WELL TO WHEEL EFFICIENCY FOR HEAVY DUTY VEHICLES

Comparison of various biofuels in a long distance lorry and a city bus

Ecotraffic

Well to wheel efficiency for heavy duty vehicles

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Abbreviations, acronyms and glossary

ACEA	European Automobile Manufacturers Association
ASPO	Association for the Study of Peak Oil & Gas
BMEP	Brake Mean Effective Pressure, cycle average pressure on the engine piston
BSFC	Break Specific Fuel Consumption
BTL	Biomass To Liquid
CBG	Compressed biogas
CC	Combined Cycle, gas turbine and steam turbine
CCRT™	Catalyzed Continuously Regenerating Particle filter
Cetane number	The ability of the fuel to ignite in compression ignition engines.
CH_4	Methane
CIDI	Compression Ignition Direct Injection
CNG	Compressed Natural Gas
CO_2	Carbon dioxide
CONCAWE	Conservation of Clean Air and Water in Europe, the oil companies' European association for environment, health and safety in refining and distribution
Cordierite	A common material for monoliths to automotive catalysts and diesel particle filters (DPFs)
CRT™	Continuously Regenerating Particle filter
CSHVR	City-Suburban Heavy Vehicle Route
CSNG	Compressed Synthetic (or Supplementa) Natural Gas
CVS	Constant Volume Sampler/Sampling, a dilution device used for dilution of engine/vehicle exhaust for emission measurements.
DDC	Detroit Diesel Corporation, one of US three biggest manufacturers of heavy-duty engines
DEER	Diesel Engine Emission Reduction Conference (USA)
DFNG	Dual Fuel Natural Gas, an engine capable of running on both gaseous fuel and diesel fuel. A small quantity of diesel fuel has always to be used for ignition, so the engine cannot run on gas only, whereas this is possible with diesel fuel.
DI	Direct Injection
DING	Direct Injection Nautural Gas
DING (GP)	Direct Injection Nautural Gas, ignited by a Glow-Plug
DING (PI)	Direct Injection Nautural Gas, ignited by Pilot Injection of diesel fuel
DISI	Direct Injection Spark Ignition
DME	Dimethyl Ether
DOE	U.S. Department of Energy
DPF	Diesel Particle Filter
ECU	Electronic Control Unit
EEA	European Environment Agency
EEV	Enhanced Environmentally Friendly Vehicle
EGR	Exhaust Gas Recirculation
EN 590	Current European diesel fuel specification
EPA	Environmental Protection Agency (USA)

ESC	European Stationary Cycle
ETC	European Transient Cycle
EtOH	Ethanol
EUCAR	European Council for Automotive R&D, the automotive manufacturer's association for research and development in Europe
EUDC	Extra Urban Driving Cycle
EUI	Electronic Unit Injector
FAEE	Fatty Acid Ethyl Esters
FAME	Fatty Acid Methyl Esters
FT	Fischer-Tropsch, process for producing synthetic fuels from synthesis gas
FTD	Fischer-Tropsch Diesel, synthetic diesel fuel
FTN	Fischer-Tropsch Naphtha, synthetic naphtha
GHG	Greenhouse Gases
GTL	Gas To Liquid
HC	Hydrocarbon
HD	Heavy-Duty
HDE	Heavy-Duty Engine
HDV	Heavy-Duty Vehicle
IANGV	International Association for Natural Gas Vehicles
IGCC	
JAMA	Japan Automobile Manufacturers Association
JRC	The Joint Research Centre, a research based policy support organisation and an integral part of the European Commission.
KAMA	Korea Automobile Manufacturers Association
KFB	Swedish Transportation Research Board. A Swedish governmental organisation that was closed down a couple of years ago.
LB	Lean-Burn, an engine using a lean air/fuel mixture, i.e. less fuel than stoichiometric conditions. This denotation is mostly used for otto engines, although diesel engines also run lean.
LCA	Life Cycle Analysis (or Assessment)
LCI	Life Cycle Inventory
LD	Light-duty
LNG	Liquefied Natural Gas (cryogenic)
LPG	Liquid Petroleum Gas
LSNG	Liquefied Synthetic (or Supplemental) Natural Gas (cryogenic)
M100	Pure methanol fuel (possibly blended with some denaturant and small doses of other additives)
MeOH	Methanol
MIT	Massachusetts Institute of Technology
NEDC	New European Driving Cycle
NG	Natural Gas
NGV	Natural Gas Vehicle
NMHC	Non-Methane Hydrocarbon
NO	Nitrogen monoxide (commonly referred to as nitrogen oxide)
NO_2	Nitrogen dioxide

NO_X	Oxides of nitrogen
NREL	National Renewable Laboratory
PING	Pilot Ignition Nautural Gas
PM	Particulate Matter
PMP	Particulate Measurement Program (the EU programme for developing new measurement methods for particle mass and number)
RME	Rape seed Methyl Ester
SAE	Society of Automotive Engineers
SCR	Selective Catalytic Reduction, a NOX reducing catalyst
SI	Spark Ignition
SING	Spark Ignited Natural Gas
SL	Stockholm Public Transport
SME	Soy Methyl Ester
SNG	Synthetic (or Supplementa) Natural Gas
SRA	Swedish Road Administration
THC	Total HydroCarbon emissions. HydroCarbon (HC) emissions including methane
TTW	Tank-To-Wheel
TWC	Three Way Catalyst
UHDSG	Uppsala Hydrocarbon Depletion Study Group
VTEC	Volvo Technology Corporation
VVT	Variable Valve Timing
WHDC	World Harmonised Duty Cycle
WHSC	World Harmonised Stationary Cycle
WHTC	World Harmonised Transient Cycle
WHVC	World Harmonised Vehicle Cycle
WTT	Well-To-Tank
WTW	Well-To-Wheel

SVENSK SAMMANFATTNING (Swedish Summary)

Det projekt som rapporteras här utfördes i samarbete mellan Ecotraffic och Volvo Technology. De scenarier, drivmedel och drivsystem som studerades fastställdes i huvudsak vid diskussioner mellan parterna. Projektet finansierades av det svenska emissionsforskningsprogrammet (EMFO) som administreras av Vägverket. Projektets huvudsyfte var att generera mer kunskap om systemeffektiviteten (well-to-wheel efficiency) för användningen av biodrivmedel i tunga fordon. De flesta publicerade studierna inom detta område har fokuserats på lätta fordon.

De miljömässiga drivkrafterna som hänför sig till användningen av tunga fordon har de senaste två decennierna för det mesta varit fokuserade på att minska de skadliga avgasemissionerna. Ytterligare reduktioner kommer att göras inom en snar framtid genom införandet av Euro V emissionsgränserna 2009 och den föreslagna Euro VI i ett senare steg. Med Euro VI är det troligt att dieselpartikelfilter (DPF) kommer att introduceras i större skala i EU och följa en liknande utveckling som i USA där partikelfilter används sedan 2007. Dessutom diskuterar Europeiska Kommissionen gränser för antal partiklar som komplement till gränserna för partikelmassa. Ytterligare reduktioner av NO_X emissionerna kommer att göras i USA till 2010, då NO_X gränsvärdet kommer att sänkas med mer än 80 % 5 jämfört med 2007 års nivå. Dessa gränser kommer att kräva användning av mycket effektiv avgasefterbehandling. De föreslagna gränsvärdena i Euro VI för NO_X kommer att kräva reduktioner av samma storleksordning som 2010 förordningen i USA.

Tidsramen för denna studie sattes till 2010+. De första pilotanläggningarna för några av drivmedelsalternativen, som t.ex. cellulosabaserad etanol och syntesgasdrivmedel skulle kunna uppföras inom 5 år. Flera år till måste adderas innan fullskaleanläggningar kan vara igång och ännu några år till innan tekniken kan bli mogen, vilket indikerar en tidsram omkring 2020. Eftersom det vore ganska svårt att förutse utvecklingen av motor- och avgasreningsteknik för en så lång tidsperiod, valdes en något kortare tidsram som kompromiss.

Två fordonstyper valdes i denna studie. En stadsbuss valdes till att representera en typisk fordonskategori som används i tätbefolkade stadskärnor. En fjärrlastbil med en maximal totalvikt på 40 ton valdes att representera ett tungt fordon som används i fjärrtransporter. Euro V emissionsgränsen för NO_X emissioner, dvs. 2 g/kWh sattes som mål för denna emissionskomponent och användning av någon form av partikelfilter förutsattes för att klara framtida partikelkrav. Selektiv katalytisk reduktion (SCR) förutsattes för dieselmotor referensen för att klara NO_X målet men drivmedel som kan klara denna gräns och partikelkravet direkt från motorn skulle inte behöva använda någon annan avgasefterbehandling utöver oxidationskatalysator.

De flesta data för drivmedelsproduktion härleddes från en systemstudie (well-to-wheel) av CONCAWE, EUCAR and EU/JRC, i fortsättningen refererad till som ”JRC studien”. Emellertid utfördes många nya beräkningar av dessa data genom att förutse att all elproduktion utanför produktionsanläggningen för biodrivmedlet skulle ske med samma effektivitet; dvs. en elverkningsgrad på 48,2 %. Processer som inkluderar el-generering och också ”exporterar” el krediterades mot en fristående el-generering som använder denna verkningsgrad.

Användningen av en gemensam process för el-genereringen har en viss inverkan på resultaten för drivmedelsproduktion som härletts från data i JRC rapporten. I några fall, som för biogasproduktion från avfall och gödsel var denna inverkan mycket lite eftersom relativt lite el-energi används. I några fall, t.ex. produktion av drivmedel från biomassa via svartlut, kan signifikanta förbättringar göras. Den största förbättringen kan ses för Fischer-Tropsch diesel (FTD) från svartlut och detta alternativ förbättras ytterligare genom att använda biprodukterna för elgenerering i kraftverk som använder kombicykel. Fall där denna omräkning minskar effektiviteten jämfört med JRC:s data är etanol från vete, halm och ved.

De flesta resultaten för stadsbussen och för fjärrlastbilen är ganska samstämmiga med varandra och, i de fall det finns skillnader, kan de förklaras. Den längre räckvidden som krävs för en fjärrlastbil och det faktum att den mest realistiska jämförelsen är för den energi per ton km som används för det gods som transporteras, ökar ofta skillnaderna mellan alternativen för fjärrbilen. I de fall gasformiga drivmedel övervägs har användning av komprimerad gas för bussen och cryogen gas för lastbilen förutsatts och det finns också skillnader relaterade till dessa teknikval. I allmänhet är vätskeformiga gaser för lastbilen mindre fördelaktiga när jämförelsen görs med andra drivmedel relativt samma jämförelse för drivmedel i bussen. Detta indikerar att användningen av gasformiga drivmedel i stadsbussar skulle vara en option att föredra framför användningen av kryogena drivmedel i fjärrbilar.

Drivmedel som har en högre massa för samma energiinnehåll än andra drivmedel har vanligen en nackdel på grund av detta faktum. Antingen blir fordonet tyngre (som för bussen) eller så minskar lastförmågan (som för lastbilen). Alkoholer är tyngre än dieselolja och i metanolfallet är den relativa skillnaden ungefär en faktor två. Gasformiga och kryogena drivmedel, liksom dimetyleter (DME), som är i flytande tillstånd vid moderata tryck, drabbas också av tyngre tankar än dieselolja.

Inverkan av olika körcykler undersöktes för båda fordonstyperna. I allmänhet var skillnaderna tämligen små men de observerade skillnaderna kunde för det mesta förklaras. Det är möjligt att inverkan av mera ”extrema” körcykler kunde ha varit större än de som studerades här.

Den mest generella slutsatsen som kan dras om energiomvandlare är att ifall den mest effektiva motortypen används, dvs. dieselmotorn, blir skillnaden mellan effektiviteten från tank till hjul (TTW) ganska liten för de flesta drivmedel. För att applicera en sådan teknik är ett betydande utvecklingsarbete nödvändigt. Det är troligt att, för några av drivmedelsalternativen, kanske en så hög verkningsgrad som för den dieseloljedrivna dieselmotorn inte kan uppnås på grund av några inneboende drivmedelsegenskaper eller av praktiska anledningar. Å andra sidan är det också möjligt att några drivmedel kan ha egenskaper som kan utnyttjas för att förbättra effektiviteten till en ännu högre nivå än för basdieselmotorn. Exempel här är den ”inre kylning” som är möjlig med direktinsprutning av alkoholbränslen och de specifika förbränningsegenskaperna för DME. På samma sätt kan också en användning av avgasåterföring (EGR) optimeras för dessa drivmedelsalternativ och möjligen också för andra drivmedel.

EXECUTIVE SUMMARY

The project reported here was made in co-operation between Ecotraffic and Volvo Technology. The scenarios, fuel and driveline studied were largely established in discussions between these two parties. The project was funded by the Swedish emission research programme (EMFO) administered by the Swedish Road Administration. The main scope of the project was to gain more knowledge about well-to-wheel efficiency of the use of biofuels in heavy-duty vehicles. Most of the previously published studies in this field have focussed on light-duty vehicles.

The environmental driving forces related to the use of heavy-duty vehicles have mostly been focussed on reducing the harmful exhaust emissions during the last two decades. Further reductions will be made in the near future through enforcing the Euro V emission limits in 2009 and the proposed Euro VI at a later stage. With Euro VI it is likely that diesel particulate filters (DPFs) will be introduced on a larger scale in the EU following a similar path as in the USA, where DPFs are used since 2007. Furthermore, the European Commission is discussing limits for particle number in addition to limits on particle mass. Further reductions of NO_X emissions will be made in the USA to 2010, when the NO_X limit will be reduced by more than 80 % compared to the 2007 level. These limits will require the use of very effective exhaust aftertreatment. The proposed Euro VI limits for NO_X will require reductions of similar magnitude as the US 2010 regulation.

The timeframe for the study was set to 2010+. The first pilot plants for some of the fuel options, such as e.g. cellulosic ethanol or synthetic gas fuels could be erected within 5 years. More years have to be added before full-scale plants could be in operation and yet another couple of years before this technology would be mature, indicating a timeframe about 2020. As it would be somewhat difficult to project the evolution of engine and aftertreatment technology for such a long period of time, a somewhat shorter timeframe was selected as a compromise.

Two vehicle types were selected in this study. A city bus (or “transit bus” in US English) was selected to represent a typical vehicle category operating in densely populated city centres. A long distance lorry with a maximum weight of 40 tonnes was selected to represent a heavy-duty vehicle operating in long-distance freight. The Euro V emission limit for NO_X emissions, i.e. 2 g/kWh, was set as target for this emission component and the use of some kind of particulate filter was anticipated to meet future particle emission limits. Selective catalytic reduction (SCR) was foreseen for the diesel engine baseline to meet the NO_X target but the fuels that could meet this limit and the particle limit on an engine-out basis would not need to use any aftertreatment but an oxidation catalyst.

Most of the data for fuel production were derived from a well-to-wheel study by CONCAWE, EUCAR and EU/JRC, referred to as the “JRC” study in the following. However, many re-calculations of the data were made by foreseeing that all electricity generation from biomass outside of the fuel production plant would be made with similar efficiency; i.e. an electric efficiency of 48,2 %. Processes that include power generation and also “export” electricity were credited against stand-alone power generation using this efficiency.

The use of a common process for electricity generation has some impact on the results for fuel production derived from data in the JRC report. In some cases, such as for biogas

production from waste and manure, the impact is very small due to a relatively little use of electricity. In some cases, such as, e.g. the fuels produced from biomass via black liquor, a significant improvement can be made. The greatest improvement can be seen for Fischer-Tropsch diesel (FTD) from black liquor and this option is further improved by using the by-products for electricity generation in combined cycle power plants. Cases where this re-calculation decreases the efficiency compared to the JRC data are ethanol from wheat, straw and wood.

Most of the results for the city bus and the long-distance lorry are quite consistent with each other and, in cases where there are differences, they can be explained. The greater range required for the long-distance lorry and the fact that the most realistic comparison is for the energy used per ton of goods transported often increase differences between options for the lorry. In cases where gaseous fuels are considered, the use of compressed gas for the bus and liquefied gas for the lorry has been foreseen and there are also differences related to these choices of technology. In general, the liquefied gases for the lorry are less favourable when compared to other fuels relative to the same comparison for the fuels used in the bus. This indicates that the use of gaseous fuels in city buses would be a preferred option in comparison to the used of cryogenic fuels in long-distance lorries.

Fuels that have a greater mass for the same energy content as other fuels usually have a drawback from this fact. Either the vehicle will be heavier (as for the bus) or the payload will decrease (as for the lorry). Alcohols are heavier than diesel fuel and in the methanol (MeOH) case the relative difference is about a factor of two. Gaseous and cryogenic fuels, as well as dimethyl ether (DME), which is in liquid state under moderate pressures, are also plagued with heavier tanks than diesel fuel.

The impact of various driving cycles was investigated on both vehicle types. In general, the differences were quite small but the observed differences could mostly be explained. It is possible that the impact of more “extreme” driving cycles could have been greater than those studied here.

The most general conclusion that can be drawn about the energy converter is that if the most efficient engine type is used, i.e. the diesel engine, the differences between the tank-to-wheel (TTW) efficiency for most fuels becomes quite small. In order to apply such technology, a considerable development work would be necessary. It is plausible, that for some of the fuel options, as high efficiency as for the diesel-fuelled diesel engine might not be achieved due to some intrinsic fuel properties or practical reasons. On the other hand, it is also possible that some of the fuels could have properties that might be utilised for improving the efficiency to an even higher level than the diesel baseline. Examples here could be the “internal cooling” possible with direct injection of alcohol fuels and the specific combustion properties of DME. Likewise, the utilisation of exhaust gas recirculation (EGR) might also be optimised for these fuel options and possibly, also for other fuels.

1 INTRODUCTION

1.1 Environmental driving forces

The environmental driving forces related to the use of heavy-duty vehicles have mostly been focussed on reducing the harmful exhaust emissions during the last two decades. Thus, considerable reductions of the regulated emission components such as, CO, HC, NO_X and particulate matter have been achieved. Further reductions will be made in the near future through enforcing the Euro V emission level. Furthermore, the future Euro VI emission limits are currently under discussion.

In the USA, the particulate limit in the EPA emission regulation for 2007 and later has practically been forcing the engine manufacturers to use diesel particulate filters (DPF). Further reductions of NO_X emissions will be made in 2010, when the NO_X limit will be reduced by a factor of 5 compared to the level today. These limits will require the use of very effective exhaust aftertreatment. It is likely that the upcoming Euro VI NO_X limits will require reductions of a similar magnitude. Furthermore, the European Commission is discussing limits for particle number in addition to the current limits on particle mass. Intensive developments of new measurement methods for particle number (and mass) are currently under way to accomplish this. For light-duty vehicles, such emission limits have already been proposed and the regulations are scheduled for approval.

When emission levels corresponding to US 2010 and Euro VI will be reached – and at the time most of the older vehicle fleet has been replaced (presumably in about a 2020 to 2030 timeframe) – it is likely that the impact on the environment and local air quality will be considerably reduced. Then, other environmental driving forces than exhaust emissions will most likely be given a higher priority. This development has already been indicated by several automotive manufacturers, e.g. Volkswagen [1]. The diagram in **Figure 1** illustrates this potential development.

In its “vision 2030” report for future use of biofuels, the European Union has identified the long-term potential of biofuels use in the EU [2]. In this report, the potential and importance of the so-called second generation biofuels was specially highlighted. Furthermore, the need for high efficiency in the life cycle and effective land use was stressed. In a separate EU project, “biofuels technology platform” (BiofuelsTP), the work of implementing the strategy and identifying the need for research and development in this field has continued. In January 2008, the first report from this work was published [3]. A discussion about increasing the blending level in petrol and diesel fuels is going on. Blending levels for ethanol (EtOH) and FAME of 10 % respectively have been proposed but there has also been opposition to such increases due to potential problems with engines and aftertreatment.

Recently (i.e. early 2008) considerable criticism has been raised by several environmental organisations, but also by the governments in several member states, about the use of biofuels. Mostly, this criticism has been targeted towards the first generation of biofuels, including the import of such or feedstock (e.g. palm oil) or fuels from countries outside the EU. Poor sustainability and bad social conditions for the workers in the industry abroad has been two major issues for debate. This illustrates the crucial importance of fuel production and how important it will be in the future to set up criteria and regulations to cover these aspects.

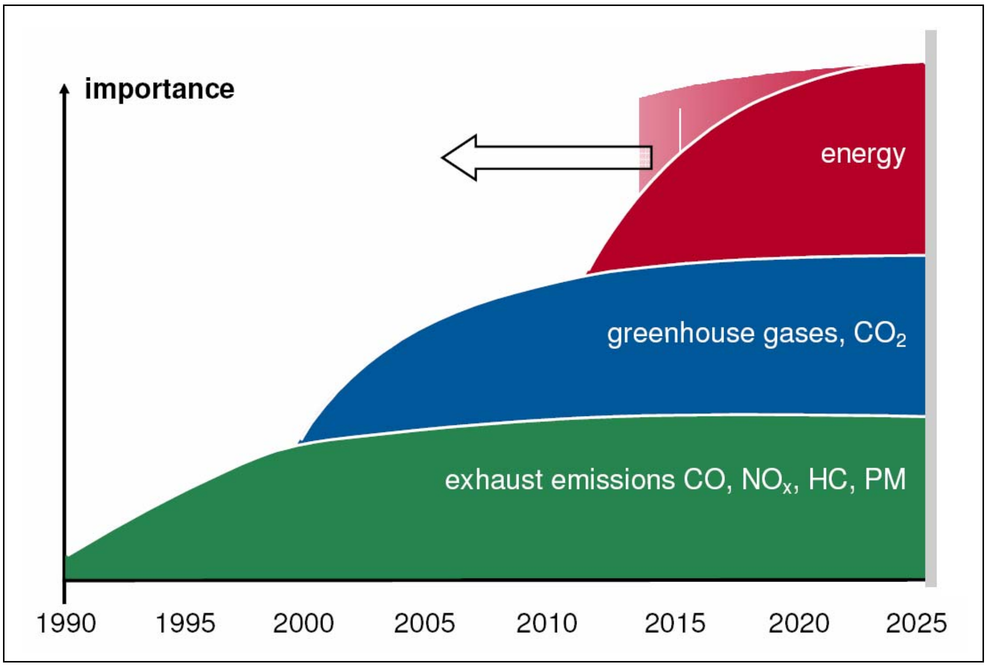

Figure 1. *Environmental driving forces (according to VW)*

For light-duty vehicles, there has already been a considerable focus on CO_2 emissions and fuel consumption. The car manufacturers of Europe, Japan and Korea (represented by their organisations ACEA, JAMA and KAMA) have adopted a voluntary reduction in CO_2 emissions by 25 % from the base year of 1995 to 2008 (ACEA) or 2009 (JAMA and KAMA) [4]. Although it does not seem likely that the target will be fully achieved, a relative reduction of some 20 % still is considerable. Legislative measures and incentives to achieve further reductions of CO_2 emissions in the 2012-2015 timeframe are currently under discussion in the EU. For owners/operators of heavy-duty vehicles, the fuel economy has always been in focus, so similar measures as for light-duty vehicles have not been motivated. However, a discussion has also stated in this field. Apparently, heavy-duty vehicles are very different from each other in comparison to the relatively homogenous group of vehicles as passenger cars. Thus, it is very difficult to determine and introduce measures to reduce the fuel consumption for heavy-duty vehicles. Ecotraffic has previously participated in several such studies where potential options to limit CO_2 emissions from heavy-duty vehicles have been investigated [5, 6].

During the last two years, crude oil prices have reached comparatively high peaks, as high and periodically even higher, than during the oil crisis in the 1970’s. One possible reason for the higher prices during the last two years is that the oil production and refining can hardly keep up with the demand any more. Oil production will soon have reached its peak. It is appropriate to discuss this matter in little more detail below.

