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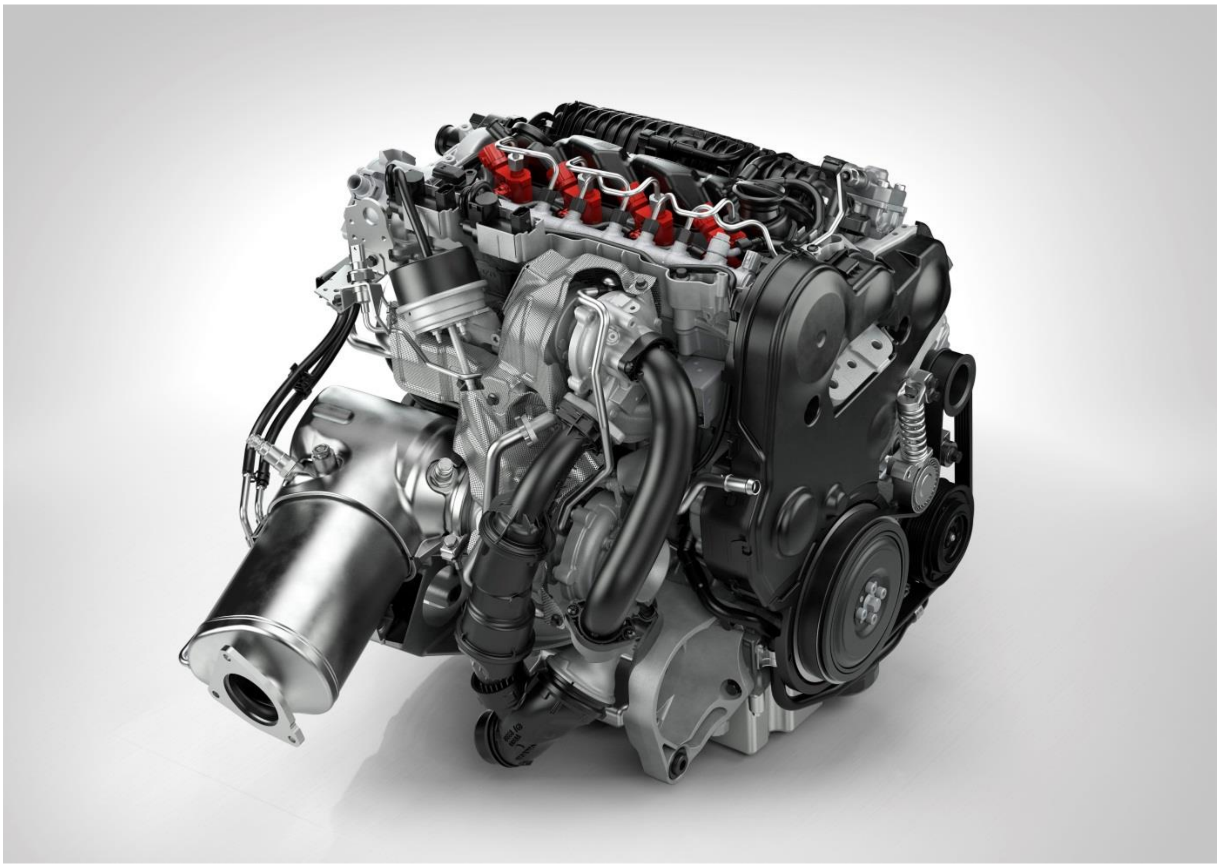

Analysis of Engine Cold Start Simulation in GT-Power

Master's Thesis in Automotive Engineering

EMMA JOHANSSON

SOFIA WAGNBORG

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ISSN 1652-8557
Department of Applied Mechanics
Division of Combustion
Chalmers University of Technology
SE-412 96 Göteborg
Sweden
Telephone: + 46 (0)31-772 1000

Cover:
The VCC 4-cylinder diesel engine

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ABSTRACT

More stringent regulations on emissions in the automotive industry increase the need of improving all areas of the engine operation. An area where the emission formation is problematic, but has not before been prioritized regarding computer simulations, is the engine start-up at cold conditions. A simulation model is an effective tool in order to study the cold start behaviour and emission formation dependent on engine characteristic parameters, without extensive assets used.

This report aims at analysing engine cold start behaviour, the critical areas and the requirements needed to implement a simulation model in GT-Power, for both gasoline and diesel engines. Two models were created, one for a gasoline engine with manual transmission and one for a diesel engine with automatic transmission, which aimed to simulate engine start-up at 20°C and -30°C. Test-rig data was used to validate the results. Simplifications for different systems were however needed in order to create the start-up models.

It was found that it was not possible to simulate the start-up at -30°C with the simplifications made, due to the extensive amount of friction at this temperature. The results show however that it was possible to create simulation models for 20°C, which correlate with test-rig data. The models can be used to get a general understanding of the start-up, but the use of the models is limited due to the necessary simplifications.

The start-up process is complex and even though the results show good correlation between the models and test runs, the subsystem needs to be considered in detail in order to create valid simulation models. The accuracy of the models are strongly dependent on how the starter motor, friction and inertia is modelled, and small deviations can give consequences leading to an invalid model. If the models are further developed and adjusted with proposed changes, they could be used to predict the behavior of different start-up events and to improve the cold start and reduce emissions.

Key words: Cold start simulation, Engine cold start, Gasoline engine, Diesel engine, GT-Power, Start-up modelling, 1D-Simulation.

Contents

Preface

In this study, an analysis of engine cold start behavior was conducted in order to create simulation models for a gasoline and a diesel engine. The work has been carried out from January to June 2014 at Volvo Car Corporation (VCC). We would like to thank VCC for enabling this project and the involved VCC employees, who helped us and provided necessary facts and data.

We would like to thank our examiner Sven Andersson, Associate Professor at the Department of Applied Mechanics, Combustion division, for his support and recommendations during this project and for the great help during our years as Master's students.

Further, we would specially like to thank our supervisor David Willermark for his genuine interest and for the great guidance and support throughout the project. We would also like to thank Stefan Bohatsch for his recommendations and the employers at Engine CAE department for their help and guidance.

Finally, we would like to thank our families and friends for their support during our education at Chalmers University of Technology.

Göteborg June 2014

Emma Johansson and Sofia Wagnborg

Notations

BDC	Bottom Dead Center
BMEP	Brake Mean Effective Pressure
CAD	Crank Angle Degree
CI	Compression Ignition
CO	Carbon Oxide
DC	Direct Current
DI	Direct Injection
DMF	Dual Mass Flywheel
EVC	Exhaust Valve Closing
EVO	Exhaust Valve Opening
FKFS	Forschungsinstitut für Kraftfahrwesen und Fahrzeugmotoren Stuttgart
FMEP	Friction Mean Effective Pressure
GDI	Gasoline Direct Injection
HC	Hydrocarbon
HEV	Hybrid Electric Vehicle
IMEP	Indicated Mean Effective Pressure
IVC	Intake Valve Closing
IVO	Intake Valve Opening
NO_x	Nitrogen Oxides
PFI	Port Fuel Injection
PM	Particle Matter
PMDC	Permanent Magnet Direct Current
PMEP	Pumping Mean Effective Pressure
RPM	Revolutions per Minute
SI	Spark Ignition
SMEP	Starter Mean Effective Pressure
SOI	Start of Injection
TDC	Top Dead Center
UHC	Unburned Hydrocarbon
VCC	Volvo Car Corporation
VVT	Variable Valve Timing
QDM	Quasi-Dimensional Model

1 Introduction

This report investigates the characteristics of engine start-up and what needs to be considered in order to create a simulation model in GT-Power. The start-up event for a direct injected gasoline and a diesel engine is studied.

1.1 Background

More stringent restrictions for fuel consumption and emissions for automotive vehicles compel the need of improve every step of engine operation. One phase that is known to be inefficient in terms of fuel consumption is the engine start-up phase. The start-up also accounts for a large part of the total pollutant emissions like hydrocarbon (HC) emissions for gasoline engines and particle matter (PM), carbon oxide (CO) and white smoke for diesel engines. Although there have been improvement in this area due to the three-way catalyst and the oxidation catalyst, further reduction of emissions is needed. Significant for start-up is the pollutants emitted before the catalysts reach the light-off temperature, the temperature at which the catalyst is considered working properly.

Computer simulations in this area have not been prioritized but could be essential for understanding and improving the behavior of engine start-up in terms of emissions and efficiency. New technologies, such as the stop-start system and hybrid electric vehicles, further increases the demand of investigating the effect of the start-up process on the engine components. Furthermore, real performance tests are expensive and if a simulation model could provide useful results, real tests for cold start could partially be substituted.

Even though there is an existing start-up simulation model in the engine simulation software GT-Power, this has not yet been implemented and validated in the engine models at Volvo Car Corporation (VCC). To create a valid simulation model which could be used as an effective tool, the cold start characteristics and model requirements needs to be investigated and clarified.

1.2 Aim

The aim of this project is to create an understanding of the engine start-up process to distinguish the critical areas as well as the requirements in order to simulate the cold start behavior in GT-Power. Furthermore, the aim is to create and evaluate two models in GT-Power, one for gasoline engines and one for diesel engines.

1.3 Objective

- Understand the functionality and critical areas as well as influencing systems of engine start-up.
- Investigate the requirements in order to simulate cold start behavior in GT-Power.
- Create a simulation model for engine start-up for both a gasoline and a diesel engine.

- Simulate the behavior of engine cold start for two temperatures, 20° and -30°C, and verify with data from real performance tests.

1.4 Limitations

Only the accuracy of the simulation model will be considered and no effort will be made to improve the engines during cold start, nor investigate emissions. One DI gasoline engine and one diesel engine will be examined during the investigation.

Only 1D gas exchange calculations will be considered and no structural or fluid mechanical calculations will be included. No measurements will be performed during the project; the data needed for validation will be collected from already performed tests.

1.5 Scope

Two GT-Power models are provided as a basis to the project from VCC. The models are based on a turbocharged 4-cylinder gasoline engine and a turbocharged 4-cylinder diesel engine. The two models will be modified in order to simulate cold start behavior for the two engines.

2 Theory

This section provides theoretical background to the engine start-up. The different systems and technologies influencing the start-up process is presented and described in order to give a general understanding. Furthermore the important aspects to be considered in order to create a model are presented.

2.1 General start-up characteristics

The start-up process of an engine can be divided into three stages; engine cranking, engine run-up and the transition to idle. During engine cranking the starter motor engages and transmits torque through a large gear ratio, about 10 times the starter torque, to the crankshaft. The starter motor is used to overcome the inertia and friction at stand still, and will disengage at the engine speed when the power created by the combustion is high enough to rotate the crankshaft independently. When the indicated work from the piston is large enough to overcome all losses, the engine starts to accelerate toward idle speed and enters the run-up phase. The engine accelerates to above idle speed and then transitions down to idle. During the idle phase the engine speed stabilizes at the idle speed (Lejsek & Kulzer, 2009).

In Figure 2.1 the different phases is shown for a four-cylinder gasoline engine. In Figure 2.1 the cylinder pressure is also displayed. During cranking, there is low cylinder pressure since no combustion occurs and the engine only pumps air. When the engine starts to run-up the combustion has started and the cylinder pressure increases. At the idle phase the ignition is retarded in order to increase the exhaust temperature to heat the catalyst and to stabilize the engine speed at idle, leading to the lower cylinder pressure seen in the figure. The figure also shows the portion of HC emissions for the start-up event. The figure is representative for the diesel engine as well, apart from the HC emission curve.

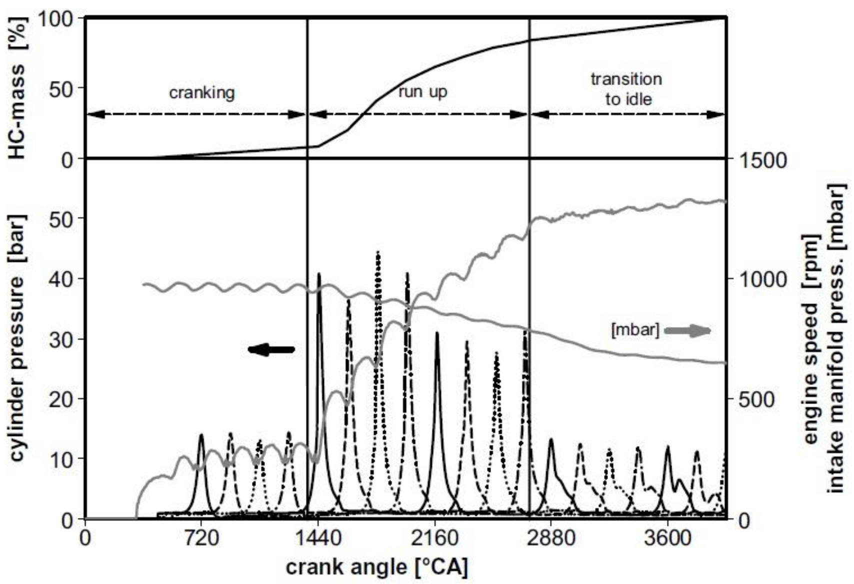

Figure 2.1 Start-up of a four-cylinder gasoline engine (Lejsek & Kulzer, 2009).

In Figure 2.1 it could also be seen that the initial idle speed is rather high at cold starts. In order to warm up the catalysts as fast as possible the engine speed is increased to about 300 rpm higher than the idle speed at normal operation until the catalysts is considered warm. The duration at this phase and how much the engine speed is increased depends on the start temperature.

Engine start-up operation is different from normal operation due to the highly transient behavior of various parameters, such as engine speed, in-cylinder pressure and intake manifold pressure. The different parameters are dependent on each other and therefore the behavior of every parameter is important to analyze. Another significant difference is the temperature at which the engine operates, during cold start the temperature is lower and it takes time for the engine to warm up. The lower in-cylinder temperature is unfavorable for the fuel atomization and fuel vaporization, which can lead to poor mixture preparation, incomplete combustion and misfire. The ambient temperature is also important since it affects the temperature of the oil in the engine. Low oil temperature increases lubricant viscosity and thereby friction losses. The contribution to the torque from friction could be up to five times more when the engine is cold compared to warm conditions (Rakopoulos & Giakoumis, 2009)

A factor influencing the in-cylinder temperature and pressure is cranking speed. Higher speed gives increased temperature and pressure, which could reduce the extended ignition delay and the minimum starting temperature. The compression ratio also affects the in-cylinder conditions; higher compression ratio gives an increase in temperature and pressure (Rakopoulos & Giakoumis, 2009).

The torque demanded by the engine during the start-up is firstly dependent on the friction, which varies with temperature, and the inertia of the rotating parts. But it is also dependent on the position of the pistons. The torque demand is the highest when the piston approaches top dead center (TDC) in the compression stroke, at the point when cylinder pressure is at its maximum, and the torque demand is the lowest when the piston approaches bottom dead center (BDC) of the exhaust stroke (Gavarraju, 2011). In the initial phase of the start-up, a breakaway torque is needed in order to overcome the static torque and to rotate the engine. This torque is much larger than the cranking torque and the magnitude is dependent on the ambient temperature, lubrication oil and the design of the engine (Emadi, 2005).

