter.

Figure 15. In the figures the effect of the varied valve timing on the break torque is shown.
5. Results - Transient

The transient cases are evaluated with respect to how the different parameters change the time to torque (i.e. the time it takes to reach 300 Nm (in this case)). Three low engine speeds were chosen (1250, 1500, and 1750 RPM) as this is where the time to torque is important (during accelerations and overtakes).

All the graphs display an initial slope where the suction engine effect is coming in, this happens before the turbocharger spins up and starts providing the increased demand for airflow.

5.1. Blade Speed Ratio

Only small effects are seen when changing the blade speed ratio of the turbine on the engines time to torque curves. For the 1250 RPM case the engine does not reach the 300 Nm target in the 2 second time frame that is given but it is shown that the higher the blade speed ratio the higher the torque is at a certain time. For the two higher engine speeds almost all cases reach the 300 Nm target within the time frame and the fastest time is 1.046 s for the 1750 RPM case with the two highest blade speed ratios.

\[ \text{Figure 16. The effect of the blade speed ratio on the time to torque for three transient cases.} \]
5.2. Heat Conduction Object

The effect of the heat conduction object on the time to torque curves can be seen in figure 17. The effect of changing the parameter is small on low engine speeds and the effect increases with increasing engine speed. For the lowest engine speed no case reached the target torque value and for the 1500 RPM case only the two lowest values for the heat conduction object failed to reach the target torque. The fastest case was the 1750 RPM case with the highest value for the heat conduction object, 1.114 s. With a higher value for the external heat conduction the reduced mass flow is changed and because of that the turbine efficiency is increased and a higher break torque is obtained.

![Figure 17](image-url)

**Figure 17.** The effect of the heat conduction object on the time to torque for three transient cases.
5.3. Exhaust Valve Lift Duration

Changing the exhaust valve lift duration changes the time to torque drastically after a certain value. The trend is similar for all three engine speeds, a too long lift duration increases the time to torque. The higher the engine speed the more likely it is that the engine will reach the targeted torque value in the given time frame. The fastest cases were the 1750 RPM cases with a lift duration close to the original one (80 - 95% of the original time), 1.114 s.

![Figure 18](image-url)

**Figure 18.** The effect of the exhaust valve lift duration on the time to torque for three transient cases.
5.4. Intake Valve Lift Duration

The intake valve lift duration affects the time to torque differently than the exhaust lift duration does. For the three simulated engine speeds the lift duration needs to be close to the original one for the shortest time to torque. For the 1500 RPM case the optimal lift duration was the original one, changing the duration by more than 5% caused the engine not to reach the torque value within the given time. For the 1750 RPM case the optimal lift duration was between the original one and a 5% increase but also here the time increased if the duration was changed too much from the original. Optimal in this case is the lift duration that would give the shortest time to torque. The two fastest cases were the 1750 RPM ones with a lift duration of 100% and 105%, 1.183 s.

Figure 19. The intake valve lift duration effect on the time to torque for three transient cases.
5.5. **Turbine Efficiency Multiplier**

Increasing the turbine efficiency reduces the engines time to torque, the higher the efficiency the faster the engine reached the torque value. This was an expected result and the software behaved as the theory dictated. The fastest case was the highest engine speed (1750 RPM) with the highest turbine efficiency, 0.909 s.

![Diagram of turbine efficiency multiplier effect on time to torque for three transient cases.](image)

**Figure 20.** The turbine efficiency multiplier effect on the time to torque for three transient cases.
5.6. Variable Valve Time

All three engine speeds behave the same when changing the valve overlap. A late exhaust valve opening and an early intake valve opening gives the shortest time to torque. It can be seen in all plots that delaying the intake valve opening gives a longer time to torque than keeping the original values (lighter color shows later intake valve opening angle (see figure 21)). Delaying the intake valve angle by more than 30 CA reverses the trend, i.e. a later exhaust opening then reduces the time to torque. The fastest time to torque was the 1750 RPM case with an 20 CA increase in intake valve opening and 5 CA increase in exhaust valve opening, 1.114 s.

![Diagram](image)

**Figure 21.** The valve lift time variation effects on the time to torque for three transient cases.
6. Discussion and Conclusions

For almost all the cases the trends are clear and they are easy to understand. For example an increased turbine efficiency will increase the turbine performance and the total engine performance (i.e break torque), a change in the blade speed ratio parameter will only affect the turbine performance and not the overall engine performance in steady state.

When interpreting the effects of the variations in the valve lift starting time the results are more complex. Not only is the value of the intake lift time important but also when the exhaust valve opens and closes. For low engine speeds and steady state simulations a very late (> 30 CA) intake valve opening decreases the torque significantly. However when the intake valve opening is not delayed more than 30 CA the exhaust valve opening seems to have little to no effect on the break torque. For high engine speeds an exhaust valve opening angle that is earlier or close to the original one is shown to be the most efficient and then the intake opening angle is shown to have a very small effect on the break torque.

7. Acknowledgements

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References