The Association for the Study of Peak Oil & Gas (ASPO) is an organisation with the aim of studying the forthcoming peak in oil production. ASPO has also concentrated on to resolve some of the issues regarding statistical data on oil reserves. The Uppsala Hydrocarbon Depletion Study Group (UHDSG) at the Uppsala University in Sweden is conducting research in this field. An important conclusion by the UHDSG group of researchers has been that the peak in oil production is not far ahead. In a recent publication by one researcher active in the UHDSG group, the importance of giant oil fields has been investigated [7]. Results from this study are shown in **Figure 2**. The contribution of those oil fields is striking; i.e. over 60 % of the production in 2005 and about 65 % of the global ultimate recoverable reserve (URR) come from those fields. In the report mentioned, the peak oil was forecasted to be reached between 2008 (worst-case scenario) and 2018 (best case scenario). However, there is not so much consensus about this conclusion. Among researchers, the oil industry and other stakeholders, there is currently an open debate about if the peak may already have been reached or if this is not going to happen until a couple of decades in the future. The mentioned report also covers unconventional oil resources such as the heavy oil from the Orinoco belt in Venezuela and oil sands in Canada. Despite the large resources in those cases, their contribution will not be great enough to offset the decline from conventional oil sources.

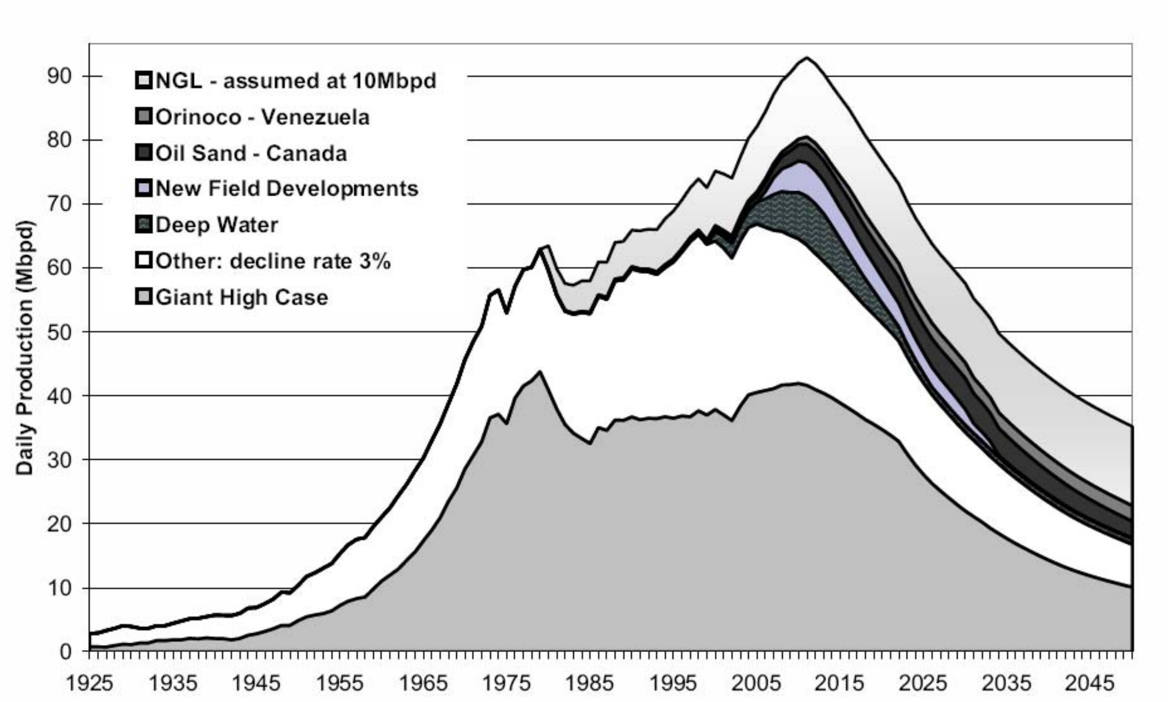

Figure 2. *ASPO's hydrocarbon production forecast*

The peak in natural gas production (not shown in **Figure 2**) is not much further away than that for crude oil. The shortage in natural gas supply already experienced occasionally in some areas around the world during the recent years is an indication of this problem. Sine the domestic resources of natural gas in Europe will not be sufficient to supply the need in the future; several projects for natural gas pipelines from Russia are currently under discussion. Certainly, there is some room for an increased use of natural gas on the local level in many EU member states but most of this supply would probably have to come from remote natural gas (RNG). To what extent the transportation sector should be included in further utilization of natural gas remains to be seen.

In summary, the short overview above has shown that, on a longer term, new energy sources have to be developed to supplement oil and gas. Biomass cannot substitute all this energy supply but it could become a very important contribution for the next decades.

The problems of energy supply and emissions of greenhouse gases have prompted the EU to propose strategies and regulations to introduce biofuels. The first directive in this field, the so-called biofuels directive, was introduced in 2003 [8][1]. This directive is currently under review and a future directive in this field is being discussed. Several reports and other documents have been made in the EU as a basis for this strategy. One of the most important of them has been

[1] Numbers in brackets designate references that are listed in the reference section at the end of the report.

the biofuels vision for 2030 [2]. In the report, it was concluded that there is a technical potential to cover between 27 and 48 % of the road transport fuel needs in the EU in 2030 if all biomass would be dedicated to biofuels production. This would be equal to 360 Mtoe. The report suggests a vision of 25 % substitution in 2030.

1.2 Emission legislation, driving cycles

Whereas the Euro 5[2] and 6 emission limits for light-duty vehicles have been adopted, the discussion regarding the Euro VI limits for heavy-duty engines has been somewhat lagging behind. One reason is that the Euro V for heavy-duty engines is introduced later than Euro 4 was introduced for light-duty vehicles. Another factor might be that the aftertreatment technology for meeting even more stringent NO_X emissions on heavy-duty vehicles has not been fully developed yet. After the introduction of SCR catalysts with urea (Adblue®) as a NO_X reducing agent this condition has now changed and further development of this technology could enable even stricter emission limits.

On July 16, 2007, the European Commission launched a public consultation on the planned Euro VI emission regulation for heavy-duty truck and bus engines. The time limit for comments on the proposal for Euro VI was set to 5 September 2007. An EU sponsored report, where stakeholder responses has been compiled, provides more background on this subject [9]. The Commission was asking for comments from stakeholders in this field on the proposed limits according to four different scenarios. The emission limits according to these scenarios are listed in **Table 1**.

***Table 1**. Euro VI scenarios for engines in heavy-duty vehicles*

Emission	**A**		**B**		**C**		**D**	
Engine type	CI[2)]	PI[3)]	CI[2)]	PI[3)]	CI[2)]	PI[3)]	CI[2)]	PI[3)]
CO (g/kWh)	4,0	4,0	4,0	3,0	4,0	3,0	4,0	3,0
THC (g/kWh)	0,16	0,66	0,55	1,05	0,55	1,05	0,55	1,05
NO_X (g/kWh)	0,4	0,4	0,2	2,0	1,0	2,0	0,5	1,0
PM (g/kWh)	0,01	0,01	0,02	0,02	0,015	0,02	0,015	0,01
NH_3[1)] (ppm)	10	10	10	10	10	10	10	10
CO_2 increase	2-3%		5-6%		Neut.		Neut.	

Notes:

1) To be applicable to vehicles using SCR (Selective Catalytic Reduction) aftertreatment technology
2) CI: Compression Ignition. Engines fuelled with diesel and ethanol
3) PI: Positive Ignition. Engines fuelled with natural gas (NG) and liquefied petroleum gas (LPG)
4) Anticipated additional CO_2 emissions resulting from the various scenarios

The impact of the proposal could be summarised as:

2 In the literature, both Roman and Arabic numerals are used for European emission limits. In this report, Roman numerals are used when referencing to European standards for heavy-duty engines (Euro I, II,…), and Arabic numerals are reserved for light-duty vehicle standards (Euro 1, 2,…).

- Scenario A has limit values comparable to the US 2010 standards. Based on the similarity between scenario A and US 2010, the compliance could be achieved by using cooled exhaust gas recirculation (EGR), a urea-SCR catalysts and a particle filter. The estimated increase in fuel consumption and CO_2 emissions for scenario A would be 2 to 3 %.
- Scenario B is the strictest of all in terms of NO_X emissions. Higher rates of cooled EGR than scenario A would be needed. In order to achieve such a high ratio of EGR, scenario B requires an improved cooling system in the vehicle. As a result, higher CO_2 emissions of around 5 to 6 % are anticipated in this case.
- Scenario C is the least stringent in terms of NO_X emissions and thus, it would have no negative impact in terms of fuel consumption and CO_2.
- Scenario D is somewhat more relaxed than scenario A and, "to a certain extent", equivalent to the US standards, without the associated fuel economy penalty.

As indicated by the forecasted impact on CO_2 emissions from the various scenarios, the Commission was interested in finding the best trade-off between CO_2 and pollutant emissions. Stricter limits will lead to higher CO_2 emissions and vice versa. Stakeholders were asked for their view on such issues.

There were two possible options for the future emission legislation in the EU. One would be that a proposal for Euro VI would be made before the end of 2007. The other would be that both Euro VI and VII are prepared in a single proposal even if this would mean that the proposal and entry into force would be delayed. In December 21, 2008, a proposal was made for Euro VI by the European Commission [10]. The proposal by the Commission largely corresponds to scenario "A" in **Table 1**. The mentioned proposal is shown in **Table 2**.

***Table 2**. Euro VI emission limits proposed by the European Commission*

Test Cycle (Ignition)	Emission limit values							
	CO (mg/kWh)	THC (mg/kWh)	NMHC (mg/kWh)	CH_4 (mg/kWh)	NO_X[4] (mg/kWh)	NH_3 (ppm)	PM (mg/kWh)	PN (#/kWh)
ESC (CI[1])	1500	130			400	10	10	
ETC (CI[1])	4000	160			400	10	10	
ETC (PI[2])	4000		160	500	400	10	10	
WHSC[3]								
WHTC[3]								

Notes:

1) PI: Positive Ignition
2) CI: Compression Ignition
3) The limit values relating to World Harmonised Stationary Cycle (WHSC) and World Harmonised Transient Cycle (WHTC) will be introduced, at a later stage, once correlation factors with respect to the current cycles (ESC and ETC) have been established
4) The admissible level of NO_2 component in the NO_X limit value may be defined at a later stage
5) PN: Particle Number emissions. A number standard is to be defined at a later stage.

On 15 July 2008, a report by the Environment Committee of the European Parliament backed an easing of the proposed NO_X limits from 0,4 g/kWh to 0,5 g/kWh, which was voted on by the Members of European Parliament (MEP) [11]. The MEPs also want the new Euro VI regulation emission limits to apply de facto from 1 January 2014, three months earlier than in the proposal from the Commission. The Commission was asked to adopt the associated technical regulation by the end of 2009. When the present report was finalised by the end of August 2008, the final decision on Euro VI emission limits had not been made.

The current Euro VI proposal has no limits for particle number emissions. The on-going work on developing measurement methods for measuring particle number emissions on heavy-duty engines has been considerably delayed. Presumably, this is a reason why no proposal has been made in this case. This does not indicate that no limits will be enforced in the future. Limits may be proposed later or else, the regulation could be amended with limits on particle number.

Regarding the objective of this study, i.e. WTW efficiency for various biofuels, one could note that the following fuels are mentioned explicitly in **Table 1**:

1. diesel fuel;
2. natural gas (NG);
3. liquefied petroleum gas (LPG) and;
4. ethanol

No one of the scenarios in **Table 1** is fully fuel neutral, i.e. there are differences between the limits for various fuels. The only exception is that similar levels always seem to apply for diesel fuel and ethanol. The THC limit is more relaxed for positive ignition engines fuelled by NG and LPG, possibly reflecting the lower health impact of THC (comprising mostly methane in the NG case) from such engines in comparison to diesel engines and the fact that limiting THC is more difficult on positive ignition engines. In some scenarios in **Table 1**, there was a slight difference in PM emissions between diesel and positive ignition engines. In all cases but scenario A, there were considerably relaxed limits for NO_X emissions from positive ignition engines in comparison to diesel engines. In the latest proposal, the NO_X level is similar for all engine technologies.

1.3 Engine and aftertreatment technology

Current technology to meet the NO_X limit in Euro IV is based on SCR aftertreatment in most cases. A few manufacturers are using cooled EGR instead of SCR. The use of SCR in combination with high-pressure injection and advanced injection timings, which can be used with NO_X aftertreatment, is sufficient to reduce the particulate level below the limit. In case of the EGR option, an open flow particle filter is generally used to reduce the particulate level, since EGR increases the particulate emissions. With further development, both options are likely to be able to meet the Euro V limits. Improved SCR catalysts and increased use of the Adblue® reducing agent can reduce the NO_X emissions. In fact, many of the engines with SCR manufactured so far, have already fulfilled Euro V. Increased use of EGR in combination with improvements of the combustion system (e.g. higher injection pressure) could also enable this option to meet the limits. Neither of the solutions is likely to need a closed flow particulate filter to meet the particulate limit but combinations of oxidation catalysts (DOC) and open flow particulate filters are likely to be seen on more engines than today. A more thorough compilation of the potential solutions for reducing emissions to meet future EU emission limits has been made by Gense et al. [9].

As indicated above, improvements of both the engine-out emission level and exhaust aftertreatment must be made if the US 2010 emission limits will be met. In contrast to Euro V, a combination of all the “best” options, as indicated above, must be used to meet the US 2010 emission limits. The US EPA is continuously monitoring the progress in this field, which has been reported in a couple of publications [12, 13]. Due to the somewhat stricter particulate limit in combination with the much stricter NO_X limit compared to Euro V, the US 2010 engines will have to use more efficient, i.e. closed flow, particulate filters. Both EGR and some kind of NO_X aftertreatment must be used. The use of SCR was debated for a long time but was finally been accepted by the US EPA as a possible solution. Another aftertreatment technology much in focus in the USA has been the so-called NO_X adsorber catalyst (NAC)[3]. In fact, the first engine certified for the US 2010 is using this kind of catalyst [14].

An article in the German journal MTZ summarised recent advancements regarding aftertreatment systems for Euro V and US 2007 [15]. Integrated active systems for DPF regeneration are used to meet US 2007 and JP05 emission limits. The introduction of fuel/hydrocarbons after the turbocharger turbine, and an oxidation catalyst in front of the DPF are used for regeneration. Independent on the regeneration strategy used, this event is controlled by the electronic control unit (ECU) via sensors for, e.g. temperature and differential pressure. Cordierite is the most commonly used filter material for heavy-duty engines. The filtration efficiency is high and yields PM levels far below the US 2007 limit of 13,4 mg/kWh (10 mg/bhp-hr). To meet the US 2010 and Euro VI NO_X emission limits, both SCR catalysts using Adblue urea reducing additive and NO_X adsorber catalysts are evaluated. NO_X reduction efficiencies in the order of 80-90 % might be needed. DPFs will be continuously developed for reduced pressure drop, increased thermal stability and durability, asymmetric filter design for increased ash capacity and filtration efficiencies from 90 to 95 % for all sizes of particles. For oxidation catalysts and NO_X catalysts higher cell density and thinner walls are desired, since they combine low pressure drop with high catalyst efficiency.

An interesting development of the EGR system both for light and heavy-duty engines would be to use the co-called “low-pressure” or “long-route” EGR system. In this system, exhaust gases are routed, via an EGR cooler, from after the turbine to before the compressor. A long-route system must use a DPF to avoid fouling and damage of the compressor in the turbocharger. If a DPF must be used anyway to meet the emission limits, no additional cost need to be taken into consideration for this component. So-called “clean burning” fuels that are not plagued with the soot formation under diffusion combustion, such as conventional diesel oil in diesel engines, would need no particulate filter, which makes this option particularly interesting for these fuels. Fuels that contain a certain percentage of oxygen can burn without any soot formation in diffusion combustion. Examples of such fuels are (but not limited to): ethanol, methanol and dimethyl ether (DME). For the last couple of years, the long-route EGR option has been used as retrofit aftertreatment system in combination with a particulate filter by the Swedish development company STT Emtec [16]. No similar system has yet been introduced by an engine manufacturer but the option is nevertheless of great interest.

1.4 Fuel quality

Ever since emission limits were first introduced for heavy-duty engines some two decades ago, fuel quality has also been on debate. First, a reduction of the sulphur level was of interest in order to reduce the particulate emissions. A tighter fuel specification in general has been a demand from the automotive and engine industry. These requirements were manifested in a so-called fuel charter by this industry, which has been updated a couple of times since the first version [17].

The latest European diesel fuel quality (EN 590) has been used as reference fuel in this study. This approach differs from the previous WTW study from Ecotraffic on light-duty cars where

[3] Alternative denotations are: storage catalyst (NSC) or NO_X trap catalyst (NTC).

the Swedish Environmental Class 1 (EC1) diesel fuel was used. Both fuels do (with reference to the latest version of EN 590) have a sulphur limit of 10 ppm_w, which is a necessary condition for using the latest available aftertreatment technology. Very low sulphur content is the most important parameter to enable the use of such emission control. Other fuel parameters, such as e.g. polycyclic aromatic hydrocarbon content is of importance regarding unregulated emission components that could cause health hazards. This was the rationale behind the introduction of the EC1 fuel in the early 1990's. With exhaust modern aftertreatment, such fuel properties are probably of less importance than previously and thus, the EC1 fuel has recently been questioned by some oil companies in Sweden. The debate about this issue is still going on.

As indicated previously, the scope of this study was not to study diesel fuel quality per se. Since the latest fuel data regarding well-to-wheel efficiency are available for EU diesel fuel only, the choice of this fuel quality was obvious.

1.5 Background to and scope of the work in this study

This study was funded by the Swedish emission research programme (EMFO) administered by the Swedish Road Administration. Initial discussions were made between Ecotraffic and Volvo Technology and a first proposal was made based on those discussions. Scania also joined the project but left the project later so Volvo Technology assumed their responsibilities.

Well-to-wheel efficiency is important for engine and vehicle manufacturers in order to choose a strategy for future fuels and drivelines. Thus, studies in this field are of interest for these manufacturers. On the national level, it is of interest to gain more information about this topic as basis for future energy strategies and introduction of new biofuels. The main scope of the project was to gain more knowledge about well-to-wheel efficiency from the use of biofuels in heavy-duty vehicles. Most of the previously published studies in this field have focussed on light-duty vehicles.

The main focus in this study has been on the vehicle drivelines. The scope was to compare the impact of various engine options for the fuel studies. Most of the data on fuel production was derived from the literature.

The scenarios, fuel and driveline combinations studied were largely established in discussions between Ecotraffic and Volvo Technology. Most of the input data for drivetrains and vehicles was provided by Volvo and considerable contributions were also made by Volvo in the field of fuel production.

Data processing and vehicle simulations were carried out by Ecotraffic, who also prepared the final report but with valuable contributions from Volvo.

2 BACKGROUND

2.1 Previous WTW studies by Ecotraffic

Lifecycle analysis (LCA) and well-to-wheel studies attracted attention first in the early 1990's. Life cycle analysis and inventories (LCI) were developed for other purposes and a new interest was focussed on alternative fuels in general.

Staff at Ecotraffic conducted one of the first comprehensive well-to-wheel studies, "Life of fuels" (LoF), in the early 1990's [18]. One of the authors of the "LoF" report, Åke Brandberg, has – although retired since several years ago – participated in collecting material and analysing conditions in one of the appendices in this study. Other work in this field from Ecotraffic as been a WTW study conducted in 2001 [19] and a "follow up" on that study one year later with the focus on strategies to introduce biofuels [20]. Part of the fuel supply data in the WTW study was based in the previous LoF report but updates were made when necessary.

2.2 Other WTW studies

Numerous WTW studies and life cycle analysis from the USA could be mentioned. Probably the most interesting are some of the studies published by researchers at ANL, where also the GREET model for lifecycle analysis has been developed [21]. Some of the studies worth mentioning is the comprehensive well-to-wheel study from 2001 co-ordinated by GM and with contribution from ANL and three oil companies [22]. The GREET model was used in this work. An update of this work was made in 2005 [23] and besides that, numerous publications have been made about the GREET model or the use of it as a tool in other studies. Other studies of significant interest have been published by Massachusetts Institute of Technology (MIT). The first study among these focussed on cars and fuels in 2020 [24]. A recent update of that study with a time horizon of 2035 instead of the previous one was published in July 2008 [25]. In their studies, MIT has focussed mostly on the well-to-tank part of the WTW chain. Although the studies carried out the USA often are very comprehensive, the conditions in the USA differ considerably from those in Europe. Therefore, the US studies have not been of primary interest here.

In a parallel work to the study conducted by GM in the USA, GM in Europe carried out a similar study, which involved the German technology development company L-B-Systemtechnik GmbH (LBST) for the WTT part of the work [26]. The most comprehensive study of well-to-wheel efficiency on European basis has been conducted by CONCAWE, EUCAR and EC/JRC [27]. In the following, this study is referred to as the "JRC" study. As discussed later in the present report, the JRC study has been used as main input for the tank-to-wheel here. This work was partly built on the work by LBST in the mentioned GM Europe WTW study.

Certainly, a number of other studies than those above could be mentioned but this would have been beyond the scope of the present study, which was not focussed on a literature survey.

2.3 The use of alternative fuels

In the EU, Sweden has been a forerunner in using alternative fuels and biofuels in particular. This is with one exception, i.e. biodiesel, which is the mostly used biofuel in the EU, but a fuel that has not received as much attention in Sweden. Instead, ethanol and biogas has mostly been used in heavy-duty vehicles. These vehicles are using dedicated engines for neat fuels.

At the end of the 1980's and the beginning of the 1990's, alternative fuels were introduced in Sweden as a measure to reduce exhaust emissions. Most of the efforts have been concentrated on ethanol and CNG/biogas. Scania has supplied the lion's share of the ethanol-fuelled buses and Volvo has sold most of the gaseous-fuelled buses. Ethanol was first tested in the cities of Örnsköldsvik and Stockholm in the late 1980's. In the early 1990's the Stockholm Public Transport (SL) purchased the first larger fleet of 30 ethanol-fuelled buses [28]. A couple of years later, CNG was introduced in buses in Gothenburg. Following the successful use of the first ethanol fleets, this fuel was later also introduced in several other cities in Sweden but the majority of the ethanol buses has remained in Stockholm. CNG has primarily been used in the southern part of Sweden and on the West Coast. Since Sweden does not have a large pipeline grid for natural gas, biogas produced via digestion of sewage sludge and waste provided a complement to natural gas in several cities. This enabled the use of gaseous-fuelled buses in municipalities outside of the area that is covered by the natural gas grid. No official statistics about the use of alternative fuels in Sweden is listed. To give some perspective, data from the Internet site about "clean vehicles" can be used [29]. In 2005, there were close to 400 ethanol-fuelled city buses (370) and about 900 heavy-duty buses and trucks fuelled by gaseous fuels. The number of gaseous-fuelled city buses has been about equal to the number of ethanol-fuelled buses over the years. Most of the other gaseous-fuelled heavy-duty vehicles are rather small, i.e. just above the weight limit (3,5 ton) for this category of vehicles. Since these data were collected, a number of buses and other vehicles have been added in several open common procurement campaigns by major bus operators and fleet owners.

The main purpose of using alternative fuels was initially to reduce exhaust emissions. Later, the focus has shifted somewhat towards emissions of greenhouse gases (GHG). It could be noted that the mentioned fuels have some problems today regarding GHG emissions. The life-cycle assessment for ethanol produced from surplus wine is not very favourable and natural gas is a fossil fuel. Other feedstock options for ethanol production and the use of biogas instead of natural gas improve the GHG emissions considerably.

In total, alternative fuels are now used by more than 10% of the city bus population in Sweden. The *use* of alternative fuels in city buses and the *fuel production* have been supported by Swedish Governmental Agencies through several research, development and deployment programmes. For example, research and development in production of ethanol from cellulosic biomass has been supported by the Swedish Energy Agency. A pilot plant is now under construction. However, most of the ethanol used in Sweden is still imported. As in the ethanol case, biogas production has also received support from the Government in research and development as well as support for the investment in production facilities and refuelling infrastructure.

In the autumn of 2002, the Swedish bus manufacturer Scania declared that they would discontinue the manufacture of ethanol buses. Later, this decision was revised after renewed interest from bus fleet operators and municipalities. However, Scania is still the only heavy-duty engine manufacturer that offers ethanol buses in the EU. A new ethanol engine family is currently under development that should be able to fulfil the Euro V/EEV emission limits. New improved gaseous-fuelled bus engines usually have emission levels below the Euro IV, V or EEV emission limits. Closed-loop air/fuel control and improved oxidation catalysts are two of the main improvements on these engines.