When the starter motor is engaged the energy equation can be established as Equation (2.1). Where IMEP is the indicated work (work on piston by cylinder gases), SMEP is the work done by the starter motor, FMEP is the total frictional losses and BMEP is the effective brake output available at the crankshaft (Rakopoulos & Giakoumis, 2009).

$$IMEP + SMEP = BMEP + FMEP \tag{2.1}$$

2.1.1 Gasoline DI engines

In gasoline direct injection (GDI) engines, unlike in port fuel injection (PFI) engines, the fuel is injected directly into the cylinders. The GDI engine therefore, in comparison to a PFI engine, improves the control of the air-fuel ratio and reduces the HC emissions during start-up. This is because direct injection eliminates the fuelling delay and the liquid film in the intake port and can thereby achieve a faster initial firing (Tong, et al., 2001). Furthermore, the manifold pressure drops very quickly from the initial atmospheric pressure with the increase of engine speed, which leads to more air induction in the cylinder during a short time. If the fuel injected is not regulated properly, which often is the case during start-up, this can cause a lean charge and thereby poor combustion and risk of misfire.

The characteristics of GDI cold start are dependent on the fuel distribution in the cylinder before ignition, which is dependent on the fuel injection, fuel vaporization and the mixing process. The factors influencing these parameters, such as fuel volatility that depends on the engine temperature, will therefore have a significant effect on the start-up performance. At cold start the low volatility components will be formed as droplets and wall film longer than the high volatility components, and therefore will more likely end up as unburned hydrocarbon (UHC) emissions. The incomplete vaporization can also lead to slow burn and misfire due to the reduced overall equivalence ratio, and thereby cause more UHC emissions (Tong, et al., 2001).

2.1.2 Diesel engines

In difference from gasoline engines, diesel engines are compression ignited. This complicates the cold start since the combustion is strongly dependent on the ambient conditions, and thereby the in-cylinder temperature and pressure. When temperature and pressure decreases, the ignition delay period is extended and the risk of misfiring increases. Misfiring is the most critical problem during diesel engine cold start as it could cause serious emissions and unstable combustion. In order to improve the start-up, the friction must be reduced or the combustion work, accelerating the engine, has to be increased.

To improve the cold start event there are different starting-aid devices, which increase the in-cylinder temperature at start-up. There are for example glow plugs and coolant heaters. Glow plugs are often located near the injector and help the charge to ignite. Heaters are often placed in the inlet system to increase the temperature of the air (Burrows, 1998).

The start-up can be further improved for diesel engines by a good injection strategy. Studies have shown that the start-up duration can be reduced by 68% when using pilot injection instead of conventional injection (Rakopoulos & Giakoumis, 2009). This technic reduces the ignition delay and thereby the risk of misfiring. In addition the injection can be slightly retarded to improve the start quality and increase the injection pressure. At low engine speed and temperature the injection pressure and thereby the spray quality is reduced, but with injection retardation it can be improved due to the more favorable air environment (Liu, 2003). Increased fueling at the start-up event can also reduce the start-up duration, since it provides higher temperature and pressure (Burrows, 1998).

The emissions formed during cold start for a diesel engine represents a large amount of the overall emissions. The main emissions are PM, CO and white smoke, which consists of condensed unburned fuel, HC, and water vapor (Bielaczyc, et al., 2001). These emissions are mainly generated by unstable and incomplete combustion caused by the elongated ignition delay, due to the low in-cylinder temperature and pressure. Another reason for the large amount of emission formation during cold start is that the exhaust cleaning devices, for instance oxidizing catalyzes, is not warmed up and is therefore not working properly. Another reason is that the engine speed is too low for the required governing speed of the turbocharger. This causes mismatch between fueling and air-supply, which gives a low air-fuel ratio and larger smoke emissions (Rakopoulos & Giakoumis, 2009).

2.2 Combustion strategy

Combustion during cold start is problematic due to the low temperatures. It is of importance that the combustion occurs without misfiring, but at the same time emissions needs to be considered and limited. In this section different combustion strategies for gasoline and diesel engines are presented.

2.2.1 Combustion strategies for DI gasoline engines

Different combustion strategies can be used in order to affect the power output, volumetric efficiency and the in-cylinder temperature and pressure. During start-up both stratified-charge and homogenous-charge combustion is of interest for a DI gasoline engine.

2.2.1.1 Homogenous-charge combustion

For homogenous-charge combustion the air-fuel ratio is close to stoichiometric and the air and fuel is well mixed at the point of ignition. The power output is controlled by the throttle, which limits the amount of air into the cylinder. A controller determines the amount of fuel to inject corresponding to stoichiometric conditions. The fuel is injected early to give the charge time to mix. This method limits the efficiency since it gives high in-cylinder pressure and increases the risk of knock. In order to control knock the spark timing can be adjusted so that the peak pressure is reduced. This does however affect the volumetric efficiency (Heywood, 1988).

2.2.1.2 Stratified-charge combustion

If stratified-charge combustion is used at start-up it can give lower emissions and a faster start. With stratified-charge combustion the fuel is injected late in the compression stroke and is ignited while still mixing with the air. This provides a range of different air-fuel ratios throughout the combustion chamber. This type of combustion is overall lean, which gives lower temperature and pressure and thereby less risk of knock. A drawback with this method is that, since the overall condition is lean, the catalyst does not work which increases the amount of emissions (Heywood, 1988).

The stratified-charge combustion is similar to a diesel cycle, which makes it possible to have a higher compression ratio and thereby higher thermodynamic efficiency. This means that less fuel is needed than for the homogenous-charge combustion, in order to provide the same torque. A difference from diesel engines is that it is easier to control the start of combustion, since the air-fuel mixture is ignited by the sparkplug.

The power produced is, in difference from a homogenous-charged combustion, controlled by the amount of fuel injected into the cylinder and it is important that the injected fuel gathers around the sparkplug for the combustion to start.

2.2.2 Combustion strategies for diesel engines

In comparison to gasoline engines a diesel engine can operate at a higher compression ratio and boost pressure, which has a positive effect on the combustion. This makes it possible for diesel engine to have an increased thermodynamic efficiency. Fuel with high cetane number as well as proper temperature conditions is important to achieve compression ignition, especially at cold starts. The combustion in a diesel engine is overall lean to avoid large soot formation (Mollenhauer & Tschöeke, 2010).

The fuel is injected into the chamber in the end of the compression stroke near TDC, where it is atomized and starts to mix with the air. The time from start of injection (SOI) until ignition is called the ignition delay period. As the piston moves closer to TDC, the premixed air and fuel ignites while fuel is still injected, called the premixed phase. This causes a fast increase in pressure and temperature in the cylinder, which heats the remaining charge. This improves the vaporization of the fuel and enhances the further combustion, which continues until all the injected fuel is consumed. This phase is called the mixed-controlled combustion or diffusion combustion. In this phase a diffusion flame burns with a low air-fuel ratio, which gives a low production of NO_x but a large formation of soot. Diesel engines usually operate with between 5-15% excess air to make sure that all of the fuel is burnt and thereby the HC emissions are reduced. The different phases can be seen in Figure 2.2 (Heywood, 1988).

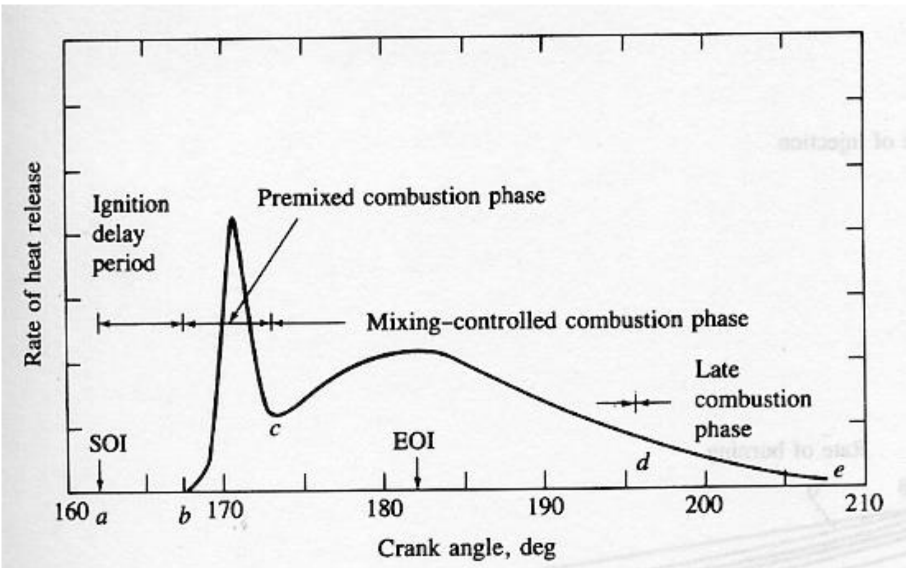

Figure 2.2 Rate of heat release diesel combustion (Heywood, 1988).

Since the fuel is injected late in the compression stroke and the mixing takes place in the cylinder, the engine speed of a diesel engine has to be limited in order to give the fuel time to evaporate and mix with the air. If the combustion takes place too late, energy could be lost. The lower engine speed affects the power output negatively, but can be compensated for by turbocharging. By using swirl the mixing of fuel and air can be improved. Swirl is created by the design of the inlet ports and the bowl in the piston (Mollenhauer & Tschöeke, 2010).

2.2.2.1 Fuel injection

The fuel injection is an important parameter to consider in order to controlling the combustion and emissions. Different injection strategies affect the rate of heat release, fuel consumption, soot formation and NO_x emissions. The penetration length as well as the spray affects the evaporation rate of the fuel (Mollenhauer & Tschöeke, 2010).

To improve the combustion and to optimize injection timing, pilot injection can be used. This means that one or more small fuel charges, pilots, are injected early in the compression stroke. These pilots are heated and vaporized and ignites slightly before the main charge is injected, causing higher temperature and more turbulence. This benefits the ignition of the main charge peak and gives a smoother combustion with lower noise (Mollenhauer & Tschöeke, 2010). In contribution it reduces the premixed peak pressure and makes the combustion more efficient (Wharton, 1991). The use of pilot injection is also beneficial for cold start, since it decreases the ignition delay. In Figure 2.3 a comparison of the rate of heat release and combustion noise with and without pilot injection is displayed.

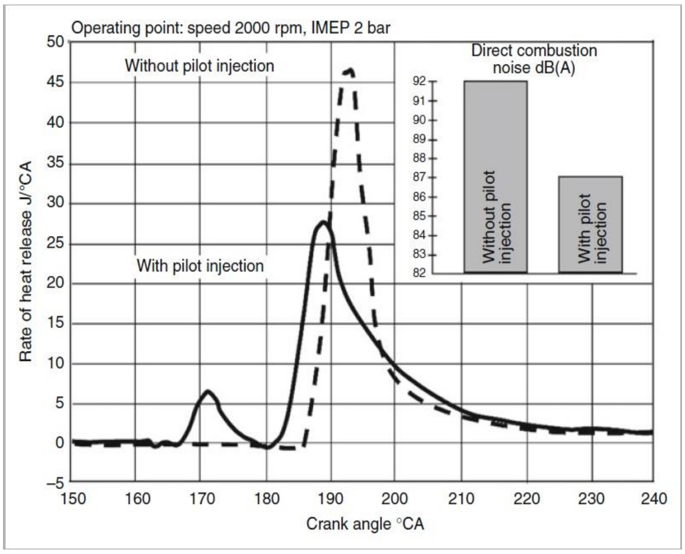

Figure 2.3 Rate of heat release and combustion noise with and without pilot injection (Mollenhauer & Tschöeke, 2010).

2.3 Cranking

The torque provided by the combustion at engine start-up is not enough to overcome the large amount of initial friction and the engine inertia. To overcome these and to build up the pressure in the fuel system the engine needs cranking, which can be done with an electric starter motor. The cranking resistance, starter motor and battery are accounted for in this section.

2.3.1 Moment of inertia

All rotating parts have inertia, which contributes to the torque demand dependent on the acceleration as described by Equation (2.2). Therefore the inertia of the engine and connected components is important to consider in transient modeling such as the start-up of the engine.

$$T = I\dot{\omega} \tag{2.2}$$

The inertia that needs to be considered during start-up differs depending on the transmission, if it is a manual or automatic. For both structures the generator and the engine inertia, which includes the reciprocating and rotating masses moment of inertia for each cylinder, needs to be accounted for (Rakopoulos & Giakoumis, 2009). But there are different connections to the transmission depending on the type of gearbox. For a powertrain with manual transmission a dual mass flywheel (DMF) transmits the torque from the engine and for a powertrain with an automatic transmission it is instead a torque converter, which gives a different contribution to the total inertia. The transmission inertia however, should not be accounted for in either assembly since it is disengaged during start-up.

2.3.1.1 DMF

The function of the DMF is to absorb excess energy and reduce the crankshaft speed fluctuations. The DMF divides the conventional flywheel into two parts, one primary mass on the engine side and one secondary mass on the transmission side. The two masses are connected with a damper spring system supported by a bearing to enable free rotation. The torque is transmitted via the spring system and by the friction of the coils of the springs and a rigid clutch is used to connect and disconnect the engine to the transmission (Cavina & Serra, 2004). In Figure 2.4, a typical construction of a DMF is shown where the damper spring system, primary side and secondary side could be seen.

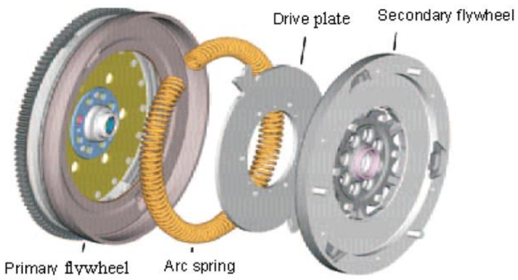

Figure 2.4 A typical construction of a DMF (Mohire & Burde, 2010).

During start-up the engine speed is run through the DMF resonance, causing the secondary flywheel to rotate with an opposite angular displacement if compared to the primary flywheel (Fidlin & Seebacher, 2006). An example of this phenomenon can be seen in Figure 2.5.

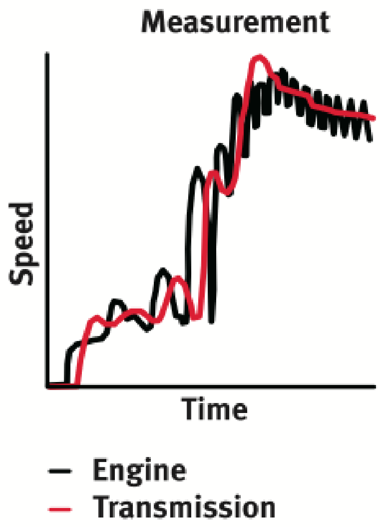

Figure 2.5 The resonance phenomenon of the DMF during start-up (Fidlin & Seebacher, 2006)

2.3.1.2 Torque converter

The torque converter transfers torque between the engine and the automatic transmission with the principle of hydrodynamic power transfer. The torque ratio improves acceleration and performance while the slip isolates the torsional vibrations from the engine to the drivetrain. The converter is composed by a turbine, stator and an impeller/pump, which can be seen in Figure 2.6. The impeller converts mechanical power from the crankshaft to fluid power, which is directed to the turbine that converts the fluid power back to mechanical power to the transmission. The stator works as a redirector of the fluid back to the impeller again. The slippage between the impeller and the turbine is used during acceleration at lower gears; the torque converter then multiplies the torque output to the transmission. The torque converter also consists of a lock-up clutch, which is not seen in the figure. The lock-up clutch is used to minimize the losses due to the slip and bypasses the torque converter when slippage is not needed (Pohl, 2003).