3 METHODOLOGY

3.1 Previous work in this area

The methodology for conducting lifecycle analysis has been defined in the ISO norm 14040 [30]. Much of the work reported here draws on previous studies in this area. The methodology used in the present study does not strictly follow the ISO norm, since many of the studies which the present study was based on does not follow this strictly either. Therefore, it is better to characterise the work reported here as a well-to-wheel study, where the definitions are not exactly similar to those in lifecycle analysis.

3.2 Literature survey

No direct literature survey has been made here although much input data had to be collected from the literature. The global mobility database (GMD) by the Society of Automotive Engineers (SAE) has been used extensively whenever there has been need for data. Other sources already known by the author and his partners, as well as search on the Internet, has been used to find missing data. The literature cited is collected in the reference list at the end of the report.

3.3 Conditions

3.3.1 Time horizon

The time horizon for this study was set to 2010+. Since the production technology for 2nd generation biofuels is not fully developed yet, it would have been logical to set the timeframe as long as to consider this technology to be mature. However, this would probably on a longer time horizon than indicated above. If one expects that the first pilot plants for some of the fuel options, such as e.g. cellulosic ethanol or syngas fuels could be erected within 5 years, we would have to add another couple of years before full-scale plants could be in operation and yet another couple of years before this technology would be mature. This would indicate a timeframe of about 2020. As it would be somewhat difficult to project the evolution of engine and aftertreatment technology for such a long period of time, a somewhat shorter timeframe was selected.

3.3.2 Vehicles

Base vehicles

Two vehicle types were selected in this study. A city bus (or “transit bus” in US English) was selected to represent a typical vehicle category operating in densely populated city centres. A long distance lorry with a maximum weight of 40 tonnes was selected to represent a heavy-duty vehicle operating in long-distance freight.

Data from Volvo Bus and Volvo Truck, as well as several other heavy-duty vehicle manufacturers were used as input data in both cases. Therefore, it cannot be stated that the vehicles are typical Volvo vehicles but rather that they are representative of European vehicles of each category.

The most important vehicle data are shown in The city bus

In **Fel! Ogiltig självreferens i bokmärke.**, some relevant vehicle data for the city bus are shown.

Table 3 (bus) and **Table 4** (lorry).

The city bus

In **Fel! Ogiltig självreferens i bokmärke.**, some relevant vehicle data for the city bus are shown.

Table 3. *Some vehicle data, city bus*

Parameter	Unit	Value	Comments
Gross weight	kg	18 000	Not fully utilised in most cases
Operating weight	kg	12 865	For the diesel-fuelled baseline bus
Average number of passengers	-	15	Average in Sweden range from 12 to 20 passengers.
Frontal area (A)	m^2	7,5	
Drag coeff. (C_d)		0,70	
Air resistance (Cd×A)	m^2	5,25	
Rolling resistance	-	0,0056	
No. of gears	-	5	Automatic transmission (converter)
Final drive ratio	-	4,33	Nominal for diesel-fuelled baseline

The impact of vehicle weight on the number of passengers that a city bus can take should be discussed. In the past, there have been examples where, for example, gaseous-fuelled vehicles have been much heavier than their diesel-fuelled counterparts. Therefore, the maximum allowable number of passengers has been reduced for these vehicles in the past. However, due to shift in weight distribution between front and rear wheels, it has been possible to avoid this drawback on modern gaseous-fuelled vehicles. Thus, the same maximum number of passenger has been foreseen for all fuels investigated.

Frontal area and drag coefficient has a relatively small impact on fuel consumption for city buses. The reason is that the vehicle speed is relatively low in this application. The rolling resistance is relatively low and is commented further in the section below about the long-distance lorry.

Automatic transmissions using hydraulic torque converters have been the most commonly used type of transmissions in city buses. A 5-speed gearbox from the database in Advisor was used for the city bus. The rationale for using a gearbox without torque converter might be discussed. Modern gearboxes using automated gearshift has been vastly improved regarding comfort and smoothness of gearshift and reduce the fuel consumption considerably compared to their counterparts using torque converters. However, since most city buses are using transmissions with torque converters, this option was chosen.

The long-distance lorry

In **Table 4**, some relevant vehicle data for the long-distance lorry are shown.

Table 4. *Some vehicle data, long-distance lorry*

Parameter	*Unit*	*Value*	*Comments*
Gross weight	kg	40 000	Max. in most EU countries
Payload capacity	kg	10 900	Nominal payload for diesel baseline
Frontal area (A)	m^2	9,7	
Drag coeff. (C_d)	-	0,56	
Air resistance (Cd×A)	m^2	5,43	
Rolling resistance	-	0,0051	
No. of gears	#	12	Automated manual transmission
Final drive ratio	-	2,85	Nominal for diesel baseline

The gross weight of the long-distance lorry was selected to 40 metric tonnes. In Sweden, a maximum gross vehicle weight of 60 tonnes is allowed. However, it was decided to use 40 tonnes, since this size is more representative for a European long-distance lorry. Therefore, it was also decided not to use the largest class of engines, e.g. up to 16 litres in cylinder capacity, but rather a somewhat smaller 13-litre 6-cylinder engine. The payload capacity varies for long-distance lorries depending on what kind of transport the vehicle has been designed for. For example, extreme designs for transporting volume goods exist. The chosen payload capacity could be seen as a compromise between various options.

Data for frontal are, aerodynamic drag coefficient and rolling resistance was provided by Volvo engineers. It could be noted that the data for drag coefficient and rolling resistance are relatively low, indicating that the vehicles are state-of-the-art for this class of vehicles. Radial tyres of the “super single” type have significantly lower rolling resistance than older types of tyres. This is illustrated by comparing to data for various model vehicles in Advisor, where rolling resistance in the range of 0,008 to 0,009 is common for many of the vehicles in the database.

A 12 speed automated mechanical transmission was anticipated for the lorry. Data for gear ratios from several commercial gearboxes for heavy-duty vehicles were collected and compared. The spread for a 12 speed transmission is often in the interval between 11:1 and 12:1. In this case, a 12:1 spread ratio was chosen with even distribution of the individual stages of 1,253:1. It should be noted that the choice of gears in practice implies that the ratio cannot be absolutely constant for all gears. For example, one of the investigated transmissions had a range from 1,23:1 for the lowest to 1,28:1 for the highest of the gears.

Vehicle weight

Vehicle weight and payload is influenced by fuel choice. For example, liquid fuels with less energy content than diesel fuel, such as, e.g. ethanol, have to carry extra weight or the incremental tank volume to obtain the same range as the diesel fuelled baseline vehicle. In practice, this reduces the maximum payload of the vehicle on the condition that the same range is foreseen in both cases. For gaseous fuels, where the tank weight can be considerably higher than a normal diesel tank, this impact can be of great importance. Thus, the option of using gaseous fuels in long-distance lorries is not practical. Similarly, the use of cryogenic fuels such as LNG is not very practical in city vehicles such as transit buses. In this case, gaseous fuels are more practical to handle. The achievable range with gaseous fuels is also acceptable in this case; whereas this would be a serious issue for long-distance lorries.

It was anticipated that similar amount of stored energy in the tank would be used for all vehicles in each vehicle category. With similar efficiency, this would give the same range. No compensation was made for those drivelines that have a lower efficiency than the baseline vehicle. Thus, the maximum difference in range might be up to 20 % between the lowest and highest range. This is probably acceptable, since iterations to achieve similar range for all fuel/vehicle combinations would result in much additional work. The diesel-fuelled baseline bus had a tank capacity of 300 litres, corresponding to approximately 300 kg of total tank weight. It was assumed that the tank would be filled to 50 % of its capacity in the vehicle simulations. The corresponding tank capacity for the diesel-fuelled lorry was 600 litres.

Data from Volvo for tank weight for gaseous fuels such as natural gas and biogas (CNG & CBG) as well as DME respectively were used to calculate the vehicle weight for those vehicles. Data were recalculated so that similar stored energy would be used in all cases.

Storage of compressed methane is not very practical for long-distance lorries. The only realistic option in this case is to use liquefied cryogenic methane. Relatively little data is available for LNG vehicles in general, since such vehicles are not commercially available in large numbers in Europe. The market leader in developing LNG engines is probably Westport Innovations Inc.; including the joint venture Cummins Westport with the US engine manufacturer Cummins. In a brochure with product information from Westport, the incremental weight over diesel-fuelled vehicle has been estimated for the corresponding LNG vehicle [31]. The largest tanks (two separate tanks) with a gross volume of 908 litres, corresponds to a diesel fuel volume of 424 litre diesel fuel. This is considered somewhat marginal for a long-distance lorry in Europe. Extrapolation of those data shows that the incremental weight for a lorry with diesel fuel tanks corresponding to 600 litres would be approximately 1 057 kg. They payload would have to be reduced correspondingly. During driving, the weight of the diesel-fuelled vehicle reduces more than the LNG vehicle while the fuel in the tank is consumed, since the energy content of methane (50 MJ/kg) is higher per kg than for diesel fuel (42,6 MJ/kg). For empty tanks of the volumes mentioned above, this impact would be 74 kg. Assuming that half tank would be more representative for normal driving, this would equal only 37 kg, i.e. this effect can be neglected. Impact of boil-off on fuel consumption when using cryogenic fuels was not considered. This is presumably a good assumption for long-distance lorries that are not parked for long periods of time, but cannot be neglected for other vehicle types. In any case, there is a lack of reliable data regarding boil-off.

Load factor

The average number of passengers for a city bus varies considerably. In Sweden the range is from about 12 to 20. Higher numbers are usually seen in larger cities, while cities with less population usually have lower numbers. In this case, 15 passengers weighing 75 kg each (including luggage) was considered realistic. The maximum passenger capacity could vary if the vehicle gross weight or maximum axle load is exceeded. This has previously been a problem for vehicles using gaseous fuels. The combination of reduced tank weight for modern tank systems in combination with a shift of weight to the front axle has enabled the bus manufacturers to retain the maximum passenger capacity for these buses. Thus, an average of 15 passengers could be used for all buses.

The load factor for long-distance lorries is seldom higher than 70 %. A load factor of 60 % has been anticipated in this study. Often the lorry runs empty in one direction and with full load in the other direction and sometimes no trailer is attached to the vehicle. To simplify the calculation of the load factor it has been anticipated here that a trailer is always attached to the vehicle. Simulations have been made by using no payload in one case and 100 % payload in the second case. Interpolation between these two points has been made to calculate the fuel consumption for 60 % load factor. Linear interpolation can be carried out since previous investigations (e.g by this author [5]) have shown that the fuel consumption varies approximately linearly with payload weight.

3.3.3 Emission targets

The Euro V emission limit, i.e. 2 g/kWh, was set as target for the NO_X emissions, as indicated above. At the time this study was initiated, the discussions about Euro VI and VII had not started yet and therefore, Euro V was set as the most reasonable emission target.

For the diesel engine baseline, selective catalytic reduction (SCR) was foreseen. A urea additive is used to reduce NO_X emissions in this concept. For the alcohol and DME engines, exhaust gas recirculation (EGR) is used to reduce NO_X emissions and the methane-fuelled engines are using a lean-burn (LB) concept. A more thorough overview of engine technology is provided below.

4 RESULTS

The results from the present study have been divided into a couple of sections according to the three steps in a WTW analysis. First, results on fuel production are presented and discussed. A comparison with data from the JRC study is made. By summarising the other steps such as, e.g. feedstock production and fuel distribution, the whole chain from resource to the tank can be obtained (WTT). Second, the fuel converters and the vehicles are described and the results from vehicle simulations are reported (TTW). And finally, the aggregated results from the two previous areas are presented (WTW).

4.1 Fuel production and distribution

4.1.1 Feedstocks

Lignocellulosic matter is the most abundant biomass category available on the planet. Other options that could be used as biofuel feedstocks are starch, sugar and vegetable oils. Animal fat could also be used in a similar way as vegetable oil. All of the latter mentioned feedstocks are significantly smaller resources than lignocellulose. Therefore, it has been obvious to concentrate on lignocellulosic matter as the main source of biomass in this study. The feedstock for biogas production should be mentioned separately. The most popular feedstock for biogas is sewage sludge and waste, since the feedstock cost in these cases is negligible and sometimes even negative. Manure (liquid and dry) is another source, which is popular in rural areas. Regarding the feedstock categories mentioned above, the feedstock used for biogas production can be a combination of all those categories.

Natural gas (NG) is a fossil resource that is complementary to crude oil for motor fuel production. Natural gas comprises mainly of methane and can be used as such in compressed form (CNG) or in liquefied form (LNG). The latter option is mainly of interest for heavy-duty vehicles and not considered for light-duty vehicles, since it must be handled in cryogenic form. Due to “boil off” and several other issues, this option is best suited for heavy-duty vehicles where the operation of the vehicle and refuelling can be handled by specially trained personnel.

4.1.2 Fuel production

The data for fuel production have been derived from the previously mentioned WTW study by CONCAWE, EUCAR and EU/JRC [27]. In the following, this study is referred to as the “JRC” study.

One specific case worth discussing is the ignition improver used for the alcohol fuels in the compression ignition diesel engine. The ignition improver used in the contemporary heavy-duty ethanol vehicles, Beraid 3540, is currently produced from fossil origin but could also be produced from biomass. Attempts have been made to estimate the energy use in production of Beraid from biomass but the assumptions made have been relatively crude [32]. The alternative of anticipating the use of fossil-based Beraid would imply a much lower energy use than Beraid from bio-origin. In fact, the energy use could be even lower than for producing one MJ of ethanol in some of the ethanol cases and this would skew the comparisons with other fuel options. Therefore, it was decided to neglect the impact of the ignition improver and just assume the same energy use for Beraid as for producing one MJ of ethanol. The same strategy was used also for methanol. Ethanol fuels must also contain denaturants to avoid ingestion and some similar solution would most likely be needed also for methanol. The contributions from these and other additives were also neglected.

Diesel fuel

No modifications have been made of the data from the JRC study for diesel fuel production. As discussed above, this diesel fuel quality corresponds to EN 590 ultra-low sulphur diesel fuel (<10 ppm_w S).

Biofuels

In the JRC study, credits have been given for integrated electricity production by comparing with production facilities of similar size and technology. In the present study, a different approach has been made. Here, it has been foreseen that all external electricity production will be made in larger power plants of 200 MW size. Consequently, the efficiency will be similar for all external power generation. Credits for integrated electricity generation at the fuel production plant have been made against this reference. One example can illustrate the impact of this methodology. If electricity is produced in a 10 MW unit at the fuel production plant, this production will be compared to the 200 MW plant and not to a 10 MW stand-alone plant. Thus, this “trade-off” will reduce the overall efficiency in the present study.

In some of the cases in the JRC study, e.g. black liquor, the technology and efficiency for the external electric power plant is not similar to the other cases in the JRC study. Since electricity use is essential for the fuels produced from black liquor, it is particularly important to compensate for these differences.

Since one of the main objectives of using biofuels has been to reduce the emissions of greenhouse gases, it is essential to use as much bio energy in the fuel cycle as possible. Therefore, those cases where the use of bioenergy is the highest have been selected from the JRC study. Alternatives with very high carbon emissions in some of the production stages, e.g. using electricity generation based on lignite as fuel, have not been considered at all. Whenever possible, fossil fuels have been replaced with biofuels in the fuel production process, such as, e.g. electricity generation and process heat. This is not the case for feedstock cultivation and transport, as well as for fuel distribution where the data from JRC have been retained.

Due to the limited scope of the present study, only a limited number of all the cases in the JRC study have been used. The selection was based on the criteria indicated above. In **Figure 3**, a comparison of the results from the present study and the JRC study has been made.

For comparison, it could be mentioned that the efficiency of diesel fuel production in the JRC study is 88,1 %, when the production efficiency is calculated for the “refinery gate” in a similar way as the fuels above.

Biogas, various feedstocks

Compared to the JRC study, replacing electricity production according to the principles indicated above had a very small impact on the efficiency.

Ethanol from wheat

Among the cases in the JRC study, it was chosen to use one where as much bioenergy as possible was used as process energy. For wheat, this implies that, e.g. wheat straw is used as an energy source (case WT4d in JRC). The methodology to correct for electricity produced from biomass, as described above, reduced the efficiency compared to the JRC study.

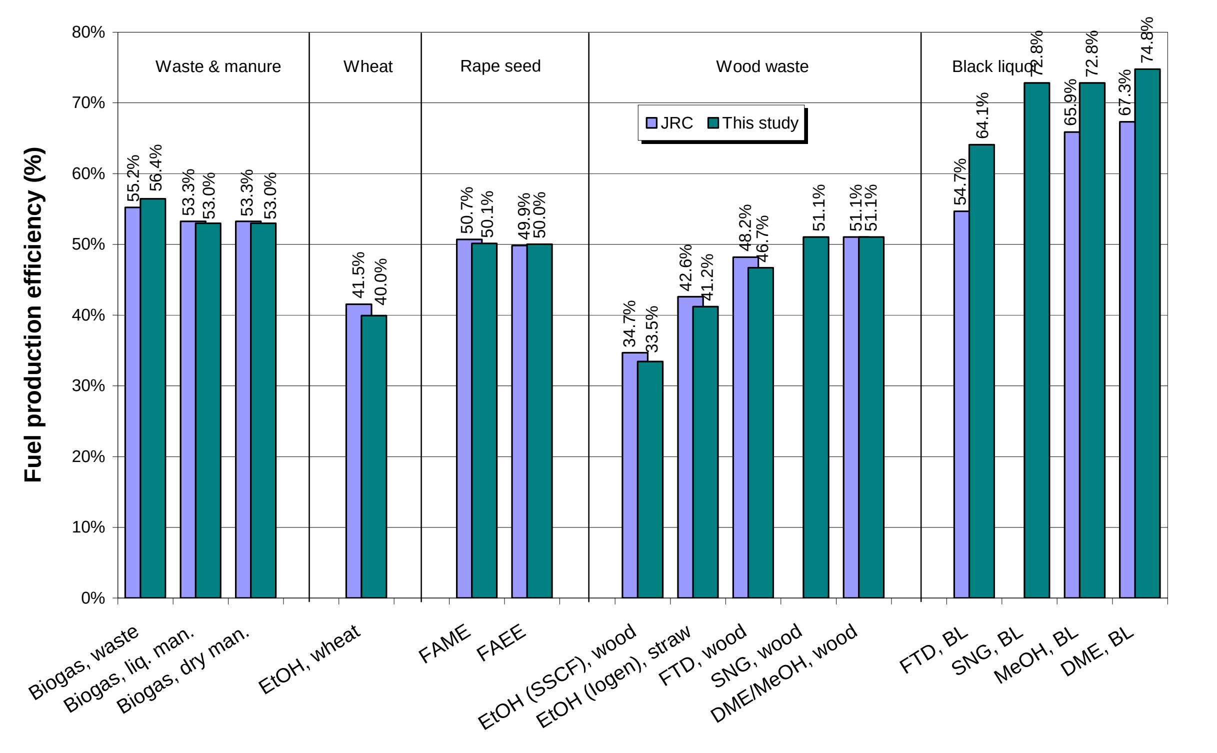

***Figure 3**. Comparison of fuel production from biomass resources*

FAME and FAEE

The impact of electricity production had almost a negligible impact on the efficiency of FAME and FAEE production.

Ethanol from wood

The production of ethanol from lignocellulosic is one of the most interesting options for the future. It was decided to make a more comprehensive assessment of the possible options here and the potential for future development. This work is described in more detail in Appendix 1. One the one hand, some of the options for producing ethanol gives by-products, such as lignin, which cannot be fully utilised in the plant. One the other hand, the most advanced processes cannot be fully self-supplied with process energy from the lignin but has to use biomass to overcome this deficit. The JRC study did not anticipate the most extrapolated technology in this area, so the last problem does not occur. However, one could still argue that there is some further development potential in this area. As for all the other fuel options in the present study, the production plants were considered to be stand-alone or partly integrated plants where low-grade energy in the form of hot water for district heating and similar purposes cannot be utilised. This is in line with the methodology in the JRC study. Thus, the corrections made in the present study considered only electricity production. In the case of ethanol production from wood waste, the correction was not negligible, as in some of the other cases. In comparison to the other fuel options from wood waste, the efficiency is relatively low.

Fischer-Tropsch diesel from wood

The production of Fischer-Tropsch diesel has been adopted from the JRC study besides the re-calculation due to the electricity production as described above.

Use of by-products

In the production of Fischer-Tropsch diesel (FTD), a significant share of the yield will be Fischer-Tropsch naphtha (FTN). It is important to find the best way to utilise this naphtha in order to maximise the total efficiency. Therefore, Nykomb Synergetics was subcontracted to carry out a study about the best use of naphtha. The report from this work is attached in Appendix 2.

According to the study by Nykomb, the naphtha share could range from 20 to 34 %. The following options to utilize the naphtha were investigated:

- Isomerisation of naphtha to produce a petrol component for blending
- Use of naphtha in a combined cycle (CC) power plant in comparison to an integrated gasification combined cycle (IGCC) fed by biomass

The study by Nykomb showed that FTN could be used for upgrading to a petrol component. However, there is a loss in efficiency due to the upgrading and furthermore, the efficiency in an otto engine is lower than in a diesel engine. The use of FTN with the highest possible efficiency would be to use it in a combined cycle power plant. Depending on size, the net electric efficiency would range from 54 to 60 %. In this study, an efficiency of 58 % was anticipated. In comparison, the net electric efficiency from biomass to electricity for an IGCC process would range from 43 to 49 %. This is in line with the 48,2 % used in the JRC study.

Synthetic natural gas (SNG) from wood and black liquor

Synthetic natural gas (SNG) or, in another definition; supplemental natural gas, can be produced from syngas via methanation. SNG is relatively similar in composition to biogas and can be utilised in vehicles in a similar way as natural gas or biogas. SNG was not considered in the JRC study.

The literature survey did not provide any good insight about the efficiency for this route. Published efficiencies range roughly from 55 to about 75 %. Even higher numbers have been mentioned but then (presumably) include full utilisation of waste heat for e.g. district heating, which has not been the case for other fuels in this study. Due to the lack of reliable data, the same efficiency as for methanol from wood and black liquor respectively was set for SNG. It is recommended to investigate this option further.

Methanol and/or DME from wood

The processes used by JRC for production of methanol (MeOH) and DME from wood are based on relatively old references from early to mid 1990’s, e.g. Katofsky [33]. It is likely that most advanced processes would provide higher efficiencies. Likewise, the comparison between production of the same fuels from black liquor would be biased in favour of the black liquor options due to these circumstances. However, no adjustments to the data for methanol and DME produced from wood were made here.

Synthetic fuels via syngas from black liquor (FTD, MeOH and DME)

The JRC study used the report from the so-called BLGMF II project by Ekbom et al. as basis for the production of synthetic fuels from black liquor [34]. According to the JRC study and communication with one of the authors of that study by e-mail, they have verified the calculations in the BLGMF II report [35].

For FTD from black liquor, the same options for using the naphtha by-product as for FTD from wood apply.

4.2 Engine technology

The background for the selection of various engine technologies is provided in this section. Due to confidentiality reasons, not all the relevant data for the baseline Volvo engine used in the calculations and simulations can be disclosed.

4.2.1 Diesel-fuelled engines

A diesel-fuelled 13-litre heavy-duty engine from the Volvo Truck engine programme was chosen as the baseline engine in this study. Data for this engine was provided by Volvo Powertrain.

Truck engine

In **Figure 4**, the power and torque of the Volvo 13-litre engine is shown. The engine has constant torque from slightly above 1 000 r/min and constant power from 1 400 to 1 800 r/min. From 1 800 to 2 100 r/min, the power is declining. At 2 100 r/min the speed limiter cut-off is very steep.