During start-up of the engine the automatic transmission is disengaged and the turbine therefore rotates freely. The turbine however rotates with a lower speed than the impeller and there is always a speed ratio between them when the loch-up clutch is not active.

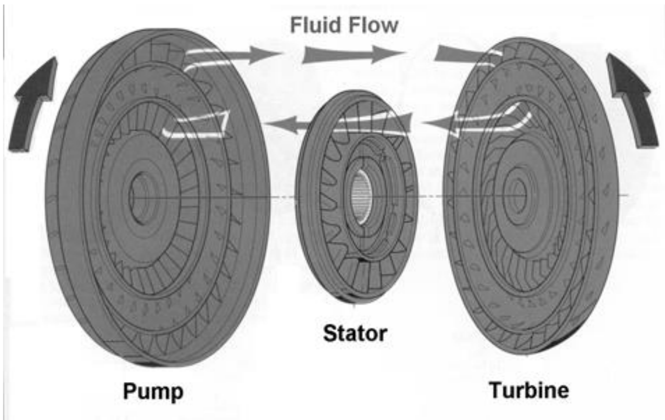

Figure 2.6 Main components and fluid flow of a torque converter (Kowalski, et al., 2005).

2.3.2 Friction

Friction losses in an engine are important to consider since it influences fuel consumption and engine durability. The energy appears as heat and is removed by the cooling system. The friction losses is the part of the work used to overcome the friction of the bearings, pistons and mechanical components and therefore correspond to the difference between the indicated and brake work by Equation (2.3) (Heywood, 1988).

$$FMEP = IMEP - BMEP \tag{2.3}$$

Friction losses in an engine can be divided in to two categories. Firstly the mechanical friction losses, which is the energy needed to overcome the resistance between moving parts. This occurs in piston assembly, valve train and bearings. The second part is the accessory work from the generator, AC and the oil-, water- and fuel pump. The total friction increases with engine speed, varies with load and is affected by the oil temperature since it influences the viscosity of the lubrication (Stotsky, 2009).

Different components in the engine operate at different lubrication modes with various contributions to friction losses. Lubrication consists of three different modes: hydrodynamic, mixed and boundary. Hydrodynamic mode is the desired mode during operation, since a liquid film is separating the components and thereby lowers the friction and minimizing wear. This mode is present in the piston assembly and in the loaded bearings. In this mode the friction is independent of the surface roughness of the parts as well as the engine speed and only dependent on lubricating properties (Heywood, 1988). When the load increases or the engine speed decreases the film gets thinner and irregular. This mode is called mixed lubrication and is present in the valve train. When the speed is reduced or the load increased further the boundary mode is reached. This mode often occurs at TDC and BDC, but can also occur in the valve train (Rakopoulos & Giakoumis, 2009).

2.3.2.1 Start-up friction

At cold start there is an increase in friction losses mainly because of the increase in viscosity of the oil lubrication, due to the lower temperature. Thereby the minimum starting speed increases as the temperature decreases. To improve start-up and to lower the resistance at cranking, the oil has to have low viscosity at low ambient temperature. Furthermore, since the viscosity decreases as the temperature increases the oil viscosity should not be unacceptably low at high temperatures to allow a wide range of starting temperatures (Xin, 2011). By reducing the increased amount of friction the starting event can be improved, since the combustion work required to accelerate the engine will be decreased (Burrows, 1998). In Figure 2.7 an example is presented on how FMEP can vary with different oil compositions and temperature during engine start-up.

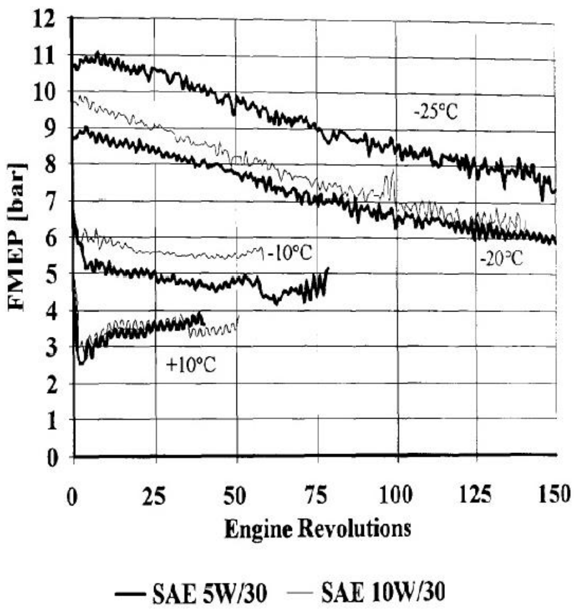

Figure 2.7 FMEP dependent on oil and temperature (Burrows, 1998).

At cold start the engine speed and temperature is low and therefore the boundary lubrication mode is present in almost all components, which means ineffective oil film and oil starvation that increases wear and friction losses. Initially the work performed by the combustion will not be able to overcome the friction in the engine and this is why the starter motor is needed. After some cycles however, the combustion output will be able to overcome the friction causing the indicated torque to be equal to the frictional torque and idle speed is reached. The increase in viscosity due to the low temperature might affect the cranking speed to be reduced. This impacts the in-cylinder temperature and pressure, as well as increasing the blowby and heat transfer, which might prolong the start-up event (Burrows, 1998).

2.3.3 Starter motor

The starter motor is provided with energy from a battery, which is charged by an alternator. The performance of the starter equipment is dependent on temperature and the electric system has to be sized so that the internal combustion engine can be started over a wide range of temperatures (Bosch GmbH, 2013).

The starter motor is usually a permanent magnet direct current (PMDC) electric motor. These motors are suitable as starting motors since they develop the high initial torque required to overcome the large inertia and friction. A solenoid is mounted on the starter motor and when the driver turns the key the battery supplies the solenoid with current and the starter motor starts. The solenoid also engages the pinion of the starter motor to the flywheel of the engine, which causes the engine to rotate (Bartilotti, et al., 2008). The starter motor is engaged until the engine reaches a certain disengagement speed; the engine is then considered to run by its own power. The starter can be rather small, since there is a large ratio between the pinion and the flywheel (Bosch GmbH, 2013). Figure 2.8 shows a schematic figure of the starter motor with the pinion and ring gear, which is connected to the flywheel.

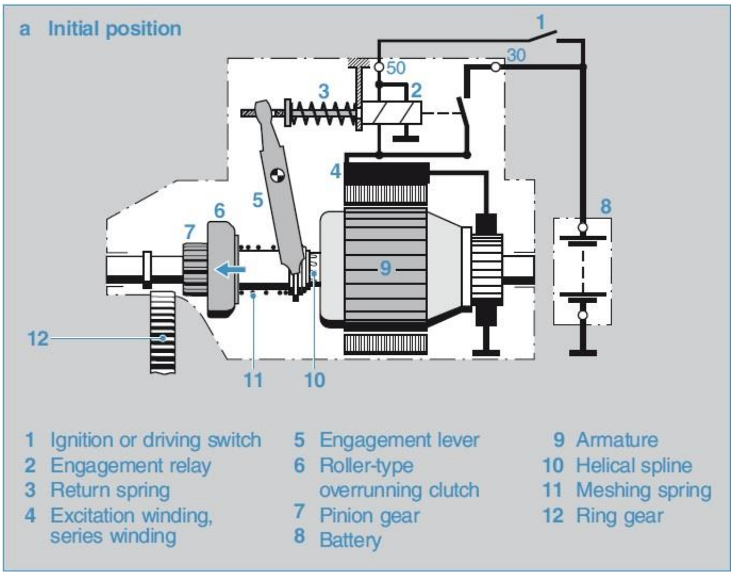

Figure 2.8 Starter motor with pinion and ring gear (Bosch GmbH, 2013).

The frequency of the cranking torque increases at higher engine speed, since the pistons move faster and thereby the torque demand changes faster. The amplitude of the torque increases when the temperature is reduced, since more torque is needed due to the increased friction. The internal resistance in the starter motor is also dependent of the temperature, which means that it can deliver less energy at lower temperatures. If the cranking speed is reduced it might extend the start, since it takes longer time to achieve the needed in-cylinder conditions for ignition and the time until the first burn occurs is increased (Burrows, 1998).

2.3.4 Battery

The battery, which supplies the starter motor with energy, is usually a 12V lead-acid battery. These batteries have low cost in comparison to the high energy. The battery is charged with an alternator, which converts mechanical energy to electrical energy, and stores it as chemical energy in the battery.

The battery needs to have enough energy stored in order to start the engine and to maintain some of the electric consumers for a certain period of time when the alternator is shut off. This is regulated with a control system. When the vehicle is running, the alternator provides other electrical consumers such as lights, fans, heaters, fuel injection and ignition system with energy. When the electrical consumers are consuming less energy than the alternator produces the extra energy charges the battery. The control system limits the current if it exceeds the maximum permitted charging current at a specific temperature (Bosch GmbH, 2013).

The efficiency of the starter motor is directly dependent on the power the battery can provide. It is therefore important that the battery can provide energy at different conditions, for example at cold temperatures. At low temperatures the internal resistance in the battery increases and it may not provide the necessary energy dependent on capacity, low-temperature test current, state of charge and internal resistance. Battery performance drops as temperature decreases, leading to a reduction in cranking speed (Burrows, 1998). The efficiency of the battery is also affected by the age; when the battery ages the efficiency decreases.

3 Modelling

In order to understand the start-up event and what needs to be considered to create a simulation model, a literature review was carried out. There were limited previous project within this area so the focus on the literature review was to gain a general understanding of the subject.

This section will account for what needs to be considered when modelling start-up in GT-Power, as well as the possibilities and approaches to implement necessary changes. Furthermore, it will present the test-rig data collected for verification.

3.1 Modelling in GT-Power

GT-Power is an engine simulation tool, which can be used for engine performance analysis. The software enables examination of different performance characteristics for various engine set-ups, such as SI, DI and HCCI combustion with different fueling systems as well as turbocharging and supercharging. It makes it possible to investigate torque and power curves and the variation in volumetric efficiency, airflow, variable valve timing and fuel consumption etc. The software allows simulations for both steady state and transient operation (Gamma Technology Inc., 2014).

There is also a post-processing program called GT-Post available. Once a simulation is run, the desired outputs such as fuel consumption and powertrain behavior, can be viewed and saved in GT-Post in form of graphs and tables (Gamma Technology Inc., 2014).

3.1.1 Predictive and non-predictive combustion model

In GT-Power the combustion model can be described as either predictive or non-predictive. For a predictive model the burn rate is predicted from various inputs, such as fuel and gas pressure, temperature, residual fraction and equivalence ratio. In the non-predictive model however, the burn rate is imposed as input to the simulation. The imposed burn rate will therefore always be followed, if there is enough fuel available, and will not be dependent on the conditions in the cylinder. The air-fuel mixture will only burn in the prescribed rate. A non-predictive model will therefore run faster than a predictive one with added complexity, but the accuracy of the results will be lower. Hence, it could be used when the intended use of the model and the variables of interest have a small effect of the burn rate. A non-predictive model also requires a high number of measurement points and therefore makes it less useful as a compliment to test-bench measurements. A predictive model however can simulate the entire operating map with only a few sets of parameters, on the basis of physical and chemical models (Reasearch Institute of Automotive Engineering and Vehicle Engines Stuttgart, 2013).

3.1.2 Example start-up model in GT-Power

There is an example model of stop-start simulation for a naturally aspirated port-injected SI gasoline engine with four cylinders available in GT-Power. The simulation starts with a running engine at 1000 RPM and then the engine is shut down and slows down to halt. When the engine speed reaches 0 RPM the engine is cranked by a starter motor and the engine starts and accelerates again.

The fuel injectors are off until the engine starts and the amount of fuel is then controlled by an air-fuel ratio that is dependent on engine speed. The combustion is described with a predictive combustion model and the Woschni heat transfer model is used to describe the wall heat transfer. However, due to the low engine speed it is adjusted by the function “Low Speed Heat Transfer Enhancement for Woschni* Models”, available in the model. This function will calculate a minimum heat transfer coefficient, in order to avoid unrealisticly low heat transfer at low engine speeds.

The starter motor model has requested torque as input, and uses a motor efficiency map to determine output torque and speed. The requested torque is a linearly decreasing curve dependent on engine speed that reaches 0 Nm at the disengagement speed. The starter motor is connected to the crankshaft with a gear ratio and is disengaged with a clutch. A 12 V battery is modelled in order to provide the starter motor with energy.

The friction is described with a simple model that considers the dependency of pressure and speed. The friction model is sufficient, since the engine is warm during the simulation, which makes the friction behavior easier to predict and no breakaway friction needs to be considered.

3.2 Engine specifications

In order to investigate the start-up behavior for cold start, VCC’s four-cylinder engines, one gasoline and one diesel engine, were considered. The gasoline engine was a turbocharged low-performance engine with manual transmission, while the diesel engine was a turbocharged mid-performance engine with automatic transmission. These engines were chosen because there were comprehensive test-rig data available and they covered a wider range of configurations. Validated steady state models of the engines in GT-Power were used as a base, in order to create the start-up simulation models.

3.2.1 Gasoline engine

The combustion during the start-up of the considered gasoline engine is both stratified and homogenous. During the run-up phase the combustion is stratified and the fuel is injected late in the compression stroke, in order to get a faster start. After the run-up, when transition to idle, the combustion changes to homogenous with a post injection. The amount of fuel injected in the post injection varies with starting temperature and increases with decreasing temperature. At -30°C the injection is split into two injections with equal amount of fuel. This is done in order to provide a locally rich mixture at the spark plug, which improves the combustion and enables great spark retard. The spark is retarded during the idle phase in order to increase the exhaust temperature to heat the catalyst and to stabilize the engine speed at idle.

3.2.1.1 Gasoline start-up process

For the gasoline engine only one test for each temperature was available, which causes an uncertainty due to the inability to compare deviations between tests runs. In

Figure 3.1, the start-up behavior in terms of engine speed from available test-rig data for the gasoline engine is presented. The figure shows an unfiltered engine speed at 20°C and -30°C as percentage of the idle speed, here referring to the idle speed after the three-way catalyst has been heated. When creating the model, regard has only been taken to the first seconds of operation, during the first idle phase.