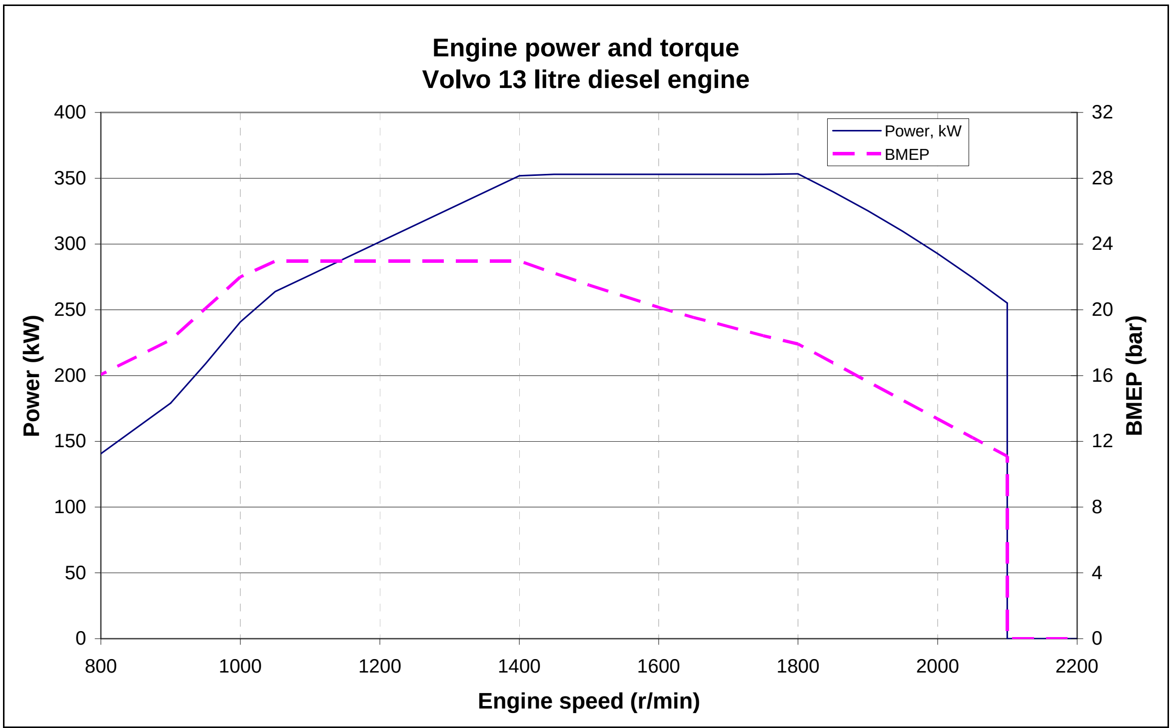

Figure 4. *Volvo 13-litre diesel engine*

Due to the structure of the computational algorithms in Advisor, it was somewhat difficult to obtain a speed limiter as steep as for the Volvo 13-litre engine. The maps (matrices) for fuel consumption and NO_X emissions are indexed with the same engine speed and torque as for the maximum torque curve. To avoid potential computational problems in having the same engine speed in two of the columns in these matrixes, a less steep speed limiter was used instead, i.e. a straight line from full torque at 2 000 to zero at 2 100 r/min. The same methodology was used also for all the other engines derivatives, e.g. scaled engines, investigated here.

Due to that some of the data on the Volvo engine was considered proprietary information by this company, not all relevant information for the 13-litre engine is shown here.

Bus engine

Scaling of the 13-litre engine above has been used to "design" a bus engine. Generally, the brake mean effective pressure (BMEP) level for city bus engines is lower than for heavy-duty engines used in long-distance lorries. Therefore, the BMEP level of the bus engine was reduced from 23 to 19 bar. A scaling factor of 0,8 (for the length scale) gave a 6-cylinder engine having a cylinder volume of 6,7 litres. The bore/stroke ratio and average piston speed was retained, implying that the engine speed (r/min) was higher for this engine than the 13-litre engine. The scaling factor corresponds to bore and stroke values in absolute numbers of 104 and 132 mm respectively.

The rationale for using a 6-cylinder engine might be discussed. For this engine size, both 4 and 5-cylinder engines might be considered instead of a 6-cylinder engine. The advantage of using fewer numbers of cylinders would be to reduce fuel consumption (lower for larger cylinder) and cost but there are also drawbacks for these options, such as, e.g. vibration, noise and harshness. There are also other pros and cons for each option but these are not further discussed here. Finally, it could be mentioned that, in Europe, most city buses have 6-cylinder engines, while the most popular bus engine (9-litre engine by Detroit Diesel Co., DDC) in the USA has only 4 cylinders.

An engine with smaller cylinder capacity per cylinder has usually somewhat higher fuel consumption than a corresponding engine with larger cylinders. To take this fact into account, the map of specific fuel consumption of the 13-litre engine has been increased by 3 %. On top of that, the reduction of the BMEP level (from 23 to 19 bar) also decreases the efficiency of the engine at low engine loads.

4.2.2 Alcohol-fuelled engines

Since the design of ethanol and methanol engines would be very similar regarding technology and design, they are discussed in the same section.

Alcohols are ideal fuels for otto cycle engines but these engines have lower efficiency than diesel cycle engines so the latter option is preferred for heavy-duty vehicles. Recent very interesting work has been conducted in this field by, e.g. MIT, on ethanol-fuelled direct injected otto-cycle engines [36]. The main focus in this case was the application of the technology in light-duty vehicles. However, a report from MIT supported by Volvo Powertrain and to be published in October 2008 has focussed on the use of ethanol and methanol fuels in a heavy-duty engine, which might indicate that there is a growing interest in this field as well [37]. In conclusion, engines that could fully utilise this potential are still far from the market. Therefore, the otto-cycle engine option has not been considered here.

Ethanol engine

As mentioned above, only the diesel cycle was considered for the alcohol engines. In the past, methanol-fuelled engines (M100) by Detroit Diesel Co. (DDC) have been used in the USA. These engines were of the 2-stroke type and not very representative of modern engines. The most interesting feature of these engines was that they used glow plugs for ignition. As mentioned already in the "Background" chapter, Scania has been the main proponent of ethanol engines in Europe. The first larger fleet of ethanol buses (30) was introduced in Stockholm in the early 1990's. Since then, the Scania ethanol engines have been modified and further developed in several generations. The Scania engines have used an ignition improver blended in the fuel for ignition.

In this study, two engine options were chosen. The first is similar to the Scania engine using increased compression ratio to minimize the use of ignition improver. The second approach has been to use a glow plug to ignite the fuel. The implications on engine design by using either of these options is discussed in some more detail below.

Ideally, the compression and expansion in an engine would be adiabatic. In practice, a polytrophic compression with an exponent lower that the ideal 1,4 for pure air is a better assumption for such processes in a real engine. If the polytrophic exponent for the compression stroke is known, the increase in cylinder pressure at top dead centre before combustion is initiated can be easily calculated according to **Equation 1**:

$$F = (\varepsilon_1 / \varepsilon_2)^p$$

Where: F is the factor for cylinder pressure;
ε the compression ratio and
p, the polytrophic exponent (or polytrophic index)

Equation 1. *Relative increase in maximum cylinder pressure*

Using the data mentioned above and a polytrophic exponent of 1,35, the factor would be approximately 1,5 and a similar factor for maximum pressure for the peak combustion pressure in the cylinder can also be assumed provided that the charge pressure would be the same. However, in practice, ethanol can tolerate somewhat lower air excess ratio than diesel fuel due to the almost "smoke-free" combustion. Assuming that the air/fuel ratio could be decreased by 10 %, maybe even 20 %, due to reduced manifold pressure, the increase in cylinder pressure would still be considerable. A decrease in charge pressure by 15 %, and corresponding decrease in cylinder pressure, would still increase the cylinder pressure by 25 % over the baseline. Thus, the engine power and torque would have to be reduced to maintain the cylinder pressure permitted for the particular engine design. If we assume that, the maximum cylinder pressure would be 140 bar in the baseline case, i.e. relatively moderate level, the pressure would be 175 bar in the ethanol case. Note that the assumed baseline in this case was for an engine of relatively low power density. If we presume that the baseline cylinder pressure would be 175, as for a modern heavy-duty diesel engine, the ethanol engine would have a cylinder pressure of 219 bar. Although about 200 bar might be considered state-of-the-art for new engine designs, it is unrealistic to assume that even higher cylinder pressure could be achievable in the near future. Consequently, the engine power and torque would have to be reduced correspondingly to maintain cylinder pressures within reasonable limits. In the case above, that power reduction would have to be some 20 % implying that a larger baseline engine would have to be used if the same power as the diesel baseline is the objective. For example, an 11-litre engine would have to be used instead of a 9-litre engine. As long as much larger diesel engines than in city buses are used for long distance lorries, there will always be larger engine platforms available for city bus applications that could be adapted for ethanol.

Another example to illustrate the loss in power density could be if a 9-litre engine of about 170 kW (230 hp) is used in a bus for ethanol as well as for diesel versions. In this case, the incremental cost of providing an ethanol version of the engine would be marginal. However, if the base diesel engine would be downsized to, say about 7 litres, or maybe 6 litres, an ethanol-fuelled version could no longer be made with the desired power and torque. The incremental cost of installing a larger engine, possibly also a different transmission and many other corresponding powertrain components, would be high. Note that a 7-litre engine would be smaller than some of the city bus engines on the market today, but nevertheless, the power density (e.g. 170 kW) is definitely possible to achieve today on diesel-fuelled engines without any exotic technology. The example above of scaling the truck engine to a bus engine is probably convincing enough to accept this hypothesis. There are also examples of commercially available diesel-fuelled engines of about 6,5 litres in size with as high, or higher, power density as mentioned above.

The balance between the demand to reduce the need for ignition improver and to increase the power density is delicate. A compromise is apparently inevitable. Scania has recently developed a new ethanol engine. They have also transferred the platform from the "old" 6-cylinder 9-litre engine to a new 5-cylinder 9-litre engine. Presumably, the durability to allow high cylinder pressure is better for the new engine. Recent information from Scania reveal that they have increased the compression ratio from 24:1 to 28:1 on their new ethanol engine [38, 39]. The objective seems to be to reduce the use of the ignition improver. Although the limits for cylinder pressures most likely could be met, as a simple calculation shows, the result would be a significant reduction of the power density compared to the diesel-fuelled baseline, i.e. some 35 % using the same calculation scheme as above. The current (2008) new Scania engine platforms comprise cylinder sizes only slightly different in bore but with a larger difference of the stroke. The number of cylinders is 5, 6 and 8. The highest power density on any Scania diesel engine today was in autumn 2007 the 12-litre Euro IV engine at 353 kW (480 hp). That would correspond to 277 kW (377 hp) on a 9-litre engine, if the power density could be maintained. A 35 % reduction in power density would give 179 kW (311 hp), i.e. well above the requirements for a city bus. Nevertheless, the loss in power density is significant and this concept would be a considerable drawback for heavier truck applications. Using the building blocks of the modular engines, the larger stroke and a 6-cylinder version could easily be applied to increase the power by 27 % due to a larger engine capacity compared to a bus engine. However, this would considerably increase the cost and impose some packaging problems in the bus chassis.

A similar calculation as above was used to "design" an ethanol engine for a city bus has also been made for an ethanol engine for long-distance lorries. In this case, the engine size had to be increased to about 17 litres to achieve similar torque as the diesel-fuelled counterpart. In addition, a compensation has to be made for that the larger engine cannot run at higher piston speed that the 13-litre engine. This implies that the engine size has to be increased to over 18 litres to provide similar power as the 13-litre engine. Since the final drive ratio is lowered correspondingly, the increase in engine torque (by increasing size from 17 to 18 litres) is not reflected in any increase in torque on the wheels; i.e. the vehicle driveability is comparable.

Methanol engine

It was anticipated here that methanol engines could be very similar to the ethanol engines discussed above. The only main difference would be that the fuel flow in the injection system would have to be increased in the methanol case. Therefore, no particular optimisation was foreseen for methanol engines in comparison to ethanol engines. The same "base" ethanol engine data was used in the simulations. The only difference was that the engine had to be scaled up somewhat to compensate for the higher fuel load necessary to achieve similar vehicle performance.

Broader use of alcohol engines that use ignition improver

The examples above have showed the importance of achieving similar power density on alcohol-fuelled engines as on their diesel-fuelled counterparts; a demand not satisfied by using the ignition improver concept. This is of particular importance if ethanol and/or methanol are to be used on a broader scale than just for a limited vehicle category. A lower power density necessitates a larger engine capacity and a higher cost is the result. Therefore, it is of importance that *a combustion concept is developed for ethanol engines that have the ability to raise the power* *density to the same level as for their diesel-fuelled counterparts.* The ignition improver concept does not fulfil this criterion and it is difficult to see that the current approach could be further developed to achieve this objective unless a radical new injection improver that would be much more effective than the current one could be developed.

4.2.3 Methane-fuelled engines

Properties of methane as a motor fuel

Natural gas, biogas and pure methane could be treated as relatively similar fuels from combustion viewpoint, although it should be noted that the composition is sometimes quite different. Natural gas often contains "heavier" hydrocarbons than methane, which biogas and "pure" methane, as in liquefied natural gas (LNG) does not. Thus, the heating value of natural gas can be some 10 % higher or more compared to biogas. These issues have been taken into considerations in the EU emission limits by using certification fuels with different heating values. Blending methane with hydrogen is yet another option that has been discussed but in this case, the combustion properties are altered significantly [40]. As relatively few studies are still available in this area, it is too early to assess the potential improvement by blending methane with hydrogen.

Methane is not a "natural" diesel fuel due to its high octane and low cetane number. Based on these properties, it is conceivable that methane would be ideal for the use in spark ignition engines. Thus, conversion of an engine from diesel cycle (compression ignition) to otto cycle (spark ignition) is an obvious option. However, there are also possibilities to use the diesel cycle for methane-fuelled engines. In this section, a brief overview of the various options is provided.

Engine technology for methane utilisation in heavy-duty vehicles

The various engine technology options for using methane in internal combustion engines can be grouped into the following three main groups:

- Spark ignited natural gas, otto cycle
- Indirect injection of natural gas, diesel cycle
- Direct injection of natural gas, diesel cycle

There are two main variations of the spark ignited natural gas engine (SING). The first use stoichiometric combustion (λ=1) and three-way catalyst (TWC), while the second operates on air excess (lean-burn, or LB). Mixed concepts, using stoichiometric combustion at a certain load and speed range and lean-burn at other areas have been used. Problems with thermal stresses and low power density have so far favoured the use of the lean-burn combustion system over TWC in heavy-duty engines. However, the greater emission potential for TWC in comparison to LB is a parameter of increasing importance that might shift the focus in the near future.

Gaseous fuels can be used in a diesel cycle. In this case, the high compression ratio of the base diesel engine is retained. Thus, the energy efficiency of the diesel engine can also be retained. Since methane has a very low cetane number, it cannot be used without ignition aid in a compression ignition engine. The most logical solution would be to use a spark plug. However, this is not easily accomplished for two reasons. First, methane requires very high ignition energy. Second, the high compression ration and turbocharging (which is normally retained from the base diesel engine) creates a very high cylinder pressure before ignition. At high cylinder pressure, air acts as an insulator, which would require an even higher voltage or smaller distance between the electrodes leading to a very short spark plug life. These effects combined, makes spark ignition unpractical in diesel cycle engines.

The simplest way to enable the use of a diesel cycle with gaseous fuels is to external mixture preparation (i.e. indirect injection or a gas mixer) and to ignite the gas with direct injection of diesel fuel. A dual-fuel concept can be made so that diesel fuel either can be used for igniting the gaseous fuel enabling up to 90 % substitution of diesel fuel or diesel fuel is used as the only fuel. The denotation dual fuel natural gas (DFNG) is often used for such concepts. A specialised version of the external mixture preparation of gas with diesel ignition is the so-called pilot injection of diesel fuel to ignite the natural gas introduced via external mixture preparation (Pilot Ignited Natural Gas, PING). In this case, a dedicated diesel injection system is used, where the system has been optimised for pilot injection only. Thus, this engine is not capable of dual fuel operation. Instead, the pilot injection can be better optimised.

When direct injection of gas (DING) is used, new opportunities arise. First, direct injection of gas enables a better control of the combustion event than using external mixture preparation. Second, if a combined injector for both gas and diesel fuel for pilot ignition (DING (PI)) can be used; the original cylinder head of the diesel engine may be retained. This facilitates an easier engine conversion. A third advantage is that direct injection improves control of engine knock. The development company Westport has been developing a combustion system for CNG and LNG that use a combined injector for gas and diesel fuel [41]. A further advantage of direct injection of gas is that a glow plug could be used for igniting the gaseous fuel (DING (GP)). It should be noticed that glow plug ignition can only be used with direct injection of gas. If and external mixture preparation is used, the ignition event would be uncontrolled.

In **Figure 5**, some of the concepts discussed above for using gaseous fuels in heavy-duty engines are depicted in a schematic way.

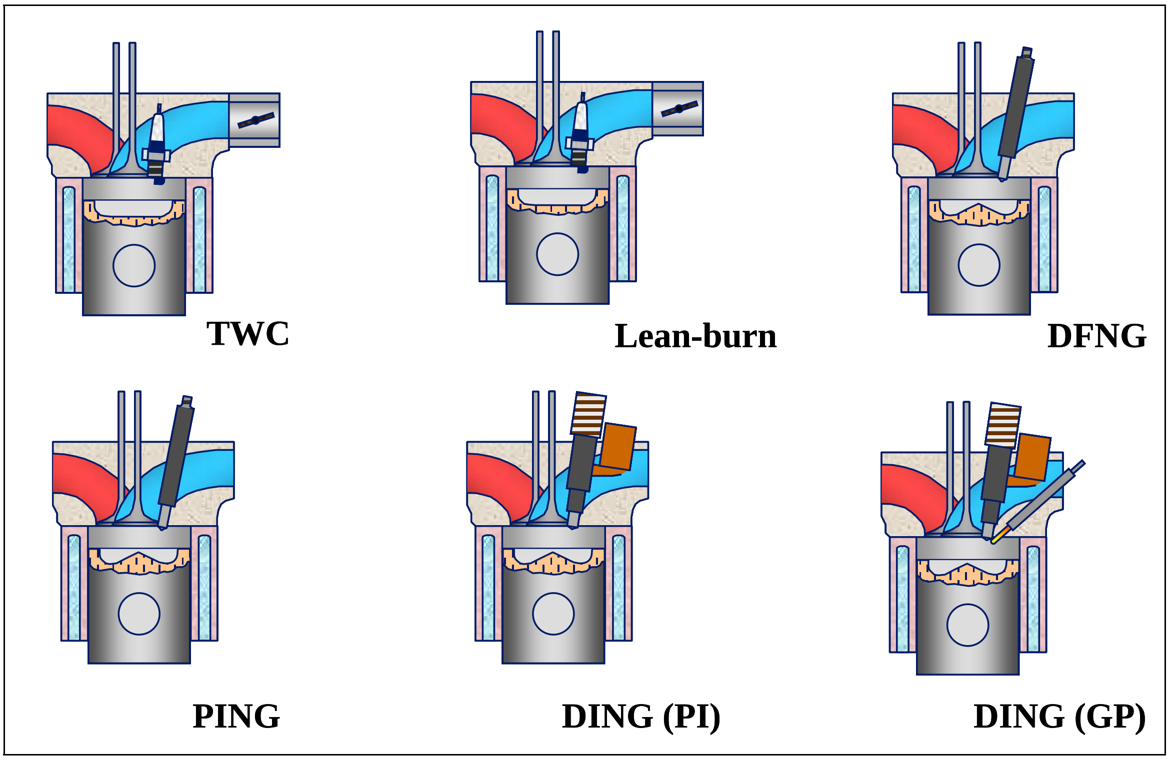

Figure 5. *Various concepts for using gaseous fuels in heavy-duty engines*

An overview of the technology used in natural gas engines has been provided by Nylund and Lawson [42] in 2000 and by Nylund et al. in 2002 [43]. The latter report is mostly focused on light-duty vehicles and advanced drivetrains, such as fuel cells. An overview of the worldwide use of natural gas fuelled buses is provided in another report by IANGV [44].

Utilisation of methane could be either in compressed form (CNG and CBG) or in liquefied form (often referred to as LNG[4] if natural gas is the fuel). The difference regarding engine performance and emissions between these options is relatively small. Injection of liquid cryogenic fuel is theoretically possible in the latter case and could bring some benefits due to the charge cooling effect of LNG evaporation.

The use of cooled EGR in combination with turbocharging provides a potential to increase the engine output in stoichiometric gas engines and, at the same time, it decreases the NO_X emissions. Very encouraging results in this field have been achieved at ETH in Switzerland [45]. It could be mentioned that most light-duty SING engines are stoichiometric as their petrol-fuelled counterparts. Turbocharging is used in some cases to restore engine power to the same level as the petrol engine baseline. In case both engines (methane and petrol) are turbocharged, the higher octane number of methane provides the conditions for similar power output.

A specific advantage of SING engines (lean-burn or TWC) is the inherently low particulate emissions in comparison to diesel engines *without* particulate filters. An additional advantage concerns the NO_X emissions, which are generally lower (lean-burn) or significantly lower (TWC) than from conventional diesel engines.

Emission potential of TWC engines

The TWC system is superior to the lean-burn concept in its emission performance for most emission components. Potentially, an emission level far below Euro V can be achieved by using the TWC concept. US 2010 can also be met, as reported by US engine manufacturers and associated suppliers. Important issues are engine calibration and the mixture control. This is particularly important in transient operation as in the ETC cycle. Engines with high engine-out emissions (as the TWC concept) rely solely on oxidation and reduction of the emission components in the catalyst. Therefore, the catalyst durability is a crucial issue for the TWC concept.

Future development of lean-burn engines

Using stoichiometric (λ=1) or slightly lean (i.e. λ in the order of 1,2 to 1,3) combustion results in high (stoichiometric) or very high (slightly lean) NO_X emissions. NO_X levels as high as 25 g/kWh have been reported. However, by increasing the air excess further, the NO_X emissions decrease again. This is the rationale behind using the lean-burn combustion system. Continuous work is going on to improve the lean-burn combustion system. This is exemplified by a study by the consultant company FEV in Germany [46]. Increasing the specific power and torque of a lean-burn engine to the same level as for its diesel counterpart is an important issue. Extending the lean limit of the combustion system is essential to reduce NO_X and THC emissions. Steady-state tests showed that, in a wide area of the load and speed map, the NO_X level was between 1 and 2 g/kWh. This implies that the NO_X limit in the Euro V and EEV regulations could be met, although no transient tests were carried out by FEV that could prove this hypothesis.

The use of EGR does not provide much real emission benefit for lean-burn engines, contrary to the cases for diesel engines and gaseous-fuelled engines with TWC. Increasing EGR on lean-

[4] LNG: liquefied natural gas.

burn engines necessitates a reduction of the excess air in order to avoid misfire, thus limiting the potential NO_X reduction by combining these measures. The author of this report has tried to re-evaluate data from a previously published paper [47] to assess this impact but found only a negligible benefit of using EGR. Hence, the practical limits for NO_X emissions from contemporary lean-burn technology might be in the order of 1 g/kWh. Some benefit of EGR might be expected during transient operation when the engine has to revert to low excess air or stoichiometric operation, provided that the intricate transient control of the EGR flow could be managed. If a lower emission level than 1 g/kWh is desired, some kind of NO_X aftertreatment will be required.

A practical limitation for the engine power from spark ignited engines is that a high boost level (and high compression ratio) increases the cylinder pressure, thus limiting the maximum gap for the spark plug, unless very high voltage is used [48]. A higher cylinder pressure increases the resistance of the air in the cylinder and necessitates some countermeasures to improve the spark. Decreasing the gap and/or increasing the voltage reduces the life of the spark plug, a considerable maintenance issue in comparison to diesel engines.

4.2.4 Direct injection diesel cycle (DING) and dual fuel (DFNG)

DING and DFNG engines use late-cycle direct injection and therefore, they are characterised as engines since they are utilising the diesel cycle from a thermodynamic point of view. Both engine types need some kind of ignition source, since methane has a very low cetane number. Furthermore, engine knock has to be avoided in one way or another if some kind of premixing of air and methane is used.

The DING engine uses a small quantity of diesel fuel (“pilot” injection, DING (PI)) or a glow plug (DING (GP)) as ignition sources. Since the injection system for the diesel fuel does not have the capability of greater injection quantities, this option has no dual-fuel properties. On the other hand, an optimisation of the pilot injection can be made to achieve lower emissions. An injection system combining the injection of *both* diesel fuel and natural gas has been developed by Westport Inc. [49, 50, 51]. Another example is the so-called “micro-pilot” injection, which has been developed by FEV [48]. Using a glow plug for ignition of the fuel is another possibility. This idea has been pursued by the development company GVH in Germany, now part of Westport Inc. [52].