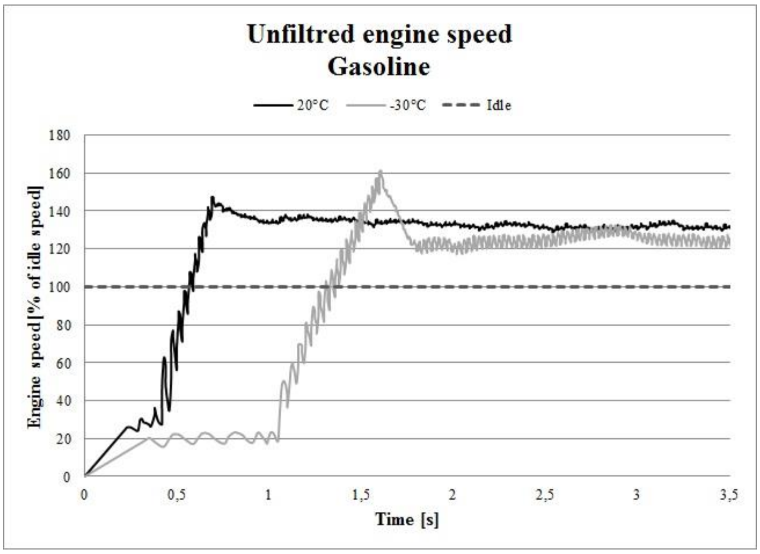

Figure 3.1 Unfiltered engine speed as percent of idle speed for the gasoline engine during start-up at 20°C and -30°C.

In Figure 3.2, the same engine speed is shown but filtered. This curve makes it possible to compare the start-up process between the measured test and the modelled. The filtered engine speed creates an understanding on the general speed variation, while the unfiltered engine speed provides the possibility to study the local speed variation.

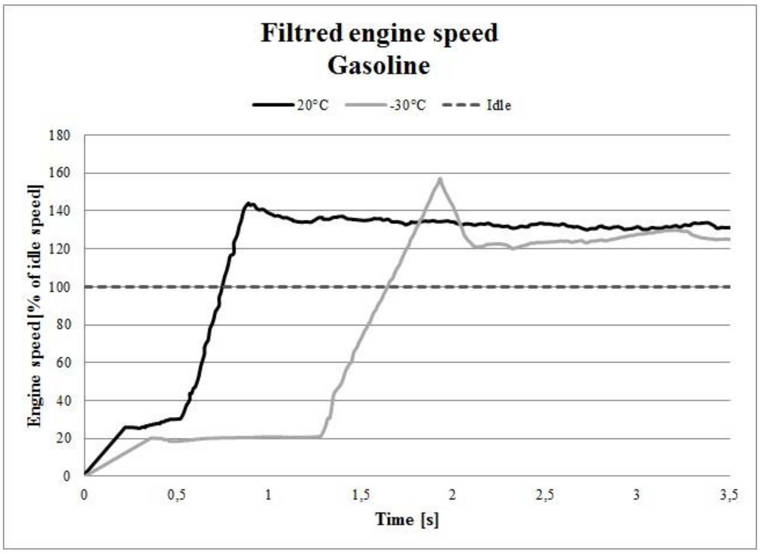

Figure 3.2 Filtered engine speed as percent of idle speed for the gasoline engine during start-up at 20°C and -30°C.

3.2.2 Diesel engine

In the injection strategy for the diesel engine three pilots and one main injection are used in order to increase the temperature in the cylinders and thereby get a more robust start-up. Pilot injections also create more turbulence and decrease the ignition delay, which further enhances the ability to start the engine.

3.2.2.1 Diesel start-up process

For the diesel engine there were multiple tests available for the different temperatures. The data was examined and it was found that the deviation between the tests was insignificant and therefore only one test for each temperature was used as reference. In Figure 3.3 the engine speed as percent of idle speed for the start-up process from test-rig data for the diesel engine at 20°C and -30°C, is illustrated. The dashed line indicates the idle speed. Even though the oxidation catalyst needs heating to work properly, it can be seen that this is not prioritized for diesel start-up at 20°C and the engine stabilizes at the lower engine speed directly. However, at -30°C the idle speed is enlarged in the same manner as for the gasoline engine.

The filtered engine speed as a percent of the idle speed for the diesel engine can be seen in Figure 3.4. In the figures it can be seen that it takes longer time for the diesel engine to stabilize at idle speed than for the gasoline engine, which is due to the lower idle speed. It can also be seen that during the transition to idle and during the idle phase, the engine speed of the diesel engine has higher and more constant amplitude then the gasoline engine. This is due to that the diesel engine generally has higher cylinder pressure than a gasoline engine.

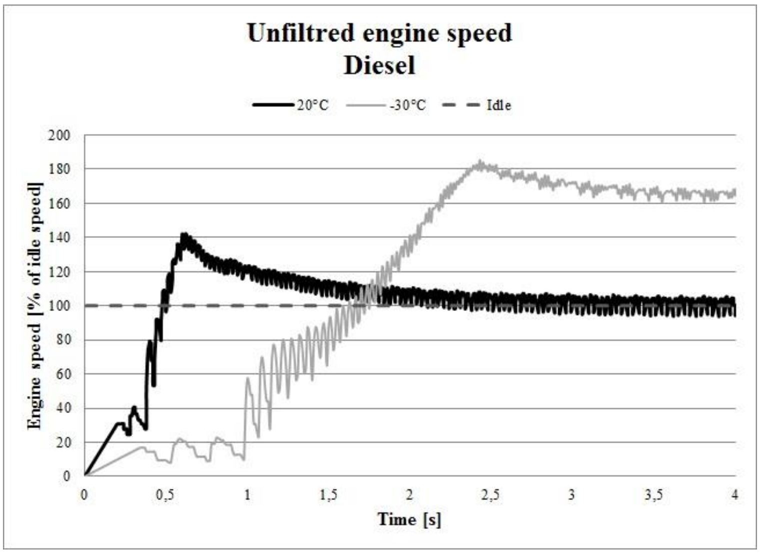

Figure 3.3 Unfiltered engine speed as percent of idle speed during start-up for the diesel engine at 20°C and -30°C.

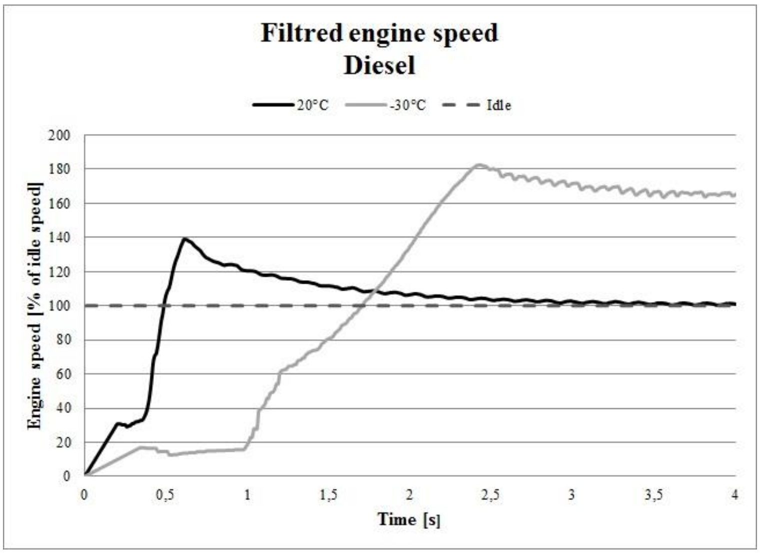

Figure 3.4 Filtered engine speed as percent of idle speed for the diesel engine during start-up at 20°C and -30°C.

3.3 General model adjustments

When simulating the start-up in GT-Power the models had to be changed from speed mode to load mode, which means that a given torque is provided to the crankshaft as an input and the speed variation due to the torque is calculated. At steady state operation the speed mode is usually used, where the engine operates at a given speed and the load variation is calculated. At steady state simulations there are often dwells in the model, to make sure of steady operation. These had to be removed in order to simulate start-up, since the calculations needed to start at the first cycle. If important parameters are dwelled for a couple of cycles the start-up event is strongly affected.

During the cranking phase the engine only pumps air and no combustion occurs until the pressure in the fuel system is high enough to inject fuel. The time it takes to build up the pressure depends on temperature and is determining at which cylinder the first combustion occurs. Therefore the fuel system needed to be modeled, in order to create a predictive model. However, there was no calibrated fuel system model available, and due to the time limit of the project a simplification had to be made. A dwell was added to the injection, which corresponds to the time it takes to build up the fuel pressure in the specific test-rig data, in order for the model to determine at which cylinder the first injection should occur.

The turbocharger places an exhaust back pressure on the engine, which increases pumping losses and is affecting the start-up negatively. To avoid the back pressure the wastegates are wide open during start-up, which has to be set in the models as well.

In similarity to the example model the cylinder wall heat transfer was calculated with the Woschni model. The function “Low Speed Heat Transfer Enhancement for Woschni* Models” was used as well, in order to make the model predict the wall heat transfer realistically at low engine speeds.

3.4 Cranking

In this section the modelling of the moment of inertia and the friction affecting the start-up process is presented. Furthermore, the modelled torque provided by the starter motor in order to overcome the cranking resistance is described.

3.4.1 Moment of inertia

As explained in Section 2.3.1, the inertia of the engine and connected parts is of great importance when simulating the start-up of the engine. In the model in GT-Power the inertia was divided into crank slider inertia and total inertia, which was modelled in two different objects. The crank slider inertia is the effective rotational inertia applied to the crankshaft due to the crank-slider mechanism of the connecting rods and pistons, which varies throughout the cycle. The object for the crank slider inertia therefore calculates the angle dependent inertia and only affects the instantaneous acceleration and torque of the crankshaft. The total inertia includes the effective rotational inertia of the crankshaft and all connected moving components, which was modelled as cycle averaged inertia referenced to the rotational speed of the crankshaft. The inertia from the components was lumped and added as total moment of inertia in the object.

3.4.1.1 Gasoline cranktrain inertia

In order to define the total moment of inertia for the gasoline cranktrain, the inertia contribution from each component was determined. The concerned components are the engine, generator and the DMF connected between the engine and the manual transmission. The contribution from the engine and the generator was known, as well as the total contribution from the DMF. There was however an uncertainty of the contribution from the DMF due to the resonance phenomenon during start-up, explained in Section 2.3.1.1. The influence of the secondary flywheel of the DMF was therefore unknown. By recommendations at VCC, the secondary side was therefore neglected and only the primary side of the flywheel was accounted for, with the knowledge that it gives an uncertainty for the model. The total moment of inertia was calculated according to Equation (3.1).

$$I_{tot.gasoline} = I_{eng.gasoline} + I_{gen} + I_{flywheel.prim} \qquad (3.1)$$

3.4.1.2 Diesel cranktrain inertia

As for the gasoline engine, the total moment of inertia was determined from the contribution of each component. The contribution for the diesel cranktrain similarly includes the engine and the generator inertia, but instead of the flywheel it includes the torque converter due to the automatic transmission. As described in Section 2.3.1.2, the turbine in the torque converter rotates freely with a speed ratio relative the impeller when the transmission is disengaged. The inertia for the converter during start-up was therefore hard to predict, besides the impeller inertia, the turbine and the fluid gives an unknown contribution. The inertia for the torque converter was therefore determined to be the inertia used to dimension the starter motor, which was in the same magnitude as the impeller inertia and about half of the turbine inertia. The total moment of inertia was calculated according to Equation (3.2)

$$I_{tot.diesel} = I_{eng.diesel} + I_{gen} + I_{converter} \tag{3.2}$$

3.4.2 Friction model

GT-Power does not have any valid friction model for the engine start-up process, since none of the present models consider the viscosity of the oil at the low temperatures and speed present at engine start-up. The initial oil starvation and the built up of oil film also needs to be considered, in order to predict the friction behavior.

The engine friction model, which was considered to be the best approximation in order to describe the friction during the start-up process, was the Chen-Flynn friction model. The model calculates the friction using a constant part of FMEP (*C*), peak cylinder pressure (C_p), mean piston speed (C_s) and mean piston speed squared (C_{ss}). The function is described in Equation (3.3), where P_{max} is the maximum cylinder pressure and S_p is the mean piston speed. The constant part of FMEP is imposed by the user, which enabled the possibility to illustrate the high friction during cold start.

$$\mathrm{FMEP} = \mathrm{C} + \left(\mathrm{C_p} * \mathrm{P_{max}}\right) + \left(\mathrm{C_s} * \mathrm{S_p}\right) + \left(\mathrm{C_{ss}} * \mathrm{S_p}^2\right) \tag{3.3}$$

Since no data or information of the friction behavior was obtained from test-rig data the factors in the equation were determined from existing low load experiments. The constant part of FMEP had to be determined, in order to represent the start-up friction. It was therefore adjusted in order to make the modelled cranking speed correlate with the measured cranking speed from the cranking test presented in the next section.

3.4.3 Starter motor model

For both the gasoline and the diesel model, the cranking torque provided by the starter motor had to be modelled. In the example model the starter motor was modeled in GT-Power, but it was considered inaccurate since the input requested torque is unknown. This would create too large deviation with reality and in order to receive a realistic input torque the starter motor would need to be dynamically modeled in detail.

In the models the cranking torque was instead provided to the crankshaft by adding a torque-generating object, where the user defines the torque. By using this object no care had to be taken to the starter motor and battery characteristics, for example age and temperature dependency and the modeling of the starter equipment could become simpler.

3.4.3.1 Cranking torque

In order to determine the cranking torque provided by the starter motor, test-rig data was collected from a cranking test. The performed test provided the torque from standstill up to constant cranking speed, without any combustion taking place.

In Figure 3.5 the cranking torque, as a percent of full load for the gasoline engine, is presented at 20°C and -30°C. The initial peak torque in the figure indicates the breakaway torque. In the figures it can be seen that more torque is needed at lower starting temperatures, which is due to the increased amount of friction. The large friction at lower temperatures also gives a lower cranking speed, which results in lower frequency and higher amplitude of the torque.

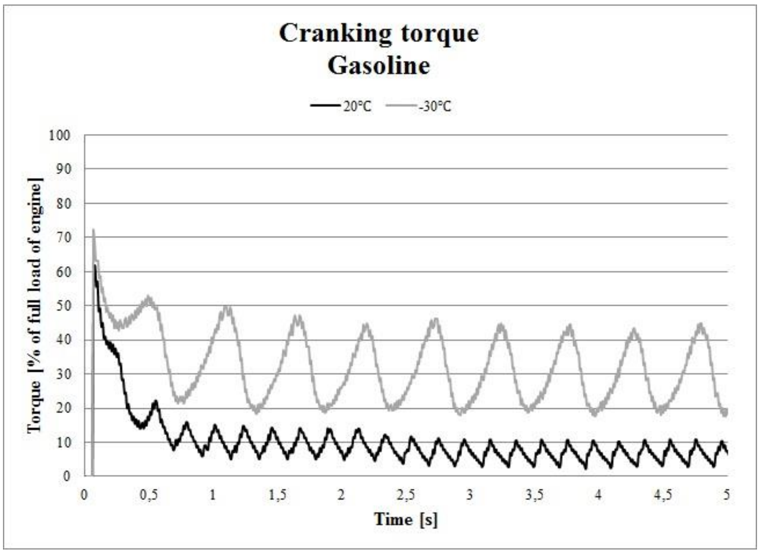

Figure 3.5 Cranking torque as a percent of full load of the engine for the gasoline engine at 20°C and -30°C.