A DFNG engine mixes natural gas before induction to the cylinder and use diesel fuel as ignition source. Since the conventional diesel injection is used in this case, this engine has dual-fuel properties. A kind of “hybrid” version of both the mentioned systems – often referred to as PING – uses pilot injection of diesel fuel and premixing of natural gas. It is anticipated here that the diesel fuel injection for PING engines is optimised for low fuel flows to improve emissions with CNG. Therefore, these engines cannot be run in diesel fuel mode as the DFNG can. Since the combustion in DING engines and, to a certain extent, also in DFNG (and PING) engines is not completely premixed combustion, soot formation can occur. Thus, future very stringent particulate emissions may necessitate the use of a particulate filter.

The Dutch institute TNO has showed in a study reported in 2000 that natural gas fuelled engines has a favourable potential to meet the European EEV (Environmentally friendly vehicles) regulation [53]. A technical and economical assessment was made on various technology options and fuels. Given time, all the engine concepts studied by the authors were found able to comply with the EEV limits. Without taxation of the fuels, the variable cost was largely found to determine the concept that was the most economically viable. Some of the *direct injection* gas alternatives – and to a lesser extent – lean-burn technology had the lowest total cost (sum of fixed and variable cost). This cost was lower than for the corresponding diesel options investigated. However, it was also noted that the introduction of direct injection for the gas engines is hampered by the research and development efforts necessary to introduce the technology.

Emission potential of DING and DFNG

There is relatively little information available about the "true" emission potential of DING and DFNG engines at the moment. In the USA, vehicles have been produced for a couple of years but most of these vehicles are, presumably, relative simple conversions. Engines have been certified for the EPA 2004 NO_X limit (enforced from October 2002) of 2,5 g/bhp-hr (3,35 g/kWh). In Europe and Sweden, a number of dual-fuel engines are operating in captive test fleets.

A recent development worth mentioning is that CumminsWestport has certified a 15-litre engine bases on the Cummins ISX engine, using direct injection of liquefied methane and pilot quantities of diesel fuel for ignition. This is the first engine fully optimised for the use of direct injection of LNG and therefore, its emission potential is a signpost of what this technology could achieve in this respect. Data on previous prototype engines and not fully optimised engines with direct injection of gas or liquefied gas have shown that meeting the PM limit of 0,01 g/bhp-hr enforced in the 2007 US EPA emission limits seems very difficult without a particulate filter. The example of the CumminsWestport engine proves this hypothesis since it has to rely on a DPF to meet the limit. Unlike highly oxygenated fuels like alcohols and DME, methane does not burn without soot formation in diffusion combustion. Note that otto-cycle engines usually rely on premixed air/fuel preparation and under those conditions, soot formation is not an issue. Some of the soot formation in the CumminsWestport 15-litre engine most likely originates from the combustion of the diesel fuel used for ignition.

Regarding NO_X emissions, the CumminsWestport engine has been certified below the 2007 US EPA emission limit [31]. The certified level is 0,8 g/bhp-hr (1,07 g/kWh). The calculated interim level for the US EPA 2007-2010 timeframe is about 1,2 g/bhp-hr (1,6 g/kWh), so this engine meets this limit with reasonable engineering margin. Consequently, it is likely that this engine would also meet the Euro V emission limit of 2 g/kWh, although the test cycles are quite different. It is likely that a direct injected engine using methane would need additional measures to meet the US EPA NO_X limit of 0,2 g/bhp-hr (0,27 g/kWh) in 2010 and the proposed Euro VI limit of 0,5 g/kWh in 2014.

Selected methane-fuelled engines

A stoichiometric gas engine using TWC for emission control would probably be the engine technology best representative of a future gaseous-fuelled engine with very low emission level. However, no representative data on such an engine were available when this work was conducted.

Instead for using a stoichiometric gas engine, the lean-burn type of engines had to be chosen. The best data for such an engine was collected from a publication by the German consultant company FEV [46]. Although not a production engine, the published data on this engine also indicate the potential for a relatively low emission level. This engine was able to receive NO_X emissions at about 1 g/kWh for a relatively large share of the steady-state load and speed range. Low steady-state NO_X emissions are not necessarily a guarantee for that a low emission level of NO_X in a transient test cycle could be achieved. However, since the steady-state NO_X level is so low, it is likely that this engine could meet the Euro V/EEV emission limit at 2 g/kWh in the *transient* ETC cycle. Some data of the FEV engine are listed in **Table 5**.

Table 5. Engine data of the FEV engine

Property	Unit	Value
Bore	mm	97,5
Stroke	mm	133
Displacement, per cyl.	cm^3	993
Displacement, total	cm^3	5958
Swirl ratio	dim. less	3,0
Number of valves	-	2
Compression ratio	dim. less	13:1
Rated speed	r/min	2400

A brake mean effective pressure (BMEP) of the FEV engine at an impressive level of 30 bar was achieved. In a later paper by the authors from FEV, the problem of spark ignition at very high cylinder pressures was recognized [48]. Therefore, we have reduced the power and torque curves to obtain a practical limitation for such engines. The same BMEP level as for the 13 litre Volvo diesel engine was chosen. At 23 bar, this is still far above he level achieved on current gaseous-fuelled engines in production, indicating that this could be an overestimate of the potential of the gaseous-fuelled engine. Further changes are a small reduction of the rated speed (corresponding to reducing the rated speed from 2400 to 2300 r/min) and a slight reshape of the torque curve in comparison to the torque of the Volvo diesel engine to better reflect the torque curve of a gaseous-fuelled engine. The fuel consumption map of the FEV engine with the BMEP limitation discussed above is shown in **Figure 6**.

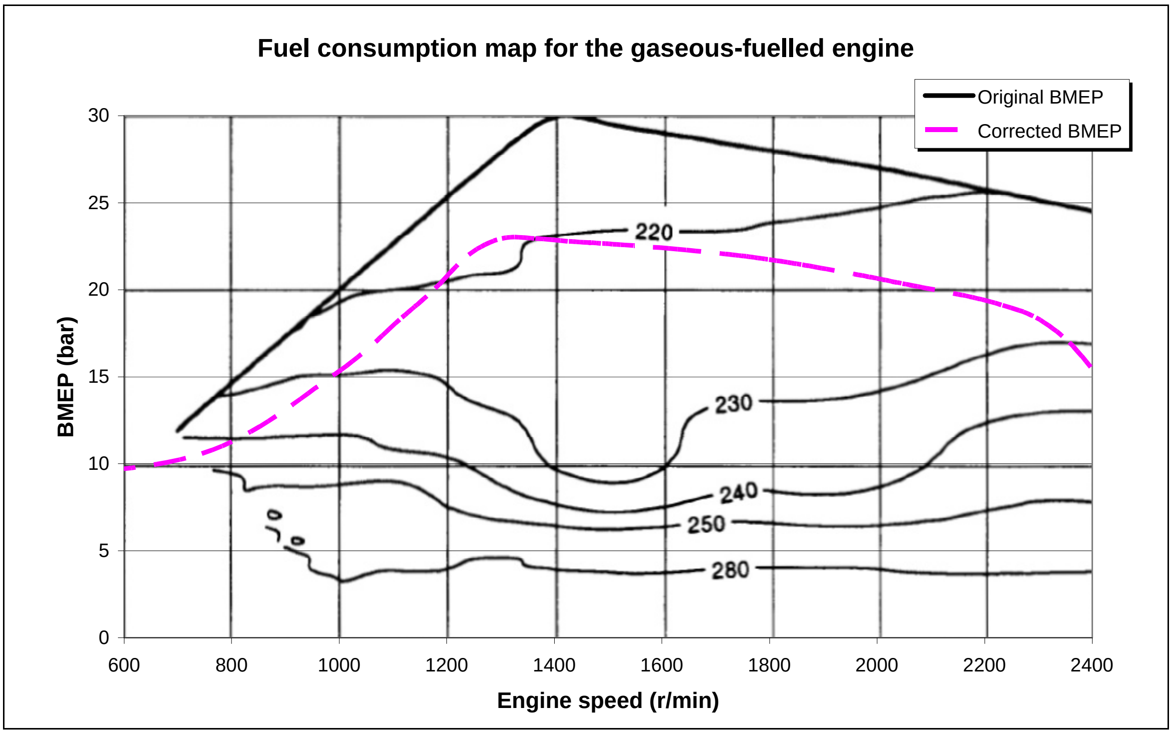

Figure 6. *Fuel consumption map for the gaseous-fuelled engine*

As can be seen in **Figure 6**, the minimum fuel consumption of the FEV engine is 220 g/kWh, which is quite an impressive level for a spark ignited engine.

To obtain a better comparison to the Volvo 13 litre engine previously shown, the power and torque curves have been corrected to a 13 litre engine using the same piston speed as for the 6 litre engine. The power and torque is shown in **Figure 7**.

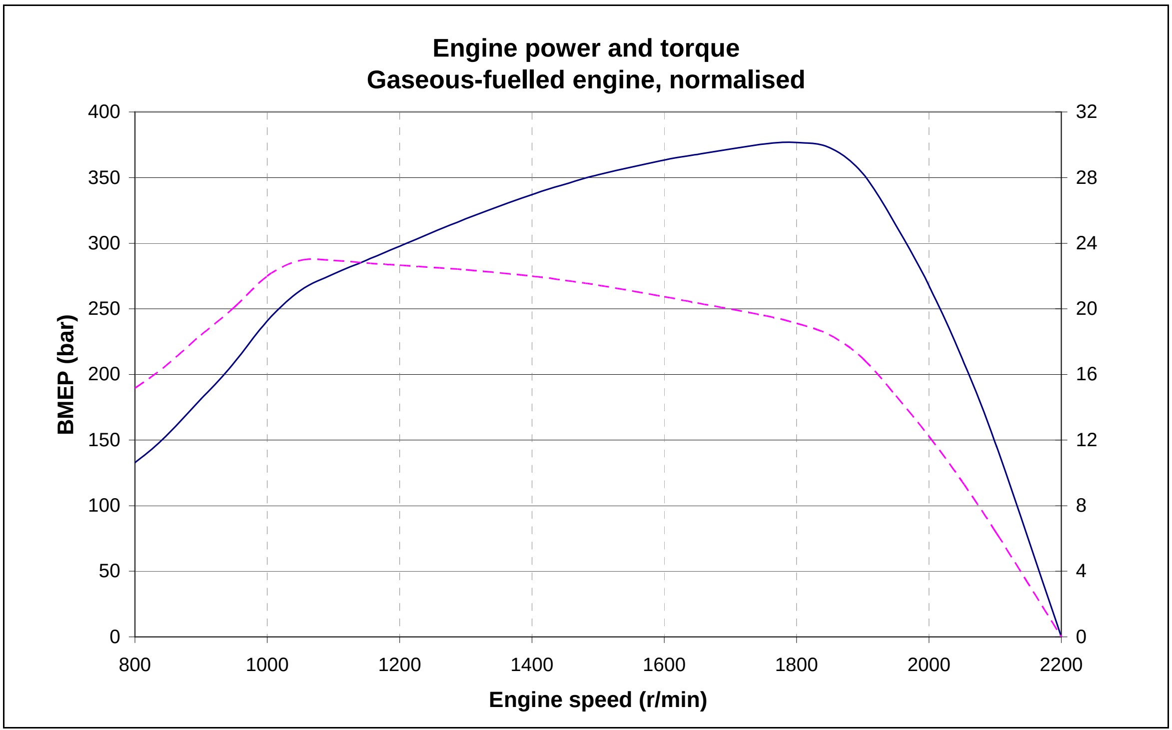

Figure 7. *The gaseous-fuelled engine*

To match the performance of the baseline diesel engines in the two vehicles studied, both the gaseous-fuelled engines had to be scaled up somewhat to compensate for the greater fuel load in comparison to the diesel baseline vehicles.

4.3 Vehicle technology

During the last 10 to 15 years numerous different official and non-official driving cycles have been developed. Many of those emanate from data logging of traffic patterns in real traffic in different environments. Also the legislative driving cycles of US and Europe have been further developed and extended in order to be more representative for real traffic. There is a tendency towards more and more “realistic” driving cycles.

In the following, a couple of driving cycles are presented and their specific characteristics are discussed. In **Table 6**, a short description of the transient driving cycles is made, and the driving cycles are described in more detail in the following text. We will not, on the other hand, go into any detail on the technique of deriving driving cycles based on data from real life traffic.

4.3.1 Driving cycles

Several potential driving cycle candidates to be used in this study are available. In the Advisor drive cycle library, supplemented with a couple of driving cycles by the author, more than 68 test cycles are available. However, relatively few of them are useful for heavy-duty vehicles.

Volvo did not have any suitable "own" driving cycle available so the choice had to be made among the available test cycles.

Heavy-duty vehicles are operating under highly varying conditions. It is difficult to find a test cycle that would be representative for all kind of heavy-duty vehicle types and in all operating conditions. A driving cycle and test procedure intended for legislative purposes have to be very well defined with regard to speed, time, setting of the dynamometer, temperature and humidity, preparation of the vehicle, test fuel etc., i.e. the degrees of freedom are relatively limited. Test cycles intended for characterising real-world emissions can provide additional degrees of freedom in this respect. Early on in this study it was decided not to solely rely on the legislative test cycles but to add at least one test cycle candidate as a supplement. The objective here was simply to see if the results would change significantly by substituting the test cycle.

It is of interest to discuss the test cycle candidates in some more detail. Therefore, an overview of some of the potential candidate driving cycles has been carried out below. Some characteristics of the driving cycles under discussion are listed in **Table 6**.

Braunschweig city driving cycle

The Braunschweig cycle was created at the Technical University of Braunschweig, Germany. It is a transient driving cycle for a chassis dynamometer, which simulates city bus driving with frequent starts and stops (**Figure 8**).

The Braunschweig driving cycle has been extensively used in projects at the test laboratory of AVL MTC, and therefore, it is very useful for comparisons. It represents tougher driving conditions compared to, for instance, the European Transient Cycle, and therefore, it is more useful in a case where the harmful emission components in the exhausts are of interest.

European Transient Cycle (ETC) and the Fige cycle

The ETC test cycle for engine dynamometers has been in use since late 2000 (Euro III) and it is used for emission certification of heavy-duty engines in Europe. The FIGE[5] Institute (now part of TÜV) in Aachen, Germany developed this test cycle. The test cycle was based on real-world road measurements on heavy-duty vehicles.

As the ETC cycle was derived from logged vehicle data, there is also a chassis dynamometer version of this test cycle available. Sometimes, this cycle is called the "FIGE" cycle to distinguish it from the ETC cycle, which is used for type approvals of heavy-duty engines. In **Figure 9**, the chassis dynamometer version of the cycle is shown.

Central business district (CBD) cycle

In the USA, the central business district (CBD) test cycle has been extensively used for in-use testing of heavy-duty vehicles. In particular, the West Virginia University (WVU) has been using this test cycle on their mobile heavy-duty chassis dynamometer. Similarly, the CBD test cycle has also been used in vehicle simulations. The CBD cycle is shown in **Figure 10**.

As can be seen in **Figure 10**, the CBD test cycle has few variations. The CBD driving cycle is not particularly representative for driving in European cities. Therefore, this option among the potential test cycle candidates for simulating city driving of a bus was not considered further.

[5] FIGE GmbH: Forschungsinstitut Geräusche und Erschütterungen. This institute is now part of TÜV Automotive in Germany.

***Table 6**. Some of the driving cycles for testing on chassis dynamometers*

Driving Cycle	Driving distance (km)	Max speed (km/h)	Average speed (km/h)	Duration (s)	Description
Braunschweig city driving cycle	10,9	58,2	22,9	1740	A transient driving cycle for chassis dynamometer simulating urban bus driving with frequent stops.
European Transient Cycle (ETC), simulated on chassis dyn., FIGE	29,5	Urban: 50	Rural: 72 Motorway: 88	1800	Engine dynamometer test originating from vehicle loggings. The three different parts of the cycle includes urban, rural and motorway driving.
Central Business District (CBD)	3,22	32,2	20,2	560	This driving cycle represents a so-called sawtooth driving pattern, composed of repetitions with idle, acceleration, cruise and deceleration modes.
World Harmonised Vehicle Cycle (WHVC), whole test cycle	20,074			1800	The chassis dynamometer version of the proposed World Harmonised Duty Cycle (WHDC) for testing of heavy-duty engines.
WHVC urban	5,322	66,2	21,3	900	Urban part of the WHVC
WHVC rural	5,827	75,9	43,6	481	Rural part of the WHVC.
WHVC motorway	8,926	87,8	76,7	419	Motorway part of the WHVC. This cycle starts at a certain speed (not from stand still)
WHVC rural and motorway	14,752	87,8	59,0	900	Rural and motorway part of the WHVC.
WHVC selection	11,992	87,8	70,8	610	From of the WHVC test cycle, part of the rural and the whole motorway part were chosen.

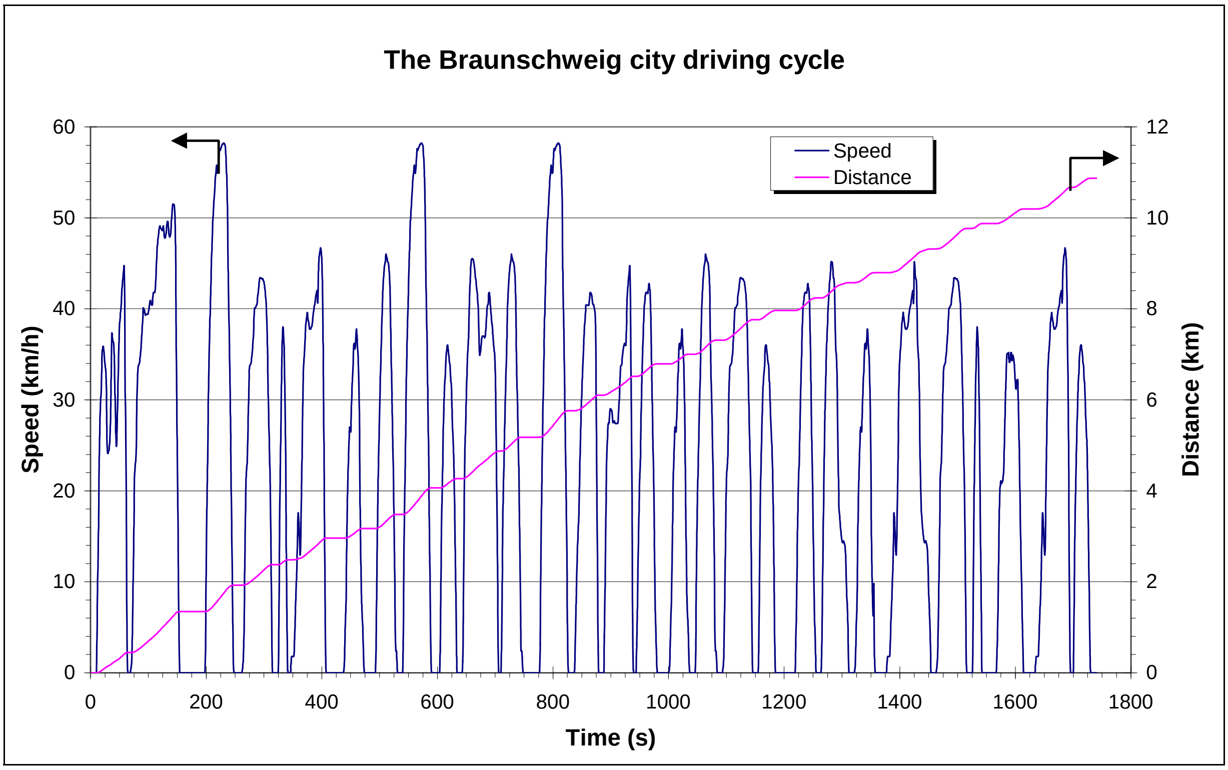

Figure 8. *The Braunschweig city driving cycle*

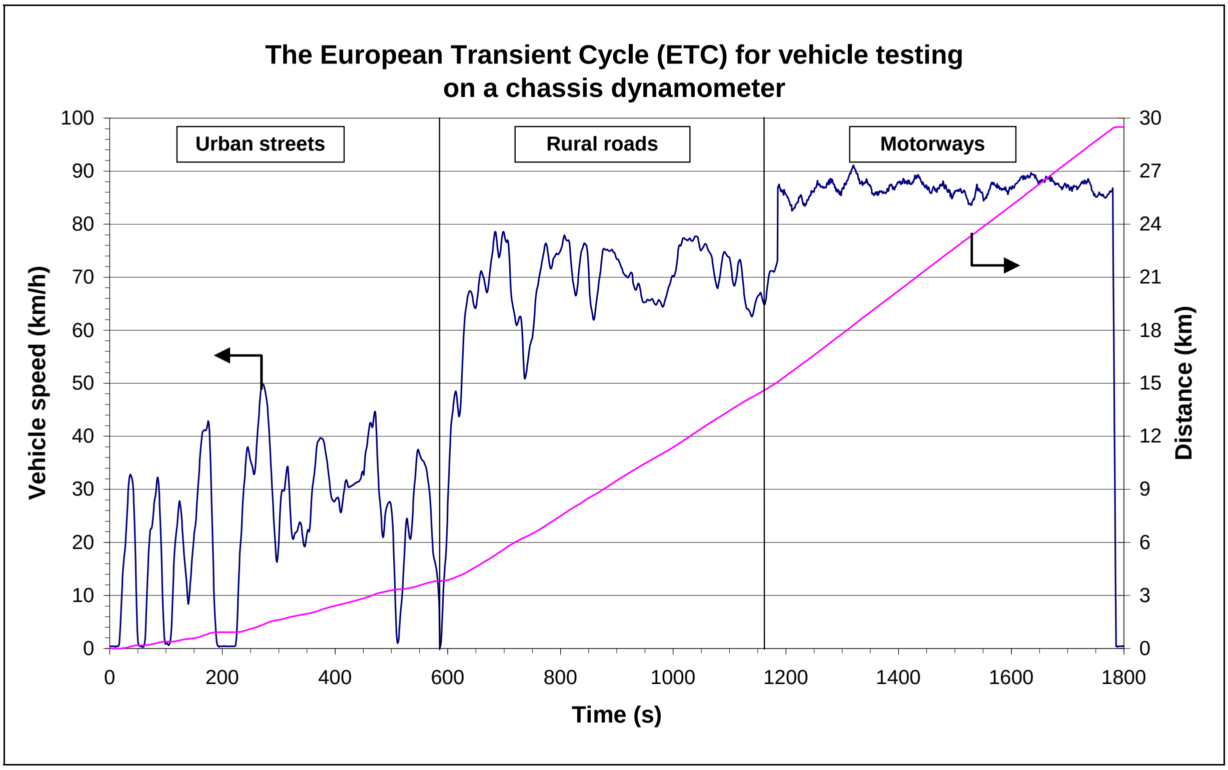

Figure 9. *The Fige cycle; the chassis dynamometer version of the European transient cycle (ETC)*

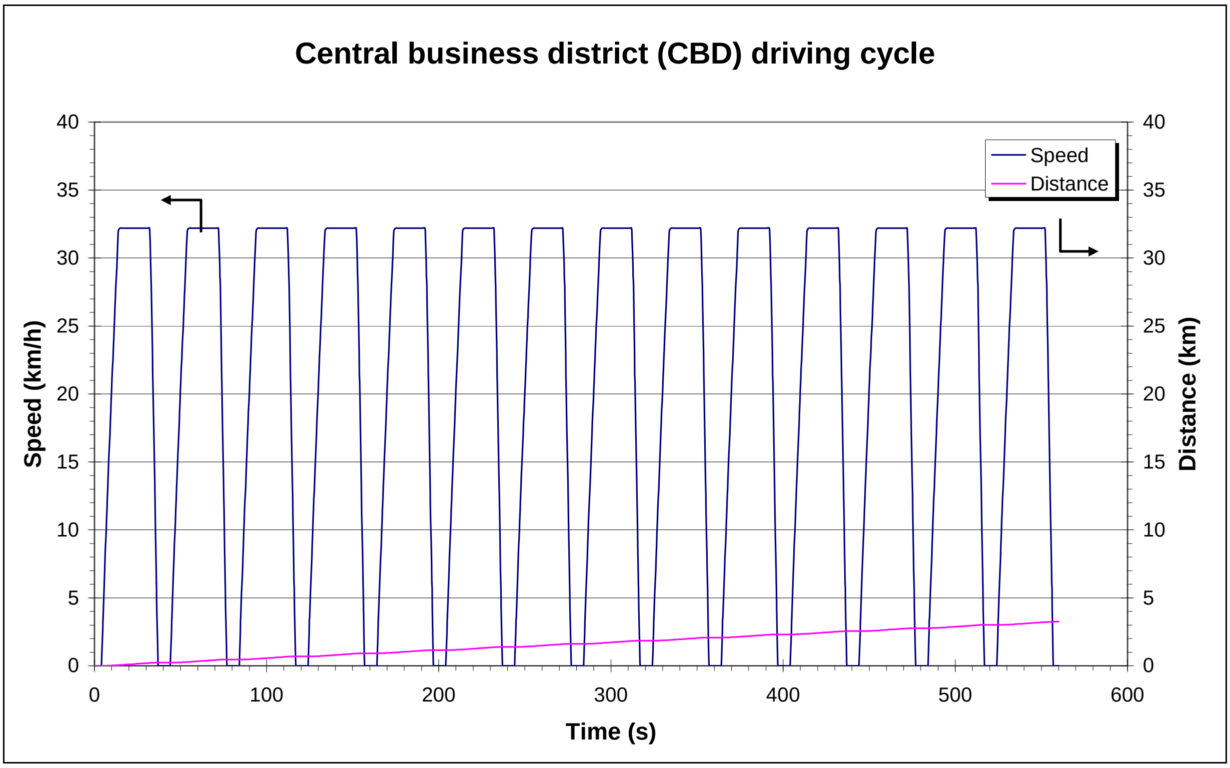

Figure 10. *Central business district (CBD) cycle*

World Harmonised Vehicle Cycle (WHVC)

A new World Harmonised Duty Cycle (WHDC) has been proposed for future use on an international level (e.g. EU, USA and Japan). This cycle was calculated based on extensive logging from heavy-duty vehicles in many countries. Eventually, this test cycle will probably replace the ETC test cycle in the EU. Presumably, this is the test cycle, which is most representative of driving with heavy-duty vehicles in the EU.