In Figure 3.6, the cranking torque for the diesel engine from similar tests as for the gasoline engine is presented. It can be seen in the figure that the starter motor torque for the diesel engine is similar to the one for the gasoline engine but differs in frequency and amplitude, which is due to the difference in engine characteristics. It can also be seen, with the knowing of that the diesel engine provides a higher maximum load, that the cranking torque for the diesel engine is larger for all temperatures. This is due to that the diesel engine has higher friction, higher compression ratio and has an automatic transmission, which causes larger inertia.

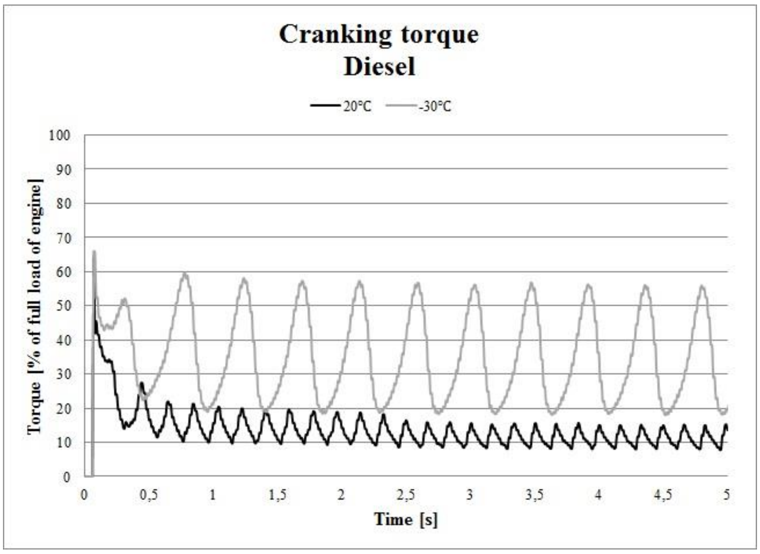

Figure 3.6 Cranking torque as a percent of full load of the engine for the diesel engine at 20°C and -30°C.

As can be seen in the figures the frequency of the torque after the first cycles is constant, due to the constant engine speed. But during an actual start-up the frequency of the torque increases when the engine is accelerated, which caused an uncertainty when using the torque in the models. The constant cranking speed from the cranking tests for both engines can be seen in Figure 3.7 and Figure 3.8. The figures show the difference in cranking speed due to the higher friction at lower temperatures. The deviation between the cranking speed and the speed when the starter motor disengages therefore is larger at lower temperatures, which increases the offset in frequency between the modelled torque and the actual torque needed for a realistic start-up.

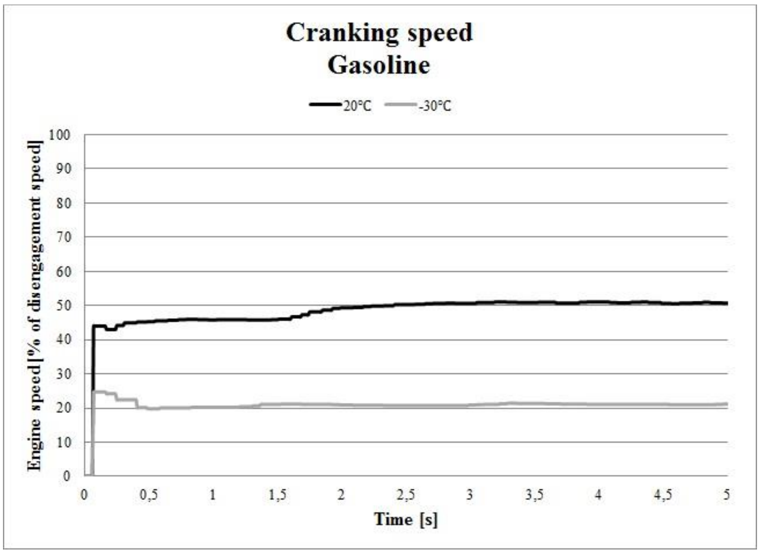

Figure 3.7 Engine speed at the crankshaft for the gasoline engines during cranking at 20°C and -30°C.

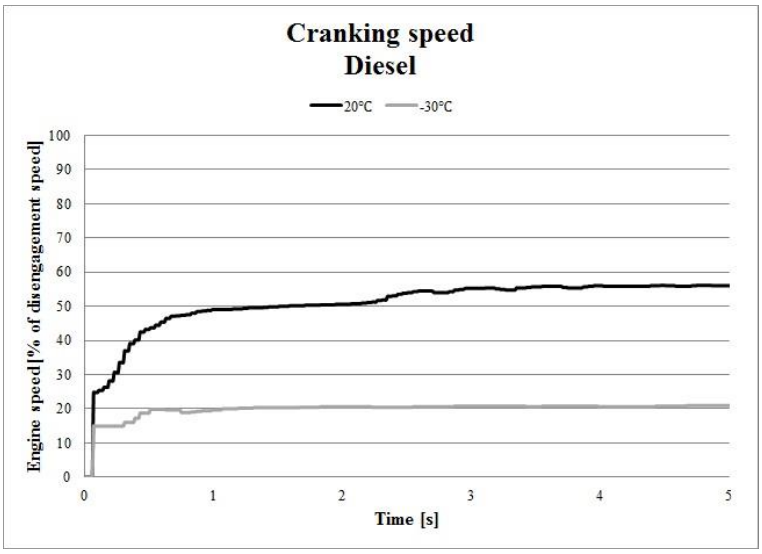

Figure 3.8 Engine speed at the crankshaft for the diesel engines during cranking at 20°C and -30°C.

Another extensive uncertainty was that the starting crank position of the engine for the cranking test was unknown. If the position does not correlate with the position in the model, this could cause uneven phasing of the torque. The torque minimum could therefore occur at the required torque maximum, leading to a prolonged or failed start-up. This problem was found to be more significant at lower starting temperatures. The extensively higher friction and thereby higher torque amplitudes increase the importance of correct torque at every point of operation.

The inaccuracy of the provided constant torque and the lack of a valid friction model were found overly significant for -30°C. The model was therefore not able to perform the simulation remotely close to the reality and was found invalid for this temperature. Hence, only the starting temperature at 20°C was considered for the rest of the modeling.

3.4.3.2 Starter engagement

The starter motor is engaged until the engine is considered to run by itself, which depends on the starting temperature and the type of engine. The duration at which the starter motor is engaged was determined from the test-rig data.

Figure 3.9 shows the engagement of the starter motor for the gasoline engine. The solid curve shows the engine speed as percent of idle speed and the dashed line indicates where the starter motor is engaged. For the dashed line 100% indicates that the starter motor is engaged and 0% indicates disengaged, while for the solid line 100% is idle speed.

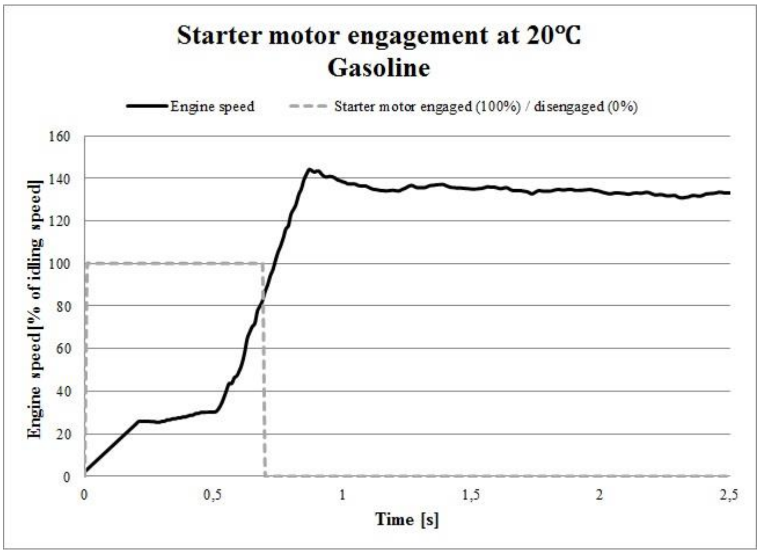

Figure 3.9 Starter motor engagement for the gasoline engine from test-rig data.

In Figure 3.10, the engagement of the starter motor for the diesel engine can be seen similarly to the gasoline engine. The starter motor is disengaged at about the same engine speed for both of the engines. However, in the figure it can be seen that the starter motor for the diesel engine is engaged during a shorter period of time, which indicates that the start for the diesel engine is faster and thereby reaches the disengagement speed faster.

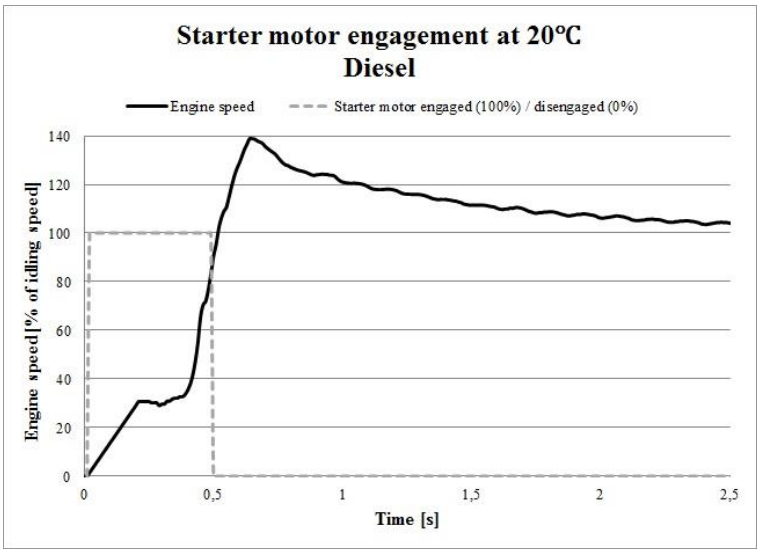

Figure 3.10 Starter motor engagement for the diesel engine from test-rig data.

3.5 Gasoline combustion modelling

The modelling in GT-Power of the parameters affecting the combustion for the gasoline engine is presented in this section, such as the combustion model, controlling and valvetrain.

3.5.1 VCC gasoline combustion model

At VCC there is a calibrated predictive combustion model available for the gasoline engine at steady state operation. The predictive model available for the gasoline engine is the FKFS combustion model, which is a quasi-dimensional model (QDM), which simulates homogenous Otto combustion based on the equations of energy conservations. A QDM splits the combustion into two zones; burned and unburned. The burned zone consists of burned exhaust gases and the unburned zone of a homogenous mixture of fuel, air and residuals. In the FKFS model these zones are separated by a flame zone that is thermodynamically considered as a part of the unburned zone (Reasearch Institute of Automotive Engineering and Vehicle Engines Stuttgart, 2013).

The combustion during the start-up for the gasoline engine is both stratified and homogenous, placing a demand on the model to be able to simulate two different combustion strategies. The predictive stratified FKFS combustion model is however not available at VCC and no other predictive model was representative for the stratified run-up, without more extensive multi-dimensional calculations and calibration.

3.5.2 Combustion model implementation

Due to the low temperatures and speed at start-up it was considered no risk of knock and the knock model in FKFS was turned off. To model the stratified combustion a non-predictive combustion model with user-imposed combustion profiles was used. One burn rate curve was chosen from multiple cycles to be representative for the entire stratified phase. With this method the combustion is independent of the injection strategy and all the fuel can be injected at one point early in the compression stroke.

The model was then intended to switch to the FKFS model, in order to make the rest of the start-up event predictive. However, in Figure 3.11 it can be seen that after the switch to the FKFS model the engine speed decreases for one cycle before it continues to run-up. This was because the FKFS model was hard coded to start the calculations at its second cycle and cannot use data from another combustion model at its first cycle. The dotted line illustrates where the combustion switches from user defined stratified combustion model to the predictive FKFS model.

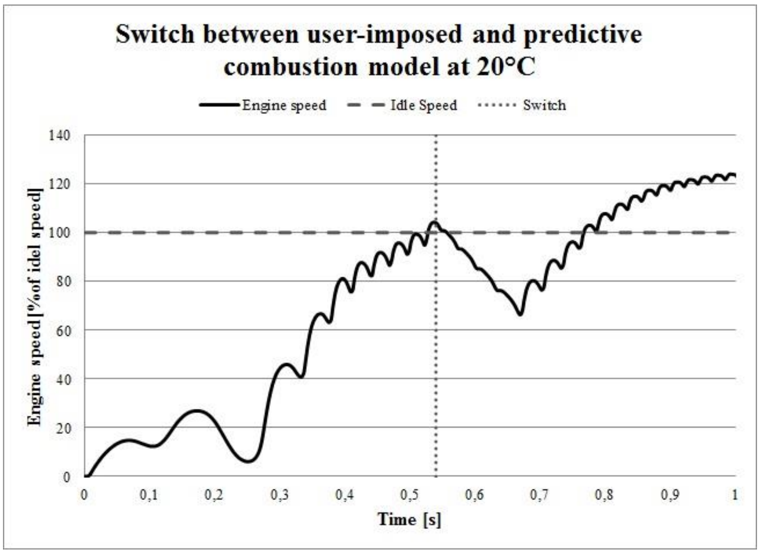

Figure 3.11 Engine speed variation as percent of idle speed when switching from stratified-charge combustion model to FKFS model.

The predictive model was considered unfeasible and instead the non-predictive combustion model had to be used for the entire start-up simulation. One burn rate curve was chosen to represent the run-up phase and one to represent the idle phase, in order to create a general model representing a typical start-up. In reality there is a deviation in burn rate for the different cycles, while in the model there is only a deviation between phases.

The spark timing had to be adjusted to make the timing of the combustion for the different cycles to correlate with test-rig data. Cylinder pressure was studied in order to verify the correlation, if the cylinder pressure correlates it indicates correct spark timing. The measured pressure from the test-rig data as percent of the maximum firing pressure for the gasoline engine can be seen in Figure 3.12. It can be seen that in this test the engine needed three pump cycles before the first ignition. The figure also illustrates that the pressure has different shape at the idle phase, which is caused by the retarded combustion.