Although likely to find international acceptance for heavy-duty vehicles in general, the relevance for long-distance lorries of the WHDC test cycle could be discussed. Urban and rural driving together comprise a relatively large part of the test cycle. The most representative part for a long-distance lorry used in transport between the EU member states would be the motorway part of the test cycle. A combination of both the rural and motorway part could also be considered, if more local transport in rural areas has a greater dominance. The use of only the motorway part of the test cycle is not practical for simulations in Advisor, since this part of the test cycle does not start from standstill but rather has a smooth transition from the rural part. Presumably, this problem could be overcome by setting the initial conditions in the simulations. However, another alternative was chosen here. A selection of part of the rural cycle was made in addition to the motorway part so that the whole test cycle starts from standstill in the rural part and ends with the end of the motorway part. In total, the length of the whole selection from the WHVC test cycle was 610 s. To obtain the complete picture of driving cycle impact, simulations were carried out for both the whole WHVC test cycle and the selection of it. The WHVC test cycle and the selected part of the WHVC are depicted in **Figure 11**.

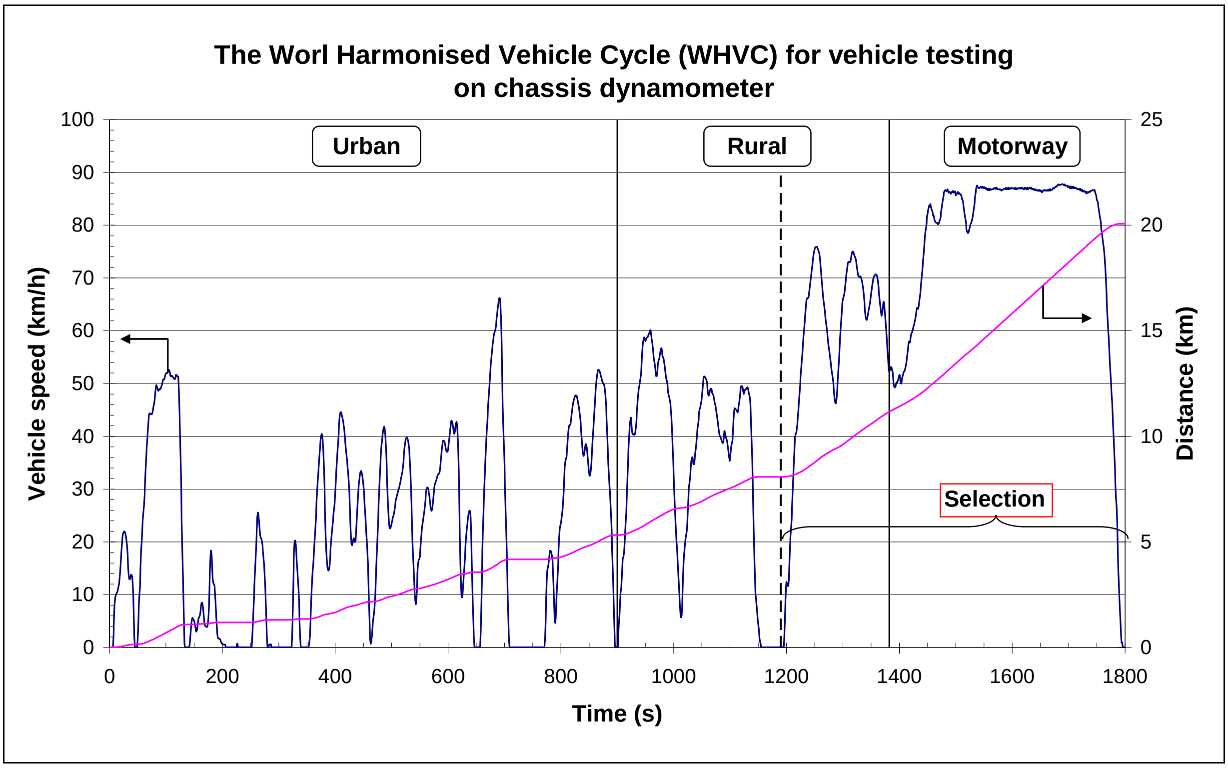

Figure 11. *The World Harmonised Duty Cycle (for chassis dynamometer testing)*

Route versus driving cycle

Previously, only traditional driving cycles have been discussed. It should be noted that specifying speed as a function of time is not the only way of setting up a test cycle. A "route" is another possibility. In this case, the vehicle speed is a function of the distance instead of time.

One example of a test schedule that is a route is the City-Suburban Heavy Vehicle Route (CSHVR). This schedule was developed by the West Virginia University as a simulation for heavy delivery trucks [54]. It has also been considered representative for the driving pattern of school buses in the USA. The CSHVR is based on driving distance and hence, it should be denoted a route, not a cycle. A time-based version of the route is also available, which should be referred to as CSHVC.

The CSHVR route was developed through the recording of speed and distance data on two heavy delivery trucks. A videotape record was also collected so that the speed and distance database could be separated into microtrips. A microtrip was defined as a burst of driving activity, typically due to driving from one delivery site to another. Free accelerations were exhibited in the test schedule and on the driver's aid by converting an acceleration ramp into an instantaneous speed jump to the desired speed. Since the accelerations are free, the scheduled route speed is now a function of distance travelled and not a function of time [54].

The rationale for using a route instead of a driving cycle is that it could take into account the driving style of the driver and the variations between vehicles much better than in the driving cycle case. For example, it is well-known that a sports car is driven quite differently from a small underpowered car. The latter vehicle might not even be capable of following a though driving cycle resulting in that the accumulated distance is less than for a more powerful car.

In the simulations carried out in this project, it was found that the vehicles could not follow the test cycles under all conditions. A well-known problem is a part of the ETC test cycle where the acceleration demand is extremely high. However, this problem should perhaps be considered as a “glitch” of the test cycle itself and not a problem related to the vehicle. There were also problems in the other test cycles. For example, the city bus and the fully loaded truck could not follow all parts of the test cycles. The long-distance lorry was the vehicle with the smallest problems in this respect. When the vehicle cannot follow the test cycle, the accumulated distance will be lower than the nominal distance. The rationale for using a route would be that the distance covered would be the same in all cases. However, the use of a route is not so easily facilitated in the Advisor programme. Therefore, this option had to be omitted.

Selected test cycles

Based on the discussion above, the selected test cycles are summarised in ***Table 7*** below. The “common” test cycles between the simulations carried out on the two vehicle categories are the ETC and WHDC test cycles. However, it should be noted that only a few results from the whole WHDC test cycle have been reported. This is indicated by the parenthesis in ***Table 7***.

Table 7. *Selected test cycles*

Vehicle	Braunschweig	ETC	WHDC	WHDC selection
City bus	Yes	Yes	(No)	No
Long-distance lorry	No	Yes	(Yes)[a]	Yes

Note:

[a] The “Yes” within parenthesis indicates that simulations have been carried out in this test cycle but that the results have only been partly reported in the present report.

4.4 Vehicle simulation

Vehicle simulations using the Advisor software has been discussed elsewhere and also mentioned in this report. In this section, some of the conditions and observations in the vehicle simulations are reported and discussed.

4.4.1 Influence of aftertreatment on fuel consumption

Urea consumption

As mentioned above, the diesel baseline engine uses SCR with a urea additive (Adblue®) for NO_X aftertreatment. The consumption of urea is taken into account by adding the energy used in urea production to the fuel consumption. A matrix for NO_X emissions as a function of speed and load was available for the baseline diesel engine. Thus, the NO_X emissions in the various test cycles could be calculated. A target NO_X level with considerable engineering margin to the limit of 2 g/kWh in the ETC test cycle (and similarly in the ESC test) would determine the urea consumption. In practice, the chassis dynamometer version of the ETC test, as conducted in the simulations, does not provide exactly similar conditions as in the “real” engine dynamometer ETC test. However, this was the only practical way of determine the urea consumption without substantial modifications of the Advisor model. In order to meet the engineering target, a NO_X reduction level of approximately 75 % was needed in the ETC test. This yielded a urea consumption of slightly less than 4,5 % of the fuel consumption. For practical reasons, this percentage was used for all test cycles, reflecting that the relative NO_X reduction would be equal under all these driving conditions. This is of course a crude estimate but since the energy used for urea production was low in comparison to the fuel consumption and the energy used in fuel production, this approximation was justified. In energy terms, the urea production equals an increase in fuel consumption by 0,7 %. Other effects associated with the use of SCR besides the urea production, such as, e.g. increased weight or reduced load capacity, are already taken into account in the simulations.

Note that while the energy used for urea production is actually applied upstream of the vehicle, this could be reported together with the energy used for fuel production. However, for practical reasons, in this section it is reported together with the energy used in the vehicle, in a similar way as the impact of the particle filter.

FTD and Rape seed Methyl Ester (RME[6]) have different engine-out NO_X level compared to diesel oil; i.e. lower in the FTD case and higher in the RME case. This could be taken into account by optimising the engine for each fuel. One option could be to chance the mapping for injection timing, i.e. to use retarded (RME) or more advanced (FTD) injection, to achieve similar NO_X level as for diesel fuel. If the engine was equipped with EGR – which is not the case here – the EGR strategy could be changed instead to maintain baseline NO_X. Data for this optimisation was not available, so another option had to be used instead. Since all these fuels would use SCR aftertreatment, it is plausible to change urea dosing to achieve the desired NO_X level. This might not always be possible for RME, which has a higher engine-out NO_X level than diesel fuel, without also increasing the catalyst size or making some other hardware modification. However, it was anticipated to be an option here. To compensate for the difference in engine-out NO_X level, 10 % less urea was anticipated for FTD and a 10 % increase in urea was foreseen for RME.

It was anticipated that the gaseous-fuelled diesel engine with glow plug ignition would also have to use SCR to meet the NO_X target although the engine-out NO_X level is somewhat lower than for diesel fuel. There is not so much information about this in the open literature but information from the USA suggests that aftertreatment must be used to meet NO_X and PM limits for 2010. Without EGR, US certified production engines have not been able to meet a NO_X level as low as Euro V, which is also an indication for that aftertreatment is needed for NO_X reduction. EGR could also be an option but was not foreseen here. A NO_X level corresponding to 30 % lower urea injection compared to the diesel-fuelled engine was assumed for the glow-plug gaseous-fuelled engine.

Particle filter

There is an impact on fuel consumption from the particulate filter anticipated to be necessary to reduce particle emissions to very low level for the engines using diesel fuel or “similar” fuels, such as RME and FTD. The engines for all other fuels were anticipated not to have to use a particulate filter. Fuel consumption can be affected in two different ways due to the use of a particulate filter. First, there is an increase in exhaust back pressure due to the pressure drop in the filter. This pressure drop varies with soot load in the particulate filter. Second, there is a fuel consumption penalty if active regeneration of the particle filter is used. It is possible to use a model of a particulate filter in Advisor. However, this model is simple and cannot take active regeneration into account. Therefore, a constant fuel penalty of 2 % has been added on the simulated fuel consumptions.

4.4.2 Influence of glow plug power on fuel consumption

For an engine running at part loads on a low-cetane fuel, e.g. alcohol or methane, it is necessary to assist the combustion with glow plugs fitted into the cylinder and in close position to

[6] Most other vegetable methyl and ethyl esters have similar properties as RME regarding NO_X formation.

the spray. The influence on engine specific fuel consumption from the extra power consumed by the glow plugs is estimated by means of simulation. This methodology had to be chosen due to lack of experimental data for modern engines.

Model

Some criteria for calculating the necessary glow-plug power can be made. For example, at high load, the glow plug is heated by combustion, so no additional electric power will be necessary. Based on engineering experience, the following criteria can be made:

- The glow power increases with engine speed due to the increased cooling flow through the cylinder.
- The glow power decreases with increased torque due to an increased temperature of gas in the cylinder.

The model is visualized schematically in **Figure 12**. As can be seen in this figure, increasing engine speed requires an increase in glow-plug power. Higher torque results in less need for glow power and at a certain torque, no glow-plug power at all will be needed.

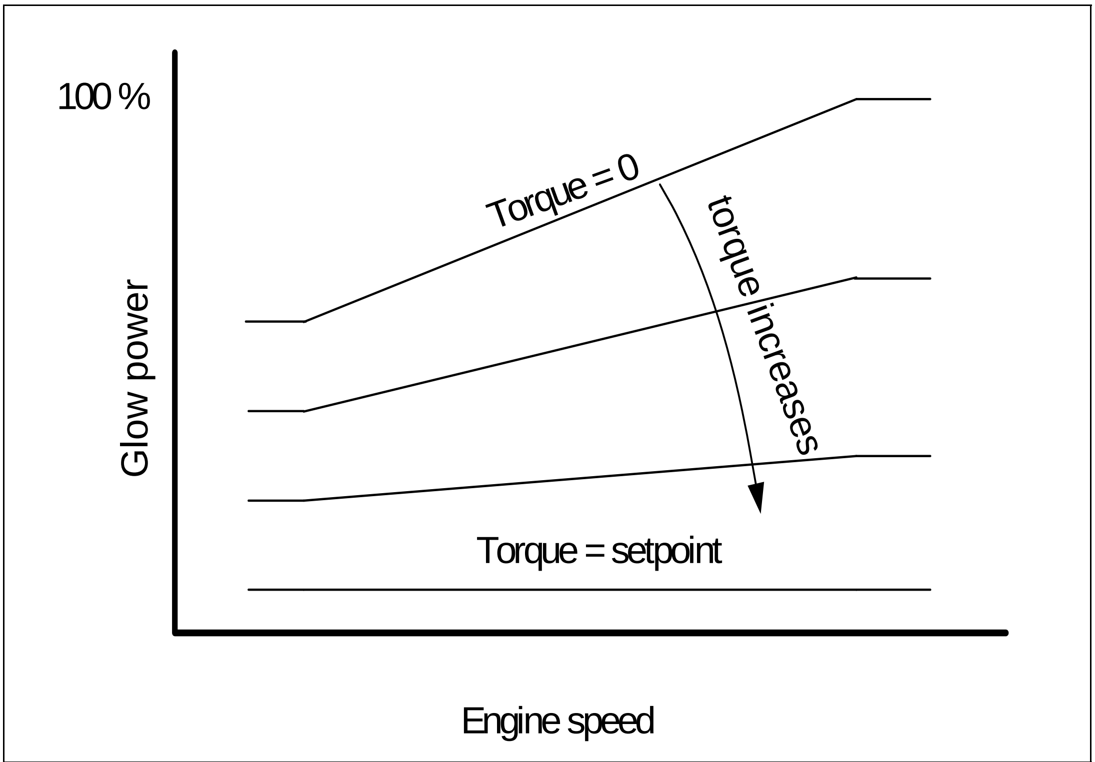

Figure 12. *Schematic presentation of glow-plug power requirement*

The glow plug power is calibrated at different set points according to:

- High set point: where the glow power is as its maximum and will remain at this power if the speed further increases. The maximum power of the glow plug sets this limit.
- Low set point: where the glow power is set to its lowest value.
- Break set point: Where the glow power should start to increase with speed. Between Low set point the glow power is constant.

- Torque set point: where the glow power should be terminated regardless of the speed.

Calculation results

The input is given from a chart where the brake specific fuel consumption (BSFC) at different speed and torque points are given from a test cycle with a diesel fuelled engine. At each point the glow power and the new BSFC, as well as the percentage change in BSFC are calculated assuming the fuel flow to the engine is unchanged. The efficiency for producing the electrical power in the alternator from engine's mechanical power to the glow plug is set 50 % for all points.

The results from calculations with various different set points for the glow plug power showed that the impact of the set points on BSFC was relatively small. The set points chosen are shown in **Table 8**. The set point for the torque is slightly below 50 % load.

Table 8. *Set points for the glow power*

Engine	*Low set point watt/speed*	*Break set Point (r/min)*	*High set point watt/speed*	*Torque set point (Nm)*
Volvo 13 litre	300/600	800	600/1800	1000

4.4.3 Auxiliary load

In Advisor, default data for heavy-duty vehicles, two large power consuming accessories are used, i.e. the air compressor for supplying pressurised air for auxiliary systems and the compressor for the air conditioning system. A constant load of 3,7 kW each (5 hp) is applied for both these compressors all the time. Although this might be appropriate in the first case, air conditioning is not used as frequently in the Nordic countries. Therefore, factors corresponding to 10 % use in the lorry case and 20 % in the city bus case have been used instead.

4.4.4 Optimisation of the final drive ratio

City bus

No particular optimisation of the final drive ratio was carried out for the diesel-fuelled city bus baseline vehicle. The final drive ratios for the engines which were significantly different in cylinder capacity, e.g. the alcohol engines with ignition improver, were scaled by piston speed. There was one exception which required additional work for optimisation of the final drive ratio and this was the otto cycle gaseous-fuelled engine. This engine was relatively small due to the very high power density but had a drawback of lower torque at low speed compared to the diesel counterparts. Furthermore, the gaseous-fuelled engine was run at higher piston speeds. Likewise, the relative increase in fuel consumption with speed was not so pronounced as for the diesel engines. An optimisation was made by varying the final drive ration in increments of 1 per cent and the engine power by increments of 1 kW. After a considerable number of optimisations, the optimum final drive ratio was found to be 8 % higher than for the diesel-fuelled baseline vehicle.

Long-distance lorry

Similarly to the bus, an optimisation of the final drive ratio had to be carried out on the otto-cycle methane engine. In this case, the optimal compromise between engine size scaling and final drive ratio was found at similar power as for the diesel baseline (353 kW) and a final drive ratio scaled by the stroke of engines (165/170,3) but multiplied by a factor of 1,06 compared to the baseline engine.

4.4.5 Fuel consumption

The fuel consumption for all vehicles were simulated using the same engine back pressure, i.e. assuming similar pressure drop for all exhaust aftertreatment devices. Although this assumption might be valid for catalyst, a DPF has a greater negative impact on the fuel consumption. First, the pressure drop is higher than for an oxidation catalyst and furthermore, it is also increasing somewhat with odometer reading due to ash accumulation in the filter. Second, an active regeneration strategy, which might be necessary to ensure proper DPF regeneration at all possible driving conditions, will also have a negative impact on the fuel consumption. The sum of these effects is assumed to be 2 %. This effect is shown separately in the diagrams below. The abbreviations used in the figures below are listed in **Table 9**.

Table 9. *Abbreviations*

Abbreviation	*Fuel*	*Engine/combustion technology*
DO-CI	Diesel oil	Diesel cycle, compression ignition
EtOH-GP	Ethanol (neat with additives)	Diesel cycle, glow-plug ignition
EtOH-II	Ethanol w. ignition improver	Diesel cycle, compression ignition
MeOH-GP	Methanol (neat with additives)	Diesel cycle, glow-plug ignition
MeOH-II	Methanol w. ignition improver	Diesel cycle, compression ignition
DME-CI	DME	Diesel cycle, compression ignition
CBG-SI	Methane (CNG, CBG and SNG)	Otto cycle, spark plug ignition
CBG-GP	Methane (CNG, CBG and SNG)	Diesel cycle, glow-plug ignition

The two other fuel options for the conventional diesel engine (FTD and RME) besides diesel oil are not shown in the following, since the only difference compared to diesel oil is a difference in urea injection.

As indicated in **Table 9**, alcohol fuels for the glow-plug option would use some additives such as, e.g. denaturants to avoid ingestion but, since the concentrations of these additives would be very low concentrations, the alcohols could be considered as neat alcohols. DME would also need some additives for e.g., improved lubricity. However, the blending level of these additives would be very low (at ppm level). As discussed above, the biggest difference between fuel specifications would be for the alcohols with ignition improvers compared to the neat alcohols.

City bus

In order to understand the results for fuel consumption presented below, the impact on vehicle test weight should be discussed first. Due to the different energy converters and fuels, the vehicle weight will differ from case to case. As described previously, the engine has to be scaled to obtain similar performance so there will also be a difference in engine power. These data for the city buses are shown in **Table 10**.

Table 10. *Impact on vehicle test weight and engine power*

Engine type	Fuel	Ignition	Weight (kg)t	Power (kW)
Diesel	Diesel oil/FTD/RME	CI	12 865	187
Diesel	Ethanol (neat)	GP	12 937	187
Diesel	Ethanol (ign. imp.)	CI	13 279	197
Diesel	Methanol (neat)	GP	13 024	189
Diesel	Methanol (ign. imp.)	CI	13 368	199
Diesel	DME	CI	13 049	190
Otto	CNG/CBG/CSNG	SI	13 456	219
Diesel	CNG/CBG/CSNG	GP	13 383	193

The minute difference in fuel tank weight due to smaller differences in fuel specification has been neglected. Thus, diesel oil, FTD and RME are treated no differently and the same principle applies for the three gaseous fuels. Hardly surprising, the heaviest bus is the one using a spark ignition engine running on gaseous fuels. The diesel version of the gas engine with glow plug and the methanol engine follow closely. There is a very close match in engine power for most of the alternatives. Upscaling of the alcohol engines due to lower power density compared to the glow-plug engines increases weigh and requires more power to maintain performance. The highest power is required for the spark-ignited gas engine. The reason is probably a different torque curve for this engine compared to the other engines, since optimisation of the final drive, which was tried, could not reduce the power below this level, without compromise with fuel consumption.

As previously mentioned, the maximum number of passengers for all the fuel options in the city bus was similar for all fuels and so would the average number of passengers be. Therefore, plots of fuel consumption per km or per passenger km would yield similar results regarding relative differences between the various options. Thus, only the fuel consumption per km is presented in this case.

In **Figure 13**, the fuel consumption for the city bus is depicted. The fuel consumption for all fuels is shown as *diesel equivalent* for the sake of easier comparison. As described above, the impact of DPF and SCR are shown separately for the engines that use this aftertreatment technology. The baseline for comparison in each of the drive cycle is the compression ignition diesel engine running on diesel oil as fuel (DO-CI).

Compared to the diesel-fuelled engine, the ethanol engine with glow plug has lower fuel consumption. In fact, the fuel consumption is lower for ethanol than any other fuel. The efficiency is slightly lower due to the glow-plug power and higher weight but this is more than compensated for by the lack of DPF and urea production for SCR aftertreatment. The relative difference, e.g. at about 2,2 % in the Braunschweig test cycle, is indeed small, but still significant. To put this in perspective, the yearly average reduction of fuel consumption due to engine development used to be in the order of 0,5 % per year. This development trend has been somewhat distorted due to the emission legislation during the last two decades, where a compromise between emission level (NO_X) and fuel consumption had to be made. This resulted in smaller improvement of fuel consumption than the previous trend of 0,5 % per year.