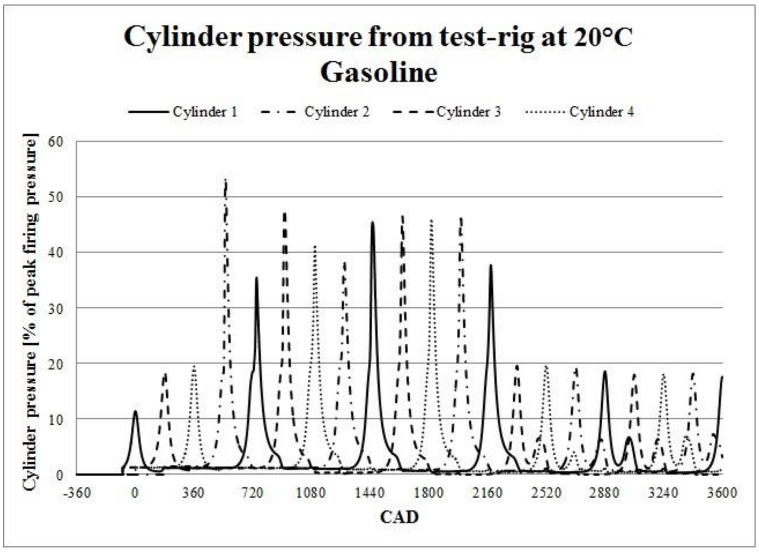

Figure 3.12 Cylinder pressure as percent of peak firing pressure for the gasoline engine from test-rig data.

3.5.3 Combustion controlling

The amount of fuel injected in the gasoline engine is determined by the amount of air coming into the cylinder. The model detects the amount of air and injects the corresponding amount of fuel, in order to obtain a predefined lambda value. During the first couple of cycles the charge is lean due to the stratified combustion. When the combustion switches over to homogenous-charge lambda equals one, in order to provide the conditions needed for the three-way catalyst.

The airflow into the cylinder is dependent on the throttle opening, density of the air and inlet pressure. The speed is too low during start-up for the turbocharger to work properly, therefore the main parameter affecting the amount of air coming into the cylinder is the throttle angle. From measurements, information was provided for the throttle angle during the run-up phase dependent on water temperature in the engine at start-up. In Figure 3.13 the relation between the throttle angle and water temperature is illustrated. In the figure it can be seen that at lower temperatures the throttle is more open in order to induct more air.

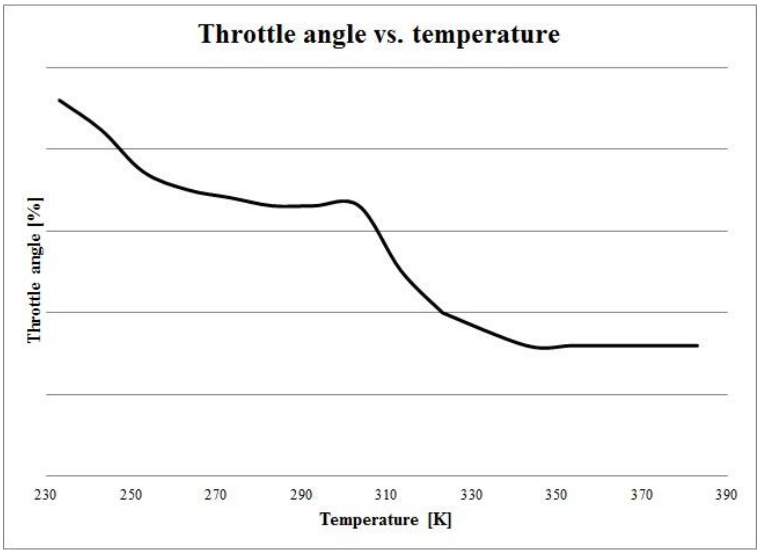

Figure 3.13. The throttle angle for the run-up phase dependent on water temperature.

However, the throttle input to the model was in diameter, and therefore the information had to be translated. In Figure 3.14 the relation between throttle angle and diameter used for translation can be seen.

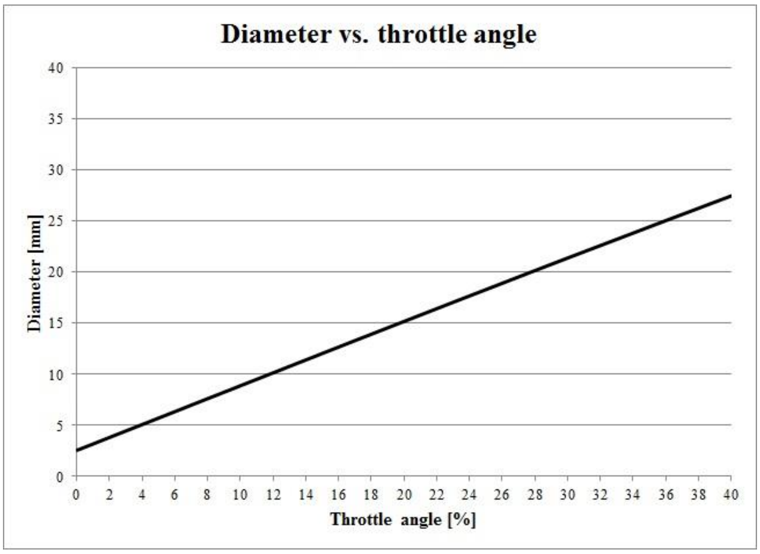

Figure 3.14. Relation between throttle angle and diameter.

When the idle speed was reached, a PI controller regulated the throttle diameter. The controller was targeting the idle IMEP, since it is hard to target the close to zero BMEP at idle. The target IMEP was determined according to Equation (3.4).

$$IMEP = FMEP + PMEP \tag{3.4}$$

3.5.4 Valvetrain

The gasoline engine at VCC has variable valve timing (VVT), which needed to be adjusted in the model in order to obtain optimal gas exchange properties. The VVT allows the valve lift to vary with engine speed, in order to increase performance, fuel economy and emissions. The exhaust camshaft can be retarded 30 CAD towards TDC and the intake camshaft can be advanced 50 CAD towards TDC, as illustrated in Figure 3.15. However, during the start-up the oil pressure is not built-up in the engine, which means that the VVT does not work and the overlap is therefore always the minimum overlap at start-up.

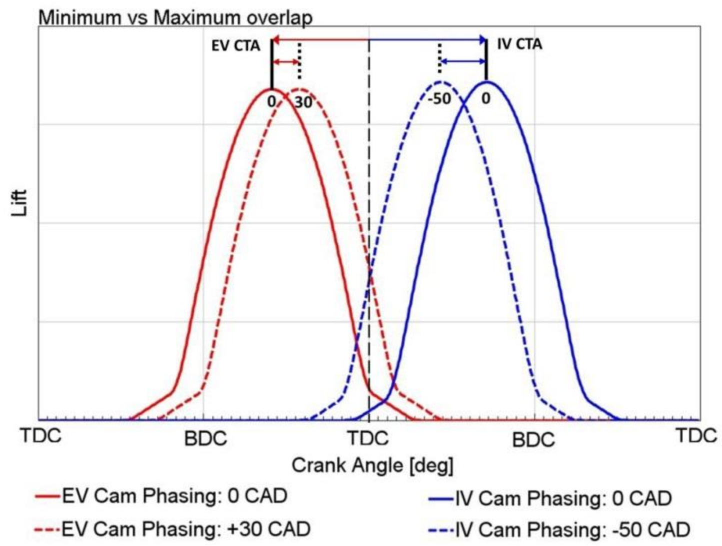

Figure 3.15. Variable valve timing overlap for the gasoline engine.

In the gasoline model it was also important to adjust the valve lash at low engine speeds. In contrary from the diesel engine, where the airflow is not as important due to less lambda restriction, the correct valve lash was needed in order to obtain correct volumetric efficiency and thereby produced torque. Different values for the lash for both intake and exhaust valve where therefore tested in order to obtain highest volumetric efficiency. It was found that the intake valve lash did not affect the volumetric efficiency and was therefore kept as in the steady state model. The exhaust valve lash however, a larger lash affected the volumetric efficiency drastically and was therefore changed in the model.

3.6 Diesel combustion modelling

In this section, the modelling of the combustion and the affecting parameters for the diesel engine, such as combustion model and controlling, is presented.

3.6.1 VCC diesel combustion model

For the diesel model, no calibrated predictive combustion model for steady state operation existed at VCC. The combustion in the steady state model is modelled with user-imposed combustion profiles, where burn rate curves dependent on engine speed and load are added. The burn rate is limited and controlled by the amount of fuel injected to the combustion chamber.

3.6.2 Combustion model implementation

For the transient start-up model the user-imposed combustion profiles were used as combustion model. One representative curve was chosen for the run-up and one for the idle phase, as for the gasoline model.

In reality the diesel engine has a glowplug heating the fuel in order to improve the ignition. However, the combustion model used for simulations was not able to account for the effect from the glowplug. The contribution from the glowplug was considered less important when using predefined combustion profiles and was therefore neglected. The glowplug is however more important to consider when simulating predictive combustion.

The start of combustion was adjusted, to make the timing of the combustion to correlate with test-rig data. The timing was, as for the gasoline combustion model, verified by studying the cylinder pressure. In Figure 3.16 the cylinder pressure from the test-rig data as percent of the peak firing pressure for the diesel engine is presented.

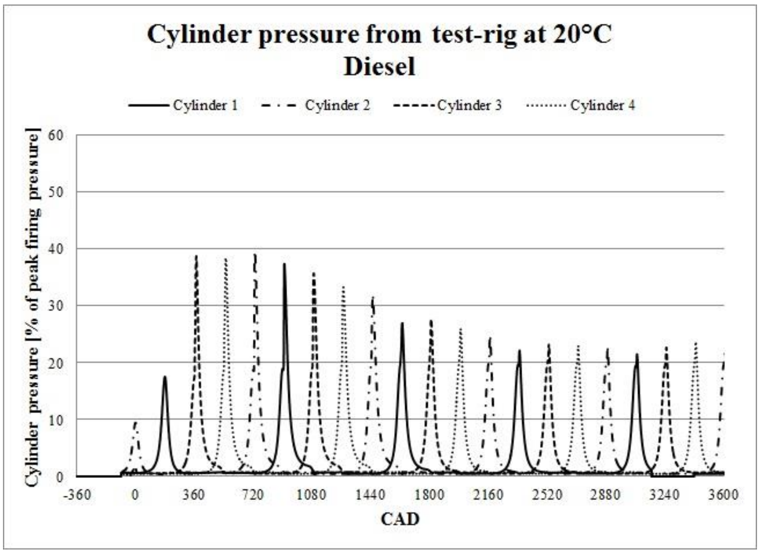

Figure 3.16 Cylinder pressure as percent of the peak firing pressure from test-rig data for the diesel engine.

3.6.3 Controlling

The amount of fuel injected to the cylinder was determined from test-rig data for the run-up phase. When the idle phase was reached a PI controller, similar to the one for the gasoline engine, regulated the amount of injected fuel. A target IMEP was set according to Equation (3.4). During the run-up phase the amount of fuel is extensively increased, in order to prevent start-up failure and to the make a deviation in fuel possible. Furthermore, there are no regulations on emissions at start-up for diesel engines, which allows the large amount of fuel.

4 Results

The engine start-up was shown to be a complex event and good accuracy of the objects in the model was needed due to the highly transient operation. Simplified simulation models were created for both the gasoline and diesel engine at 20°C. The results are presented in this section.

4.1 Starter torque and friction

As previously mentioned in Section 3.4, the start position of the crankshaft was unknown for the measured starter motor data. Therefore the starter motor torque from test-rig data was modified, in order to create even phasing. Besides the starter motor torque, the engine cranking speed was dependent on the friction. When the constant part of FMEP was changed it affected the engine speed and thereby the starter motor torque correlation. Therefore, the modification of the starter torque and determination of the constant part of FMEP had to be done in an iterative process.

In Figure 4.1 and Figure 4.2 the simulated engine speed corresponding to the modified starter motor torque from the cranking test without any combustion taking place, can be viewed for the gasoline engine and diesel engine respectively. As can be seen in the figures, the frequency of the torque and the engine speed correlates which indicates that the starting position of the crankshaft, after the modification, is the same.

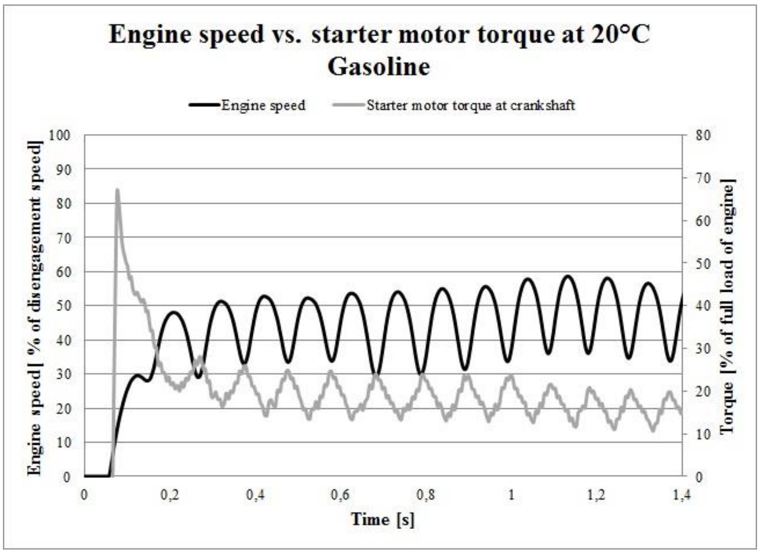

Figure 4.1 Engine speed at cranking and the starter motor torque and after modification for the gasoline engine.

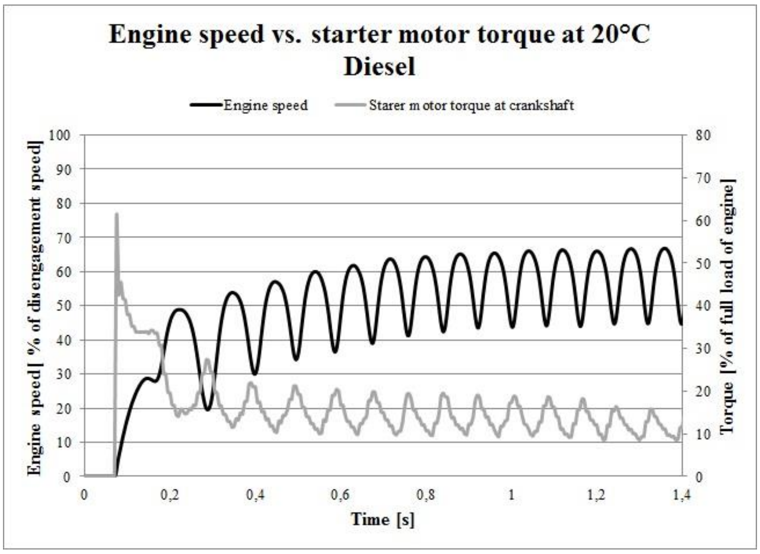

Figure 4.2 Engine speed at cranking and the starter motor torque after modification for the diesel engine.

The final cranking speed compared with the speed from the cranking tests for the gasoline engine and diesel engine respectively, is shown in Figure 4.3 and in Figure 4.4. In the figures, the dashed line is the filtered simulated engine speed used for comparison. The figures present the best achieved results by using the iterative process of determining the starter torque and friction. The constant part of FMEP for the gasoline model found to be 2.77 bar and for the diesel engine 2.4 bar.