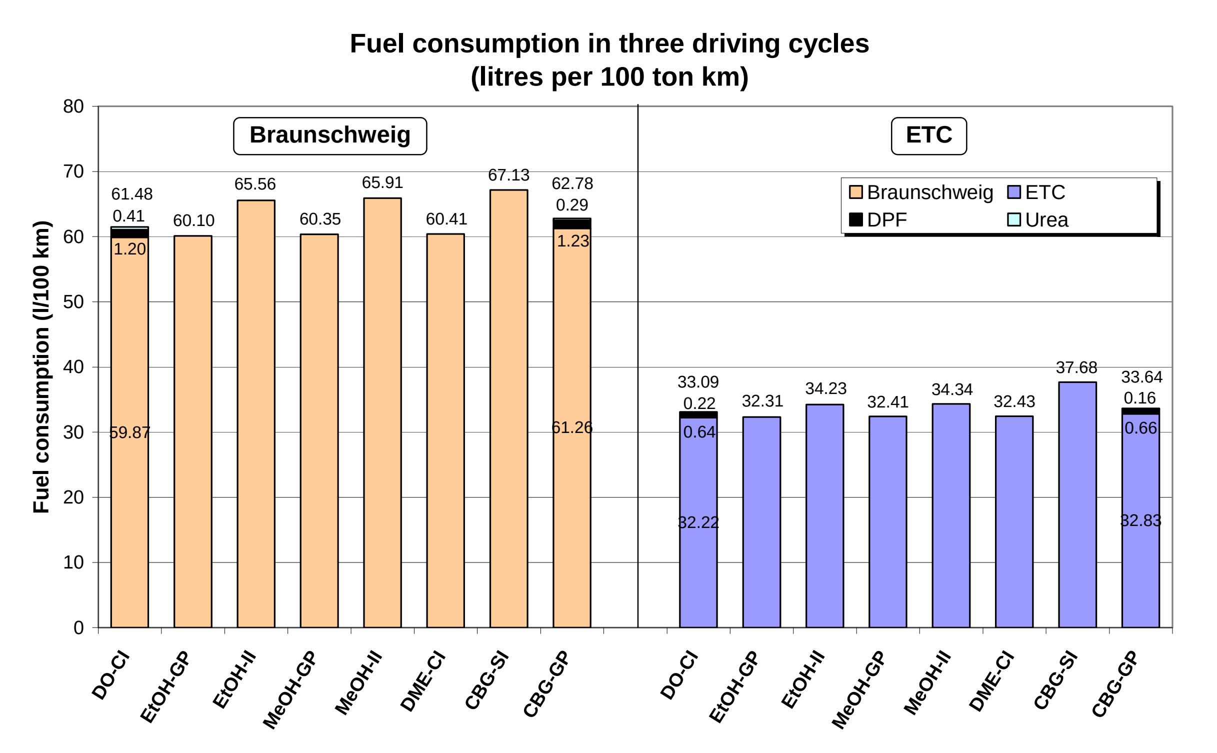

***Figure 13**. Fuel consumption (l/100 km) for the city bus*

Also the glow-plug methanol engine shows a similar trend as the ethanol engine, albeit at a marginally higher level of fuel consumption. The difference between the two alcohols is due to the heavier fuel tank for methanol in comparison to ethanol.

The alcohol engines using ignition improver have higher fuel consumption than the glow-plug equipped counterparts. The relative difference for ethanol is about 9 % higher fuel consumption for the engine with ignition improver in the Braunschweig test cycle. This is attributed to the lower energy density of the engine which necessitates upscaling of the engine and the resulting higher part load fuel consumption and the greater weight of the vehicle. The relative difference, at 6 % is lower in the ETC test cycle. The reason for this difference has not been investigated in detail. The average engine load is lower in the ETC test, which should give a greater difference but, on the other hand, the higher vehicle weight for the engine with ignition improver has a greater impact on the frequent accelerations in the Braunschweig test cycle. The sum of these effects seems to be dominated by the greater weight resulting in a slightly smaller relative difference between the ethanol options in the ETC test cycle. A similar effect as for ethanol can be seen also for methanol.

DME has fuel consumption very similar to methanol with glow plug. The DME engine has higher efficiency at part load due to the absence of glow plug power but it is hampered by slightly greater vehicle weight than the methanol engine. It is interesting to note that such a small difference in weight (25 kg) can be distinguished in the vehicle simulations.

Hardly surprising, the gaseous fuelled bus with the spark-ignited engine has the highest fuel consumption of all options investigated. The relative difference compared to diesel fuel is +9,2 % and +11,4 % in the Braunschweig and ETC tests respectively. It is actually surprising that the difference compared to the diesel-fuelled baseline is not greater. It is not uncommon to see a difference in the order of 20 to 30 % between a diesel bus and a methane-fuelled bus. The explanation here is that the lean-burn engine selected is so advanced. It has a very high power density and high efficiency. Often gaseous-fuelled engines have to be larger than their diesel-fuelled counterparts (e.g. 10 litres compared to 7 litres) due to a lower power density. Note that the gaseous fuelled engine used in the simulations here is not a production engine but it illustrates the potential for this technology. If a real production engine had been chosen, the difference could have been greater but, on the other hand, the comparison made here could be considered more “fair” for the gaseous fuels. As noted above, the difference in fuel consumption compared to diesel fuel was somewhat higher in the ETC test (+11,4 %) than in the Braunschweig cycle (+9,2 %). The probable cause for this observation is that the engine load is lower in the ETC cycle. A spark ignited engine has lower efficiency at part load compared to a diesel engine and this trend will be more pronounced in the ETC cycle than the Braunschweig cycle.

The fuel consumption for the gaseous-fuelled engine using a diesel cycle with glow plug is lower than for the spark ignited engine. This is due to the high efficiency of the diesel cycle. The efficiency is hampered somewhat by the need for a DPF and urea for exhaust aftertreatment but still the absolute level is very good. This shows that overall efficiency close to a diesel-fuelled engine can be achieved also with gaseous fuels, if this engine concept is developed. Probably this is the rationale also behind the development of dual-fuel engines that has the potential of achieving similar energy efficiency as the glow-plug concept investigated here.

Finally, the relative differences between the two investigated driving cycles are seen clearly. The fuel consumption is much higher in the Braunschweig cycle compared to the ETC cycle due to more frequent stops and harder accelerations.

Long-distance lorry

In contrast to the bus, the gross weight was set to 40 tonnes and any difference in drivetrain and tank weight has to be compensated by lower load capacity and consequently, a lower average payload. These weights are not listed here but the differences between options in this respect should be considered when the results below are discussed.

In **Figure 14**, the fuel consumption in litres per 100 km is shown and in **Figure 15**, the fuel consumption per ton km is shown taking the differences in payload capacity into account.

In contrast to the bus where compressed gases (CNG/CBG/CSNG) are used, liquefied cryogenic gases (LNG/LBG/LSNG) are used for the long-distance lorry. The longer range for a long-distance lorry necessitates this change, since the weight penalty would be too big for compressed gases. In the figures, LBG is used as denotation for all the three options, since the fuel consumption and energy use in the vehicle would be practically similar for all three gaseous fuels.

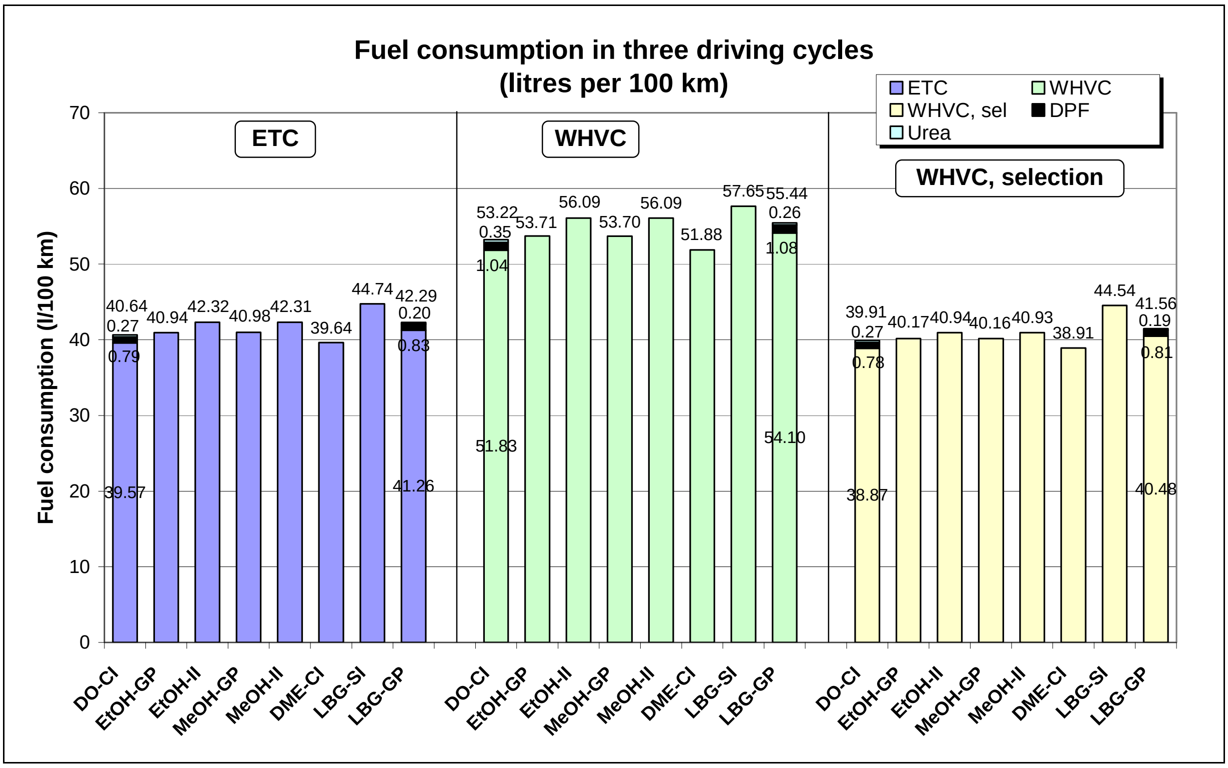

Figure 14. Fuel consumption (l/100 km) for the long-distance lorry (in diesel oil equivalents for the urea consumption)

Most of the results in **Figure 14** are similar to those already discussed above for the bus and need not to be repeated. Some remarks can be made anyway.

For the long-distance lorry, DME has slightly lower fuel consumption than the diesel baseline. The reason here is that the maximum gross weight is limited and the average weight is, in fact, somewhat lower for the DME-fuelled lorry than for diesel-fuelled lorry simply because the weight of DME per unit energy is higher and therefore, the weight with empty tank, as well as on average, will be lower for the DME lorry. On the other hand, the payload capacity will be lower, implying higher diesel-equivalent fuel consumption per ton km for DME, as discussed below. This is also the cause for the differences noted between the bus and the lorry.

In a comparison between the driving cycles, ETC and the selection from the WHVC are quite similar, while the whole WHVC shows considerably higher fuel consumption than both the other test cycles. Since the fuel consumption in “real-life” long-distance traffic seldom is as high as 50 l/100 km, the WHVC test cycle might be less representative for this kind of vehicles than the two other test cycles. Since the ETC test cycle is a full cycle in contrast to the selected part of the WHVC, the ETC is used for comparing the total WTW efficiency as discussed in the next section.

Much more interesting than the fuel consumption per distance driven (**Figure 14**) is the fuel consumption per ton km for the long-distance lorry. This is shown in **Figure 15**.

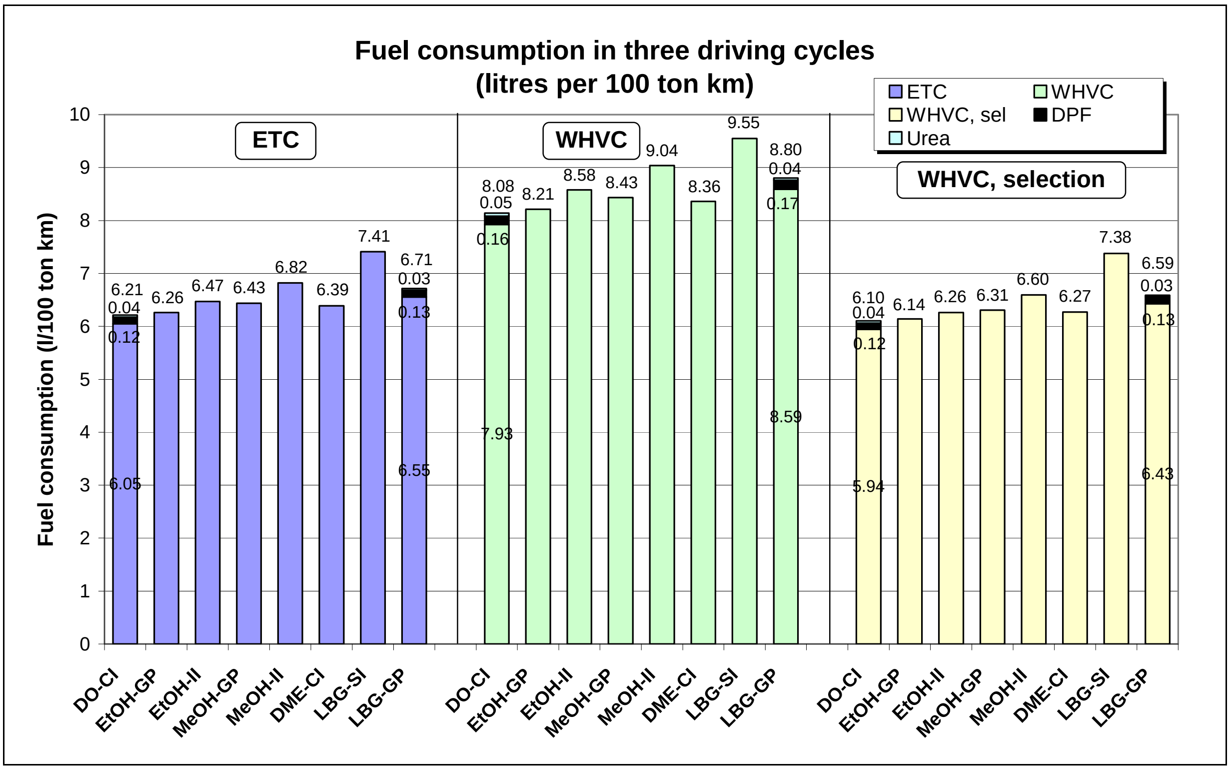

Figure 15. *Fuel consumption per ton of transported goods (l/100 ton km) for the long-distance lorry (in diesel oil equivalents for the urea consumption)*

When the differences in payload capacity are taken into account, the fuel options that are hampered by greater tank weight show less favourable results (**Figure 15**) compared to the previous graph (**Figure 14**). For example, in this case even DME, which was better than diesel fuel in the bus, has higher fuel consumption than the diesel baseline, although the difference (+2,9 %) is relatively small after all.

The greatest difference in relation to the diesel baseline can be seen for the spark ignited gaseous-fuelled engine, where the relative difference between these two options is +19,3 % higher for the gaseous-fuelled engine. The relative difference is almost twice as high as for the bus even if cryogenic gases are used for the lorry to reduce the weight penalty. The longer range of the long-distance lorry compared to the city bus still makes its tribute as a reduction in maximum payload.

Compared to the results for the bus, the alcohol fuels and DME show less favourable results in the long-distance lorry. The reason here is again the higher weight penalty for these fuels in the lorry. A small difference that can be noted for the lorry compared to the results for the bus in the Braunschweig cycle is that DME has lower fuel consumption than methanol in all investigated driving cycles. However, the differences compared to the glow-plug methanol engine are relatively small (<1 % in all driving cycles).

As for the bus, the alcohol engines using ignition improver always have higher fuel consumption than the glow-plug counterparts. However, the relative differences, e.g. +3,4 % for ethanol in the ETC test, are smaller for the lorry than for the bus (+5,9 %). The plausible explanation is that the power-to-weight ratio is much lower for the lorry, implying that the engine will operate at higher load than the engine in the bus. This compensates somewhat for the lower efficiency at part load of the larger engine for ethanol with ignition improver compared to the glow-plug engine, since the average load is shifted upwards. Another influence is the weight penalty that should be more pronounced for the lorry than the bus but this is counteracted by the load shift.

4.5 WTW efficiency

In this section the WTW results are obtained by adding WTT and TTW data. Note that the results on fuel production shown in **Figure 3** include only this particular step of the WTT chain and nothing else, while all the steps of the WTT chain are included in the results below.

The results for the two vehicle categories, i.e. buses and lorries, are reported separately. Not all possible combinations of fuel and drivelines or even all investigated combinations are reported here. Instead, a selection of some of the most interesting data is presented in the graphs below.

The feedstock/fuel/driveline combinations are grouped together in logical combinations. First, a distinction between intensive and extensive feedstock farming is made. For example, agricultural products usually belong to the former category while cellulosic matter belongs to the later category. The classification cannot be strict and the main purpose is to group options together in a couple of graphs. A certain kind of distinction between the diagrams is accomplished by using different colour codes for the bar. Regarding feedstock for biogas, waste and manure might originate from various feedstocks, where most is grown in intensive cultivation but usually, no “upstream” energy use and emissions are included in a well-to-wheel analysis for these feedstocks.

In the WTW summary, urea production is taken into account in the by adding this energy use to the fuel production (WTT) and thus, it is not part of the energy used in the car (WTW) as in a previous section.

Note that the scale on the y-axis is different for each vehicle category (i.e. MJ/km or MJ/100 ton km) for practical reasons. Diesel oil is shown separately as reference in all the diagrams below, although this fuel is produced from fossil feedstock and not from biomass.

4.5.1 City buses

In order to recall and to summarise the WTW chains investigated for the city bus, the list in **Table 11** has been compiled. The abbreviations are used in the figures below. Only the results from the Braunschweig cycle have been shown for the city bus. One reason is that this test cycle is more representative of city driving than the ETC cycle and the other reason is that the differences between these test cycles regarding fuel consumption has already been discussed.

Table 11. Driveline and fuel combinations for the bus

Cult.	Feedstock	Fuel	Ignition	Abbreviation
Foss	Crude oil	Diesel	CI	DO/CO/CI
Miscellaneous	Waste	CBG	SI	Wa/CBG/SI
	Waste	CBG	GP	Wa/CBG/GP
	Liquid manure	CBG	SI	LM/CBG/SI
	Liquid manure	CBG	GP	LM/CBG/GP
	Dry manure	CBG	SI	DM/CBG/SI
	Dry manure	CBG	GP	DM/CBG/GP
Intensive cultivation	Wheat/straw	EtOH	II	WS/EtOH/II
	Wheat/straw	EtOH	GP	WS/EtOH/GP
	Straw (Iogen)	EtOH	II	St/EtOH/II
	Straw (Iogen)	EtOH	GP	St/EtOH/GP
	Rape seed	FAME	CI	RO/FAME/CI
	Rape seed	FAEE	CI	RO/FAEE/CI
Extensive cultivation	Wood (SSCF)	EtOH	II	Wo/EtOH/II
	Wood (SSCF)	EtOH	GP	Wo/EtOH/GP
	Wood	FTD	CI	Wo/FTD/CI
	Wood	CSNG	SI	Wo/CSNG/SI
	Wood	CSNG	GP	Wo/CSNG/GP
	Wood	MeOH	II	Wo/MeOH/II
	Wood	MeOH	GP	Wo/MeOH/GP
	Wood	DME	CI	Wo/DME/CI
	Black Liquor	FTD	CI	BL/FTD/CI
	Black Liquor	CSNG	SI	BL/CSNG/SI
	Black Liquor	CSNG	GP	BL/CSNG/GP
	Black Liquor	MeOH	II	BL/MeOH/II
	Black Liquor	MeOH	GP	BL/MeOH/GP
	Black Liquor	DME	CI	BL/DME/CI

In **Figure 16**, results for the bus running on fuels produced via anaerobic digestion to biogas (waste, liquid and dry manure), ethanol from wheat/straw, ethanol from straw and biodiesel (FAME and FAEE) from rape seed are shown.

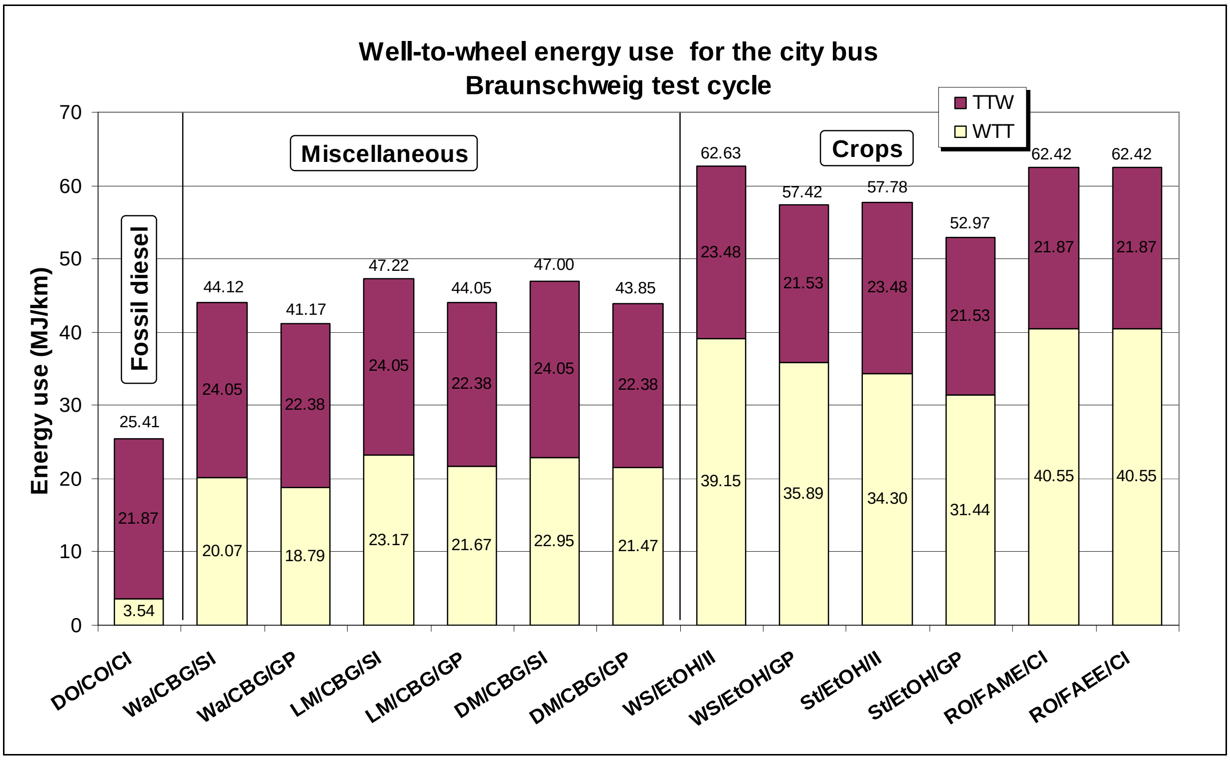

Figure 16. *Well-to-wheel energy use for the city bus in the Braunschweig test cycle (miscellaneous feedstock and crops from intensive farming)*

The first observation that can be made in **Figure 16** is that the total energy use for diesel oil is far lower than for all the biofuels. This is also valid for all other graphs below. The reason is the kind of "transformation" that nature already has made on crude oil and which facilitates refining to end products so much easier than for "virgin" biomass. On the other hand, much of the energy used for producing these biofuels and, in some cases, even the majority of the energy, is non-fossil, which enables a considerable reduction of greenhouse gases. Another remark is that both the two categories of feedstock for the biofuels are very different and thus, these results should not be directly compared to each other.

The results for the biogas fuels from various feedstocks are relatively similar besides the difference that is attributed to the fuel converted. Clearly the diesel cycle is more efficient than the otto cycle, which has been pointed out several times earlier.