It can be seen that the modelled cranking speed for both the gasoline and the diesel engine is slightly higher than the measured speed from the cranking test. However, the largest deviation is shown to be after the starter motor is disengaged in a real start-up with combustion. Therefore the difference was considered small enough to provide realistic results.

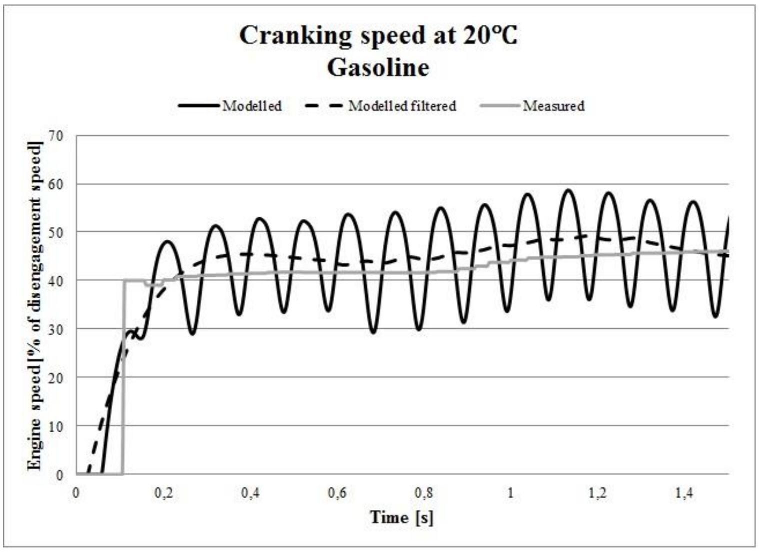

Figure 4.3 Correlation between measured and modelled cranking speed for the gasoline engine.

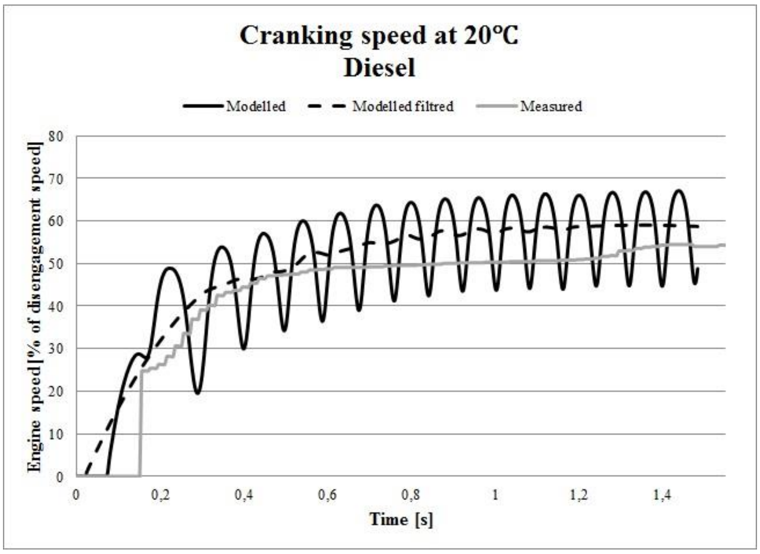

Figure 4.4 Correlation between measured and modelled cranking speed for the diesel engine.

4.2 Combustion

When the adjustment for the cranking phase was done the models were simulated with active combustion. Figure 4.5 shows the cylinder pressure as percentage of peak firing pressure for the gasoline engine from the test-rig data.

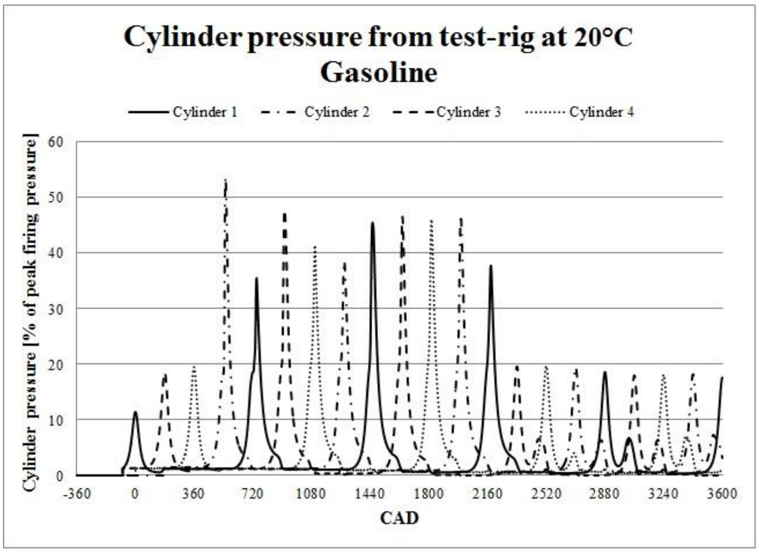

Figure 4.5 Cylinder pressure as percent of peak firing pressure from test-rig data for the gasoline engine during start-up.

In Figure 4.6, the result for the modelled cylinder pressure for the gasoline engine is illustrated. It can be seen in the figure that the modelled cylinder pressure generally matches the cylinder pressure from the test-rig data. By studying the width and the height of the cylinder pressure peaks, the accuracy of the combustion can be determined. If comparing the figures it can be seen that the width is nearly the same for most of the pressure peaks. However, some of the pressure peaks differ in height during the run-up phase. The deviation is probably due to that only one burn rate curve was used for this phase, and in reality there is a variation in burn rate for the different cylinders.

A difference can be also seen for the first cylinder in the third cycle. In the test-rig data the cylinder pressure is high, while for the simulation the combustion has been retarded and the cylinder pressure is low. This is because GT-Power determines which burn rate curve to use in the beginning of each cycle and not for each cylinder. The burn rate curve is therefore changed in the beginning of the cycle closest to the transition to idle.

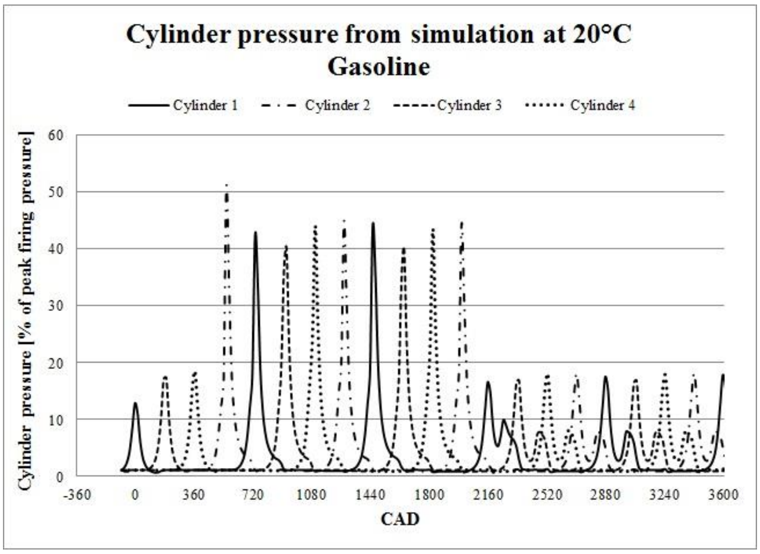

Figure 4.6 Cylinder pressure as percent of peak firing pressure from simulations for the gasoline engine during start-up.

The cylinder pressure for the diesel engine from the test-rig data can be seen in Figure 4.7 and from the simulations in Figure 4.8. If comparing the figures it can be seen that the cylinder pressure for the modelled diesel engine generally matches better than for the gasoline engine. This could imply that the chosen burn rate curve for the diesel engine is more representative for the combustion during the run-up phase. However, the pressure from the simulation is slightly lower during the transition to idle than the pressure from the test-rig data. This is due to how the combustion in the model is controlled, the PI controller, which lowers the amount of injected fuel in order to follow the target IMEP value. The magnitude of the pressure peaks also varies more during the idle phase due to the same reason.

For the diesel engine it can be seen that the first cylinder pressure registered does not occur in the same cylinder for the model and the test-rig data. This is due to that in the test-rig data the engine do not start at the first cylinder in the firing order, which is the case in the model.

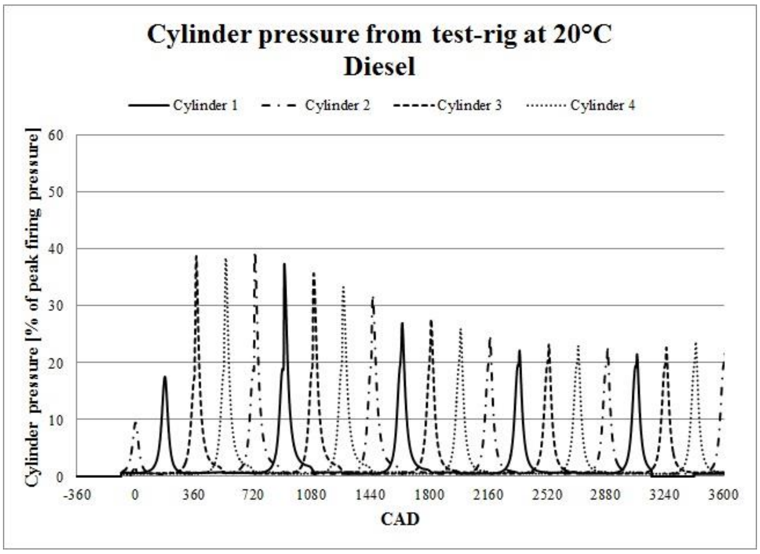

Figure 4.7 Cylinder pressure as percent of peak firing pressure from measurements for the diesel engine during start-up.

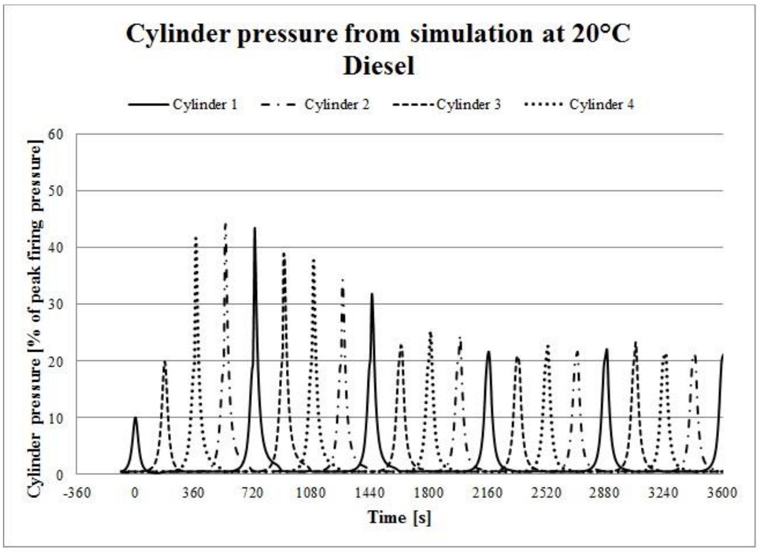

Figure 4.8 Cylinder pressure as percent of peak firing pressure from simulations for the diesel engine during start-up.

4.3 Engine speed

The final studied parameter to verify the results was the engine speed. In Figure 4.9 the unfiltered engine speed as percentage of the idle speed for the gasoline engine can be seen. In the figure a comparison between the modeled engine speed and the measured engine speed is shown.

The curves do not correlate entirely during the run-up phase. The largest deviation can be seen in the local amplitude of the engine speed in the beginning of the phase, which creates an offset for the whole run-up. The smaller local amplitude in the modelled engine speed indicates that the approximation of the DMF inertia was inadequate. This implies that the secondary side and the behavior of the DMF affects the characteristics of the start-up and needs to be modeled correctly.

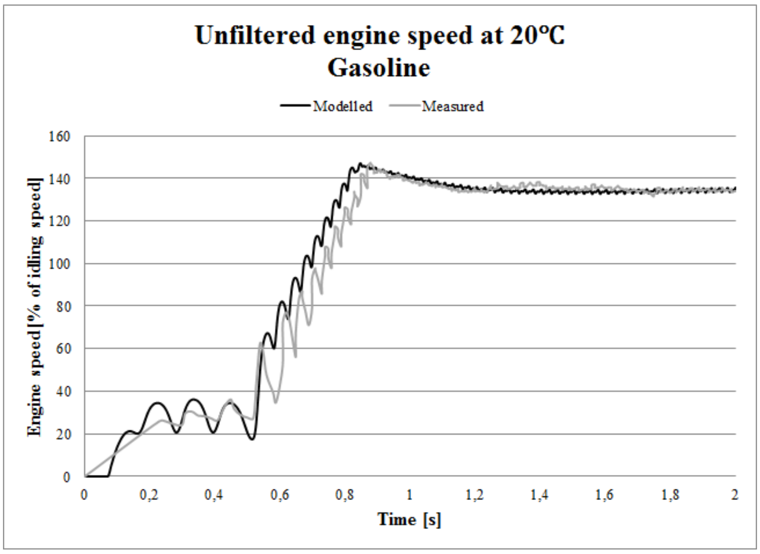

Figure 4.9 Unfiltered engine speed as percent of idle speed from test-rig data and simulations for the gasoline engine.

The offset of the curves can clearly be seen in the filtered engine speed in Figure 4.10, where the gradient of the curves can be compared more easily. It can be seen that besides the offset, the curves have the same gradient and equal behavior. Furthermore, the comparison is only made between the model and one start-up test, and since there is a deviation between different start-up tests. It is possible that the deviation is smaller if comparing the modelled engine speed with other tests.

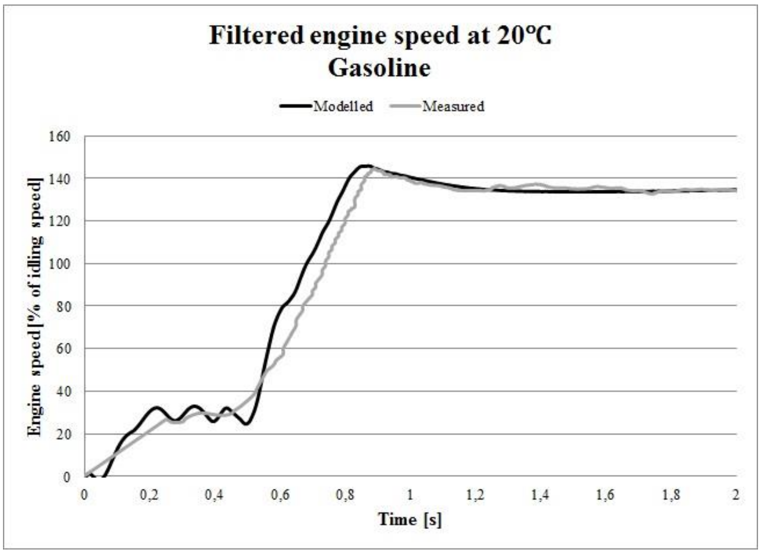

Figure 4.10 Filtered engine speed as percent of idle speed from test-rig data and simulations for the gasoline engine.

In Figure 4.11 the total FMEP for the gasoline engine during the start-up can be viewed. In the figure it can be seen that the total FMEP drops drastically from 3.2 bar to almost 2 bar. This is due to that the constant part of FMEP had to, inaccurately, be reduced to 1.6 bar for the idle phase. If the constant part of FMEP is not reduced, FMEP exceeds the produced IMEP and the controller is not able to maintain the desired idle speed. This shows that the estimated friction is too high during the cranking and run-up phase as well, which most probably is a result of the incorrect inertia. Furthermore, the figure shows that the high initial breakaway friction is not accounted for in the friction model. As presented in Figure 2.7 in Section 2.3.2.1, the highest total FMEP is in reality at the start and then decrease as the lubrication is heated and the oil film is built up. This states that the friction model is inaccurate for this part of the start-up. However, if comparing the figures it can be seen that the amount of friction is in the right magnitude.

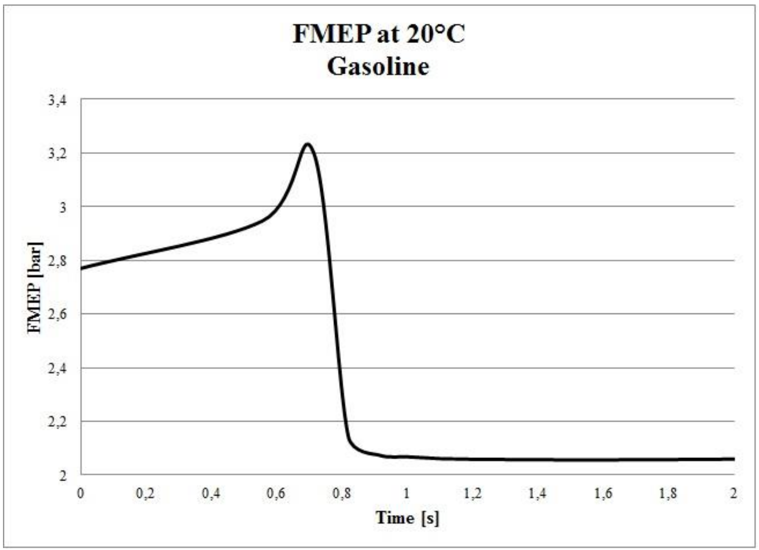

Figure 4.11 Total FMEP for the start-up of the gasoline engine.

In Figure 4.12 the unfiltered engine speed as percent of idle speed for the diesel engine can be seen. The results show, in opposite from the gasoline engine, good correlation between the model and the test-rig data. However, a small deviation can be seen in the filtered engine speed in Figure 4.13 when the engine transitions to idle speed. In contradiction from the pressure at this point, which was too low, the engine speed is too high. This implies that the friction is slightly low, and the adjustments and assumptions for the friction and inertia are partially incorrect and improvements can be done.

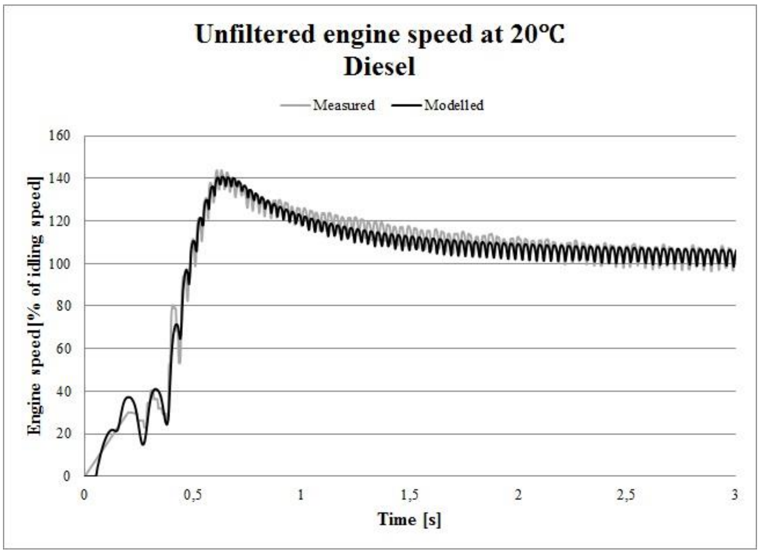

Figure 4.12 Unfiltered engine speed as percent of idle speed for the diesel engine from measurements and from simulation.

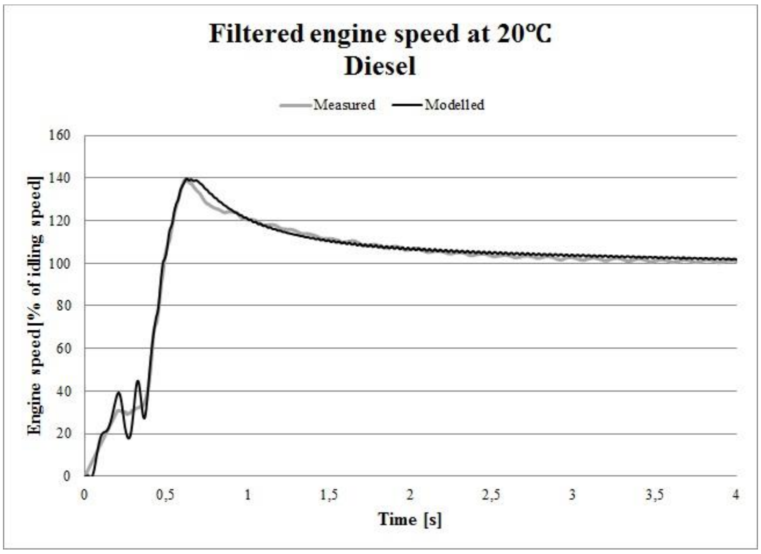

Figure 4.13 Filtered engine speed as percent of idle speed for the diesel engine from measurements and from simulation.

In Figure 4.14 the total FMEP for the diesel engine is shown. It can be seen that the FMEP is more realistic than for the gasoline engine, which is due to that in this model the constant part of FMEP remained the same for the whole start-up. This shows that the friction model is a reasonable approximation for this model at a temperature of 20°C. However, the breakaway friction, as for the gasoline engine, is not accounted for.

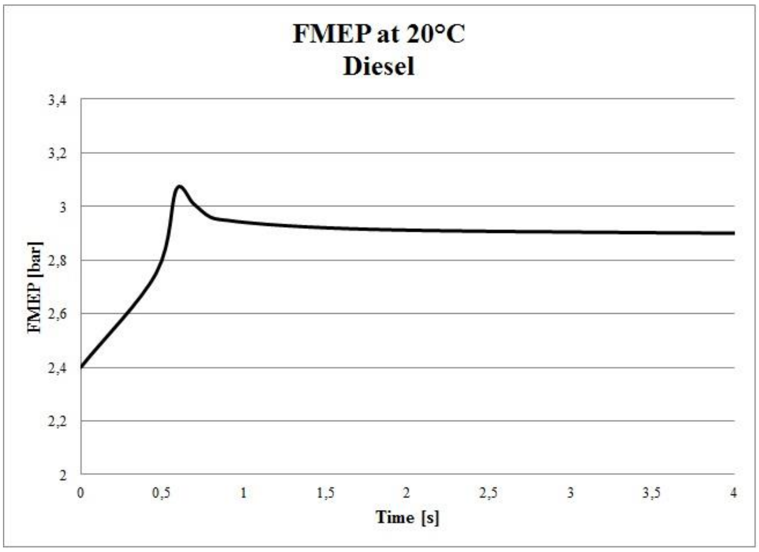

Figure 4.14 Total FMEP for the start-up of the diesel engine.

5 Discussion

Previous studies have aimed to improve the engine start-up or to reduce emissions during the event, but none was found that investigated the accuracy of a simulation model. Therefore this study is a first stage investigation, which provides a basic understanding for what needs to be considered in order to create a predictive simulation model. This section presents the insufficient simplifications and the possible improvements as well as possible use of the model.

5.1 Results

The results show that it is possible to create a model that gives an understanding of the start-up event at 20°C, however due to limited time simplifications were necessary. Even though the results give a good correlation between the model and the test-rig data, the start-up process is complex and the model would need to be improved in order to be predictive. Because of the simplifications the use of the models is limited and it is only valid for 20°C. When trying to simulate for lower temperature the start-up failed during cranking. This is mainly because lower temperatures give higher requirements of the accuracy of the models and with the necessary simplifications the deviation became too large.

5.2 Improvements

The engine start-up models for both the gasoline and diesel engine have some uncertainties due to simplifications. In this section the effects of the simplifications as well as suggestions of improvements are discussed.

5.2.1 Fuel system

One of the simplifications made that affects the accuracy of the model is the duration of the cranking phase, which in the models is determined in order to correlate with test-rig data. In reality it is dependent on the time it takes to build up pressure in the fuel system to enable fuel injection. In order to create a predictive model, the fuel system has to be modelled. If a model of the fuel system was added, the model would be independent of test-rig data for this application.

5.2.2 Combustion

The combustion models appear to be representative for both the gasoline and the diesel engine, since the combustion curves used for the run-up and the idle phase were chosen from several cycles. In order to make the models representative for different temperatures and independent from test-rig data the combustion models needs to be predictive. A predictive model is also necessary in order to study the effect different parameters have on the combustion and burn rate during start-up.

For the gasoline engine both a predictive stratified-charge and homogenous-charge combustion model would be needed, in order for the entire start-up process to be predictive. The change between the combustion models would have to be operated by a controller depending on the engine speed. The controller also needs to adjust the spark timing, which at the idle phase is retarded.

A predictive combustion model is also desirable for the diesel engine. A controller is needed in order to adjust the amount fuel during the run-up phase. In the created model it is determined from test-rig data and is therefore only valid for one temperature.

5.2.3 Friction and moment of inertia

One of the largest uncertainties in the models is the modelling of the friction and the moment of inertia. As previously discussed, GT-Power has no valid friction model for the low temperatures and engine speed at cold start. At the same time, the friction during start-up has a large effect on the behavior and a precise model is of great importance. Furthermore, there were limited possibilities to validate the assumed FMEP used in the model. But, if comparing the used FMEP and the one presented in Figure 2.7 in Section 2.3.2.1, it can be seen that it is in a valid range. The FMEP in the figure is however not for the same oil present in the current study, but could provide an understanding of the magnitude.

In order to create models that predict the friction, a friction model that is valid for low temperatures and engine speeds is needed. The model needs to consider the viscosity and lubrication mode of the oil, and be dependent on the transient engine speed and temperature. The friction model also need to consider the large initial friction that causes the breakaway torque.

The moment of inertia is of great importance for the start-up characteristics. As can be seen in the results, the modelling of the inertia for especially the DMF, but also for the torque converter, affect the total inertia extensively. Both the DMF and the torque converter need to be accurate modelled in order to establish a valid start-up model. In order to capture the complex behavior of the DMF, it should be modelled in detail with the primary side, secondary side and the characteristics of the spring system that connects the two components. The same applies for the torque converter; the effect of the oil needs to be established and modelled in order to get valid results.

5.2.4 Starter motor

A large weakness in both models is the starter motor torque. The torque-generating object provides a torque defined by the user, which in the created models comes from test-rig data. The used torques has constant frequency, which causes a problem that is most significant at lower temperatures. At these temperatures the torque gives a lower constant speed at crank shaft, due to the higher friction, and thereby a larger difference between the cranking speed and the speed when the starter motor disengages. This causes a large difference in frequency between the cranking torque and the torque when the starter motor disengaged, which creates a large offset.

Another significant problem is the unknown starting crankshaft position. If there is a mismatch in the model compared with the cranking test it could result in not enough required torque and the cranking might fail. The modification of the applied torque in this study becomes more difficult at lower temperatures when the amplitude of the torque increases.

Therefore the starter motor needs to be modelled dynamically and in detail. This could be done by using objects in GT-Power for electric motor and battery or by modelling in an external program, for example Simulink. This requires an advanced model of the motor and battery, as well as a control system. The model has to predict the torque dependent on engine speed and temperature, and would require a specification of the starter motor and the battery. The specification would need to include speed, temperature, age and state of charge dependency.

A possibility to improve the starter motor torque, if a dynamic model is not feasible, is that the torque-generating object instead provided a torque with varying frequency dependent on engine speed. This application would still be a simplification, but might get the model to work at lower temperatures. However, the torque applied needs to be collected from tests where the starter motor torque is monitored during a start-up with combustion. There have not been any measurements of this at VCC and thereby a specific test set-up is needed to get accurate results. The torque needs to be measured for a range of temperatures to make the models complete.

5.3 Future applications

If the improvements suggested above are implemented and validated the models can be used in order to study different start-up cases. This section will give suggestions on future use for the models.

One possible future application is to study how different parameters, for example valve timing, lubrication oil, injection strategy and timing and cranking speed, effects the start-up process for both engines. A parameter study could define areas which can improve the start-up, in order to for example increase the volumetric efficiency or reduce friction losses.

The models could also be used in order to examine the emission formation during start-up. More stringent emission regulations will increase the demand of this type of simulations. One important part of reducing emissions is to study how the heating of the catalyst could be improved. Different possibilities for heating the catalyst could be investigated and evaluated by using the model, and could save time and assets compared to real performance tests.

Another future application of the models is to study starts and stops for hybrid electrical vehicles (HEV). However, in order to make the model valid for a hybrid start-up, the model needs to be modified and validated. The characteristics of a hybrid start-up are not the same as for a regular cold start and therefore needs to be considered. For example, the engine usually is warm at a hybrid start since the engine is more frequently turned off and on, which influences the friction. In a HEV, an integrated starter generator (ISG) usually cranks the engine which function might be different than a regular starter motor. The function and engagement strategy of the ISG needs to be considered in order to create a valid model for this application.

6 Conclusions

The study shows that the start-up event is complex, and in order to create a simulation model the behavior of the engine and connected components needs to be considered in detail. The start-up event is strongly dependent on the cranking torque applied by the starter motor, which is dependent on temperature and engine speed. The friction has a large impact on the start-up as well, since it initially is large due to the high oil viscosity. The start-up is also strongly affected by inertia from the DMF and the torque converter, due to the temperature and speed dependency.

It was possible to create a model in GT-Power that simulates the start-up process for a four-cylinder gasoline and diesel engine respectively. The models show trends for 20°C, but in order to get more accurate results and to simulate lower temperatures more advanced models are needed.

In order to create an accurate cold start model, the following improvements to the model are needed:

- Predictive combustion models for the gasoline and diesel engine respectively, as well as stratified combustion model for the gasoline engine, in order to investigate the burn rate and combustion behavior during start-up.
- Model of the fuel system is needed in order to predict the build-up in fuel pressure.
- Complete modelling of the starter motor and battery is necessary in order to provide a torque dependent on temperature and speed.
- A friction model predicting the variations in FMEP for low engine speed and temperatures.
- Detailed modelling of connected components, such as the DMF and the torque converter, in order to obtain correct inertia behavior.

If the models are further developed and adjusted with the proposed changes above, the models could be used to predict the behavior of different start-up events. It would also be possible to use the models in order to investigate methods to improve the start-up and reduce emissions.

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