By assessing the energy use for producing ethanol from wheat and straw, it can be concluded that the process that uses the cellulosic biomass only, i.e. the straw in the Iogen process, has somewhat lower energy use than the production of ethanol from wheat with mainly straw as the process energy. The previously discussed difference in efficiency between the engine with ignition improver and the glow-plug engine is apparent here as well.

Finally, the difference between FAME and FAEE is very small and can only be seen on the third digit after the decimal point (not shown in **Figure 16**). In contrast to the JRC study, non-fossil methanol has been foreseen here. Bio-methanol has higher total energy use than fossil methanol but the benefit is the reduction of greenhouse gases. In a case study, different production routes of ethanol and methanol could be compared with each other.

In **Figure 17**, results for the lorry running on fuels produced from wood or from wood indirectly via black liquor are shown. Only the results from the WHVC test cycle have been shown for the long-distance lorry. One reason for this selection is that this test cycle is considered to be the one best representing the driving pattern of a lorry among the test cycles investigated and the other reason is that the differences between these test cycles regarding fuel consumption has already been discussed.

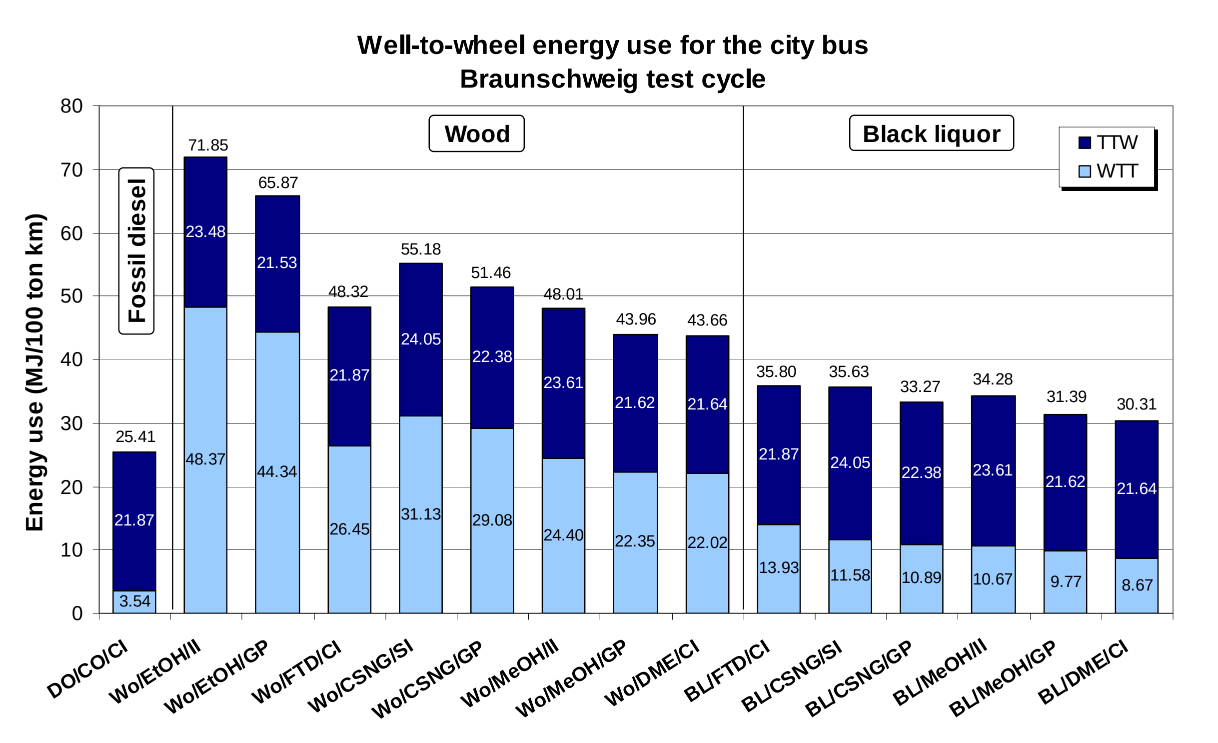

Figure 17. *Well-to-wheel energy use for the city bus in the Braunschweig test cycle (extensive cultivation)*

The results in **Figure 17** show the highest energy use for ethanol from wood among all the fuels produced from this feedstock. The relatively low efficiency of ethanol production from cellulosic biomass is seen in that the WTT bar is much higher than the TTW bar. As the discussion in the appendix shows, there might be some improvement here with the most advanced technology for ethanol production but the same selection of technology as in the JRC report is also made here. The previously noted difference between the two options for alcohol engines (ignition improver and glow-plug) can be seen also here.

The CSNG fuel has lower energy use than ethanol in general. This is in spite of the higher energy use for compressing the gaseous fuel for refuelling compared to ethanol. The relatively high energy use in ethanol production was discussed above. However, there is much uncertainty about the efficiency for SNG production as well so this picture might change. The relatively high efficiency of the lean-burn otto engine (in comparison to a stoichiometric engine) and the high efficiency of the glow-plug diesel engine are two conditions for the favourable results for CSNG.

The lowest energy use among the wood-based fuels can be seen for the glow-plug methanol engine and the DME engine with compression ignition. The difference between these options is very small, while the methanol engine with ignition improver has a higher energy use. There should also be some room for improvement here, since the JRC data for methanol and DME production was based on data from the early 1990’s.

Utilising the route of producing biofuels via black liquor instead of using the woody biomass directly significantly reduces the energy use. Due to the improved efficiency of bio-electricity production compared to the original data from Nykomb/Chemrec, the relative differences between the two feedstocks is even higher here than in other publications as, e.g. the JRC report. The greatest improvement in fuel production due to the improved electricity production can be seen for FTD produced from black liquor. When also the most favourable case for using the FTD bi-products is anticipated, as in this case, the energy use for FTD is not so much higher than for the other fuels. A case study has been made also for the other options for FTD production but it was chosen to show only the best possible option here. Using “worst case” assumptions might increase the fuel consumption by some 20 %. Drawbacks for FTD compared to the other fuels produced from this category of biomass is, besides the disadvantages in fuel production, are that aftertreatment is needed for both particle (DPF) and NO_X (SCR) emissions. As for the results for woody biomass, the energy use for methanol and DME as also the lowest among this category of feedstock.

4.5.2 Long-distance lorries

The results for the long-distance lorry are presented below. In contrast to the fuel/driveline combinations for the city bus, the only exception is that the compressed methane fuels (CBG and CSNG) are replaced with liquefied versions of the same fuels (LBG and LSNG). Therefore, a table corresponding to the table for the bus (**Table 11**) with explanations of the abbreviations is not shown here.

In **Figure 18**, the results for the lorry running on fuels produced via anaerobic digestion to biogas (waste, liquid and dry manure), ethanol from wheat/straw, ethanol from straw and biodiesel (FAME and FAEE) from rape seed are shown.

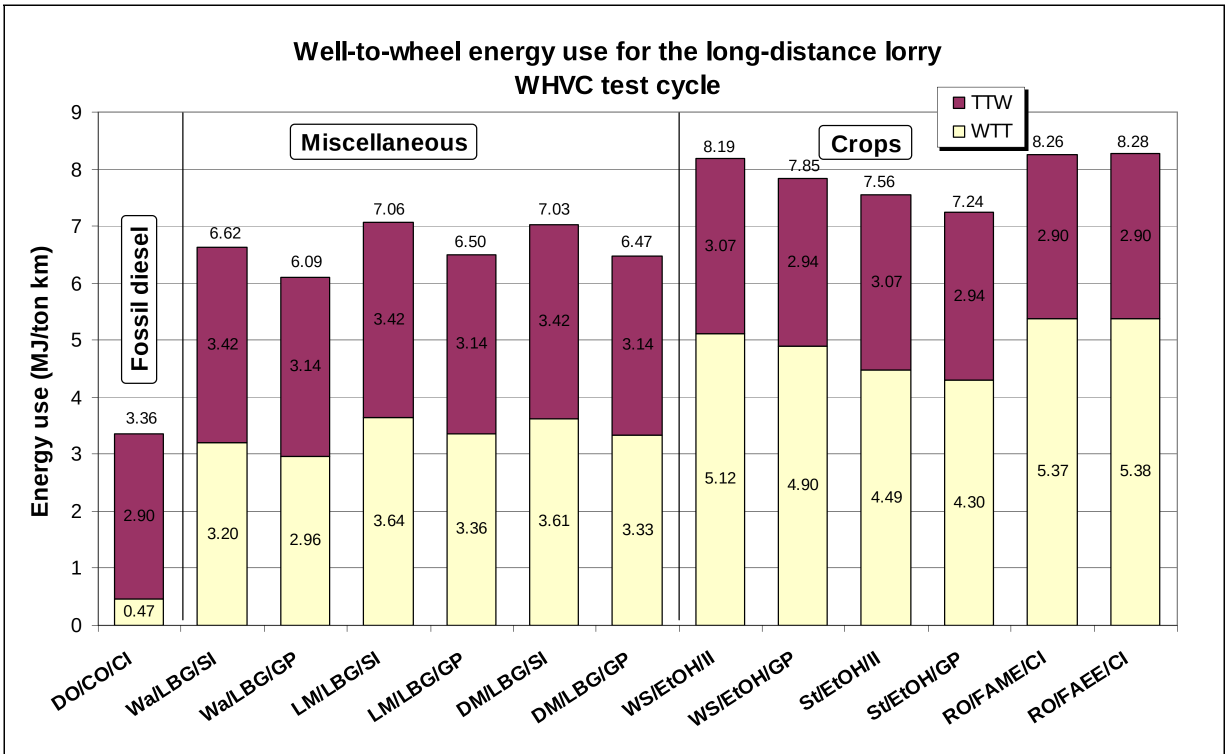

Figure 18. Well-to-wheel energy use for the long-distance lorry in the WHVC test cycle (miscellaneous feedstock and crops from intensive farming)

The results in **Figure 18** indicate quite a similar picture as the previously discussed results for the city bus (**Figure 16**) and need not to be repeated in detail here. Instead, a few remarks can be made where the results differ.

As previously noted in the discussion about fuel efficiencies, the advantage of a glow-plug ethanol engine is not as great for the lorry in the WHVC test cycle as for the bus in the Braunschweig test cycle. As for the bus, the difference between FAME and FAEE is minute also for the lorry.

In **Figure 19**, results for the lorry running on fuels produced from wood or from wood indirectly via black liquor are shown.

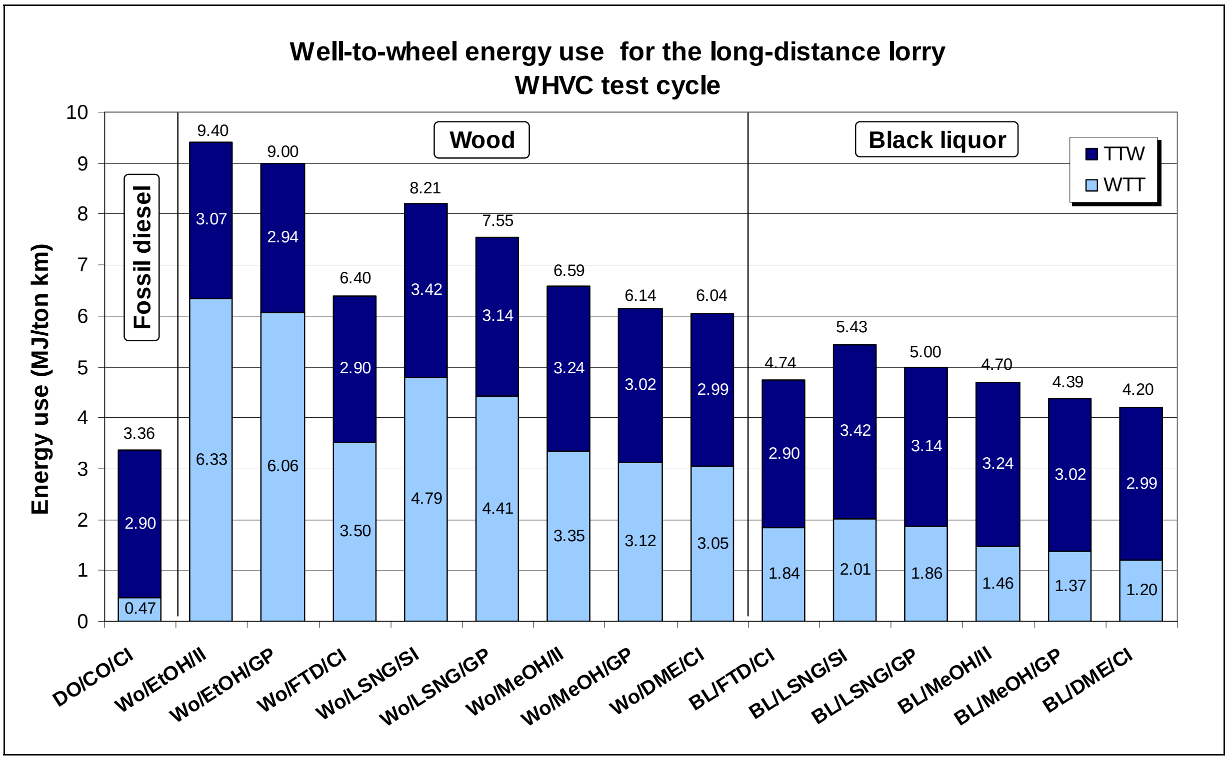

Figure 19. Well-to-wheel energy use for the long-distance lorry in the WHVC test cycle (extensive cultivation)

The results in **Figure 19** indicate quite a similar picture as the previously discussed results for the city bus (**Figure 17**) and need not to be repeated in detail here. Instead, a few remarks can be made where the results differ.

As noted before, the difference between the two alcohol concepts (ignition improver and glow-plug) are smaller for the lorry in the WHVC test cycle than for the bus in the Braunschweig test cycle.

The liquefaction of methane (LSNG) uses more energy than can be compensated for by much lower energy used for refuelling in comparison to compressed methane (CSNG). Additional differences can be noted in impact on vehicle weight due to the different storage methods and the greater range for the lorry. For practical reasons, compressed methane is not a viable solution for long-distance lorries. So, in general there is a slight disadvantage for biogas in long-distance lorries compared to city buses in relation to other fuels produced from woody biomass or black liquor.

The difference between methanol and DME are somewhat greater for the lorry than the bus. The drawback of heavier tanks for methanol is more pronounced for the lorry that has a greater range than the bus. Also, the payload is lower in the methanol case.

5 DISCUSSION AND CONCLUSIONS

5.1 Comparison with results from other WTW studies

Compared to the results in the study by JRC, differences have been noted in the calculation of fuel production. In the present study, it was decided to use as much bioenergy as possible in the production of the biofuels, so therefore these routes were chosen for comparison. Biomass was used for heat and power generation in the present study but not for cultivation and transport of feedstock, as well as for fuel distribution and dispensing (including electricity for this purpose). Although the balance for emissions of climate gases could be improved if this was the case, the contribution from transport and distribution is usually relatively small. The use of bio-energy often increases the total use of energy in the WTW analysis compared to the use of fossil resources (e.g. coal and natural gas) but improves the basis for comparison among various fuel options.

The WTW study by JRC used a passenger car as end use for the fuels and in the present study, heavy-duty vehicles were studied. Therefore, the basis for comparison was an otto engine in the JRC study while a diesel engine was the baseline in this study. The fact that it is very difficult to improve on the efficiency of a diesel engine can be seen in the results. On the other hand, it has been anticipated in this study that engines utilising the diesel cycle could be developed for all fuels studied. This includes gaseous fuels as methane. Instead of foreseeing for example, a dual fuel engine using methane and diesel fuel (for pilot ignition), a concept using a glow-plug is the most “extreme” solution, since no additional fuel or fuel additive (e.g. ignition improver) has to be used. It can be noted that the higher efficiency of glow-plug ignition in a diesel engine instead of a stoichiometric engine, as for passenger cars in the JRC study, improves the ranking of gaseous fuels in comparison to other fuels.

Compared to the latest WTW study by Ecotraffic [19] it can be noted that a higher efficiency for production of SNG has been anticipated here. In the present study, the same efficiency for SNG production as for methanol production has been foreseen. This would require a different kind of gasifier, in contrast to the older Ecotraffic study, where a similar gasifier was anticipated for all fuels. As mentioned above, the use of the diesel cycle also for the gaseous-fuelled engine improves the ranking of SNG and biogas.

5.2 Conclusions

The use of a common process for electricity generation has some impact on the results for fuel production derived from data in the JRC report. In some cases, such as for biogas production from waste and manure, the impact is very small due to a relatively little use of electricity. In some cases, such as, e.g. the fuels produced from biomass via black liquor, a significant improvement can be made. The greatest improvement can be seen for FTD from black liquor and this option is further improved by using the by-products for electricity generation in combined cycle power plants. Cases where this re-calculation decreases the efficiency compared to the JRC data are ethanol from wheat, straw and wood.

Most of the results for the city bus and the long-distance lorry are quite consistent with each other and, in cases where there are differences, they can be explained. The greater range required for the long-distance lorry and the fact that the most realistic comparison is the energy used per ton of goods transported often increase differences between options for the lorry. In cases where gaseous fuels are considered, the use of compressed gas for the bus and liquefied gas for the lorry has been foreseen and there are also differences related to these choices of technology. In general, the liquefied gases for the lorry are less favourable when compared to other fuels relative to the same comparison for the fuels used in the bus. This indicates that the use of gaseous fuels in city buses would be a preferred option in comparison to the used of cryogenic fuels in long-distance lorries.

Fuels that have a greater mass for the same energy content as other fuels usually have a drawback from this fact. Either the vehicle will be heavier (as the bus) or the payload will decrease (the lorry). Alcohols are heavier than diesel fuel and in the methanol case the relative difference is about a factor of two. Gaseous and cryogenic fuels, as well as DME, which is in liquid state under moderate pressures, are also plagued with heavier tanks than diesel fuel.

The impact of various driving cycles was investigated on both vehicle types. In general, the differences were quite small but the observed differences could mostly be explained. It is possible that the impact of more “extreme” driving cycles could have been greater than those studied here.

The most general conclusion that can be drawn about the energy converter is that if the most efficient engine type is used, i.e. the diesel engine, the differences between the TTW efficiency for most fuels becomes quite small. In order to apply such technology, a considerable development work would be necessary. It is plausible, that for some of the fuel options, as high efficiency as for the diesel-fuelled diesel engine might not be achieved due to some intrinsic fuel properties or practical reasons. On the other hand, it is also possible that some of the fuels could have properties that might be utilised for improving the efficiency to an even higher level than the diesel baseline. Examples here could be the “internal cooling” possible with direct injection of alcohol fuels and the specific combustion properties of DME. Likewise, the utilisation of EGR might also be optimised for these fuel options and possibly, also for other fuels.

6 ACKNOWLEDGEMENTS

Patrik Klintbom and his colleagues at Volvo Technology and Volvo Powertrain are acknowledged for his contributions regarding input data for fuel production and vehicle data, as well as for valuable comments regarding data evaluation and the preparation of the report.

Ralf Barkhage at Ecotraffic is acknowledged for conducting calculations on engine maps regarding the glow plug power and for valuable contribution to the report. Åke Brandberg at Ecotraffic is acknowledged for his contribution in assessing the potential for future production of lignocellulosic ethanol.

Tomas Ekbom at Nykomb Synergetics is acknowledged for his contribution in assessing the potential for alternative use of Fischer-Tropsch products.

This project was funded from the EMFO research programme administered by the Swedish Road Administration.

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Introduction

When searching for systems with capability to become long term sustainable for road transports the emphasis concerns low emissions of gases enhancing the greenhouse effect, particularly carbon dioxide of fossil origin. As sufficient improvement of energy converters in vehicles to meet the requirements does not seem possible, the acceptable system has to rely on motor fuels from renewable feedstocks. These are basically all conversions from the radiation of the sun to the earth.

The basic fact is that the radiation represents in energy terms a daily inflow about 10,000 times the total use of all kinds of energy carriers in the entire world today.

The direct conversion of the radiation, varying with time and geographical position, for moving vehicles meets obstacles by lack of suitable easily handled storable energy carriers. Electricity produced by photovoltaic cells can be used for electrolytic production of hydrogen from water Hydrogen may also possible to produce from water by bio-chemically aided radiative splitting. Hydrogen, however, is the lightest compound of the earth and therefore the most difficult to handle for distribution and storage. In combination with other atoms, carbon (and oxygen), hydrogen can be captured in chemically bound energy carriers such as easily handled liquid hydrocarbons or alcohols.

Indirect capture of radiation energy from the sun uses the photosynthesis, developed naturally since billions of years to form the basis for all life on the earth. Primary products are built by water and carbon dioxide via carbohydrates (sugars). These are simple sugars (monosaccharides) such as hexoses ($C_6H_{12}O_6$) or pentoses ($C_5H_{10}O_5$) and polymers of two (disaccharides) or more units (polysaccharides, starch, cellulose, hemicellulose). This is truly renewable fuel/pathway. It is estimated that the energy captured yearly in biomasses is about 10 times the total use of all kinds of energy carriers in the entire world today. The major part ends in woody plants as lignocellulose. With an energy resource of such a potential it can hardly be claimed that lack of energy will be a major issue even when fossil resources have been utilised to greater extent. The challenge is, however, to identify and develop which and how the renewable feedstocks can be utilised and be converted in much greater extent.

When carbon dioxide from the atmosphere participates in a cycle of synthesis and combustion with a cycle time of less than about 100 years it can be considered renewable. All fossilised fuels originate from biomasses formed long time ago, often under conditions not longer prevailing on the earth, and therefore not are renewed appreciably. Peat is the youngest fuel with cycle times of several thousands of years.

Feedstocks - Conversion processes

Biomass feedstocks can be converted to motor fuels by two routes, thermo-chemically via gasification to synthesis gas and synthesis to a variety of compounds or bio-chemically via hydrolysis and fermentation to most commonly ethanol. The former is dealt with elsewhere and only the biochemical is described here.

Feedstocks containing **simple sugars**, mono- and disaccharides, as in sugar cane and beets can be directly fermented to ethanol of the sugars recovered as water solution by pressing or leaching. The yeast strain Saccharomyces cerevisiae readily ferments hexoses, which are the main carbohydrates in these feedstocks, but not pentoses. The basic overall reaction is:

$$C_6H_{12}O_6 \;==\; 2\ C_2H_5OH + 2\ CO_2 + \text{heat}$$

51 %m is ethanol and 49 %m carbon dioxide as side product. Some unwanted by-products such as glycerol, acetic acid, lactic acid, fusel oil, furfural also are formed. Some formation of new cell mass will also occur and is required to maintain a healthy culture. The reaction proceeds with maximum efficiency at 35^{o}C and stops when the temperature exceeds 43^{o}C. As the reactions are slightly exothermal the temperature of the fermenting substrate will rise and slow down the reaction rate unless cooling is applied. The common yeast can convert sugar solutions up to 16-22 %m concentration resulting in beer with 8-11 %v ethanol. Yield figures of 80-100 litres per ton of fresh sugar beets have been claimed at conditions prevailing in Sweden. The sugar cane based industry in Brazil, producing about 16 million m^3 yearly, is the most well-known example for ethanol production from hexoses but there is a large potential in other tropical regions.

Another sugar source, although small and declining, is the spent liquor from the sulphite pulping process which about 3 % sugars, 65 % of which is directly fermentable to ethanol. The substrate may be concentrated before fermenting to increase the ethanol content to reduce the steam consumption at the following distillation. Yield figures may be as high as 50 litres of ethanol per ton of fresh wood chips charged to the pulping. Only one such plant remains in Sweden (MoDo at Örnsköldsvik) producing about 10,000 t/yr ethanol.

With **starch plants** as feedstocks the polymeric structure does not allow direct fermentation but must first processed by an enzymatic conversion step (saccharifaction) to fermentable sugars. Many different agricultural feedstocks, tubers such as cassava, Jerusalem artichokes and potatoes, and grains such as corn (maize), wheat, milo, rice, barley and rye are used. The first step is preparing a 15-20 % solution of monomeric sugars that can be fermented by yeast. Residual parts of the feedstocks form by-products to be used as animal fodder products. Corn is the most well-known feedstock for 12 million m^3/yr ethanol production in the U.S. In Sweden commercial the wheat based plant at Norrköping (about 50,000 t/yr ethanol) is the first of its kind with a second bigger one decided.

Lignocellulosic feedstocks, being the by far largest potential resource, is a category of many plants including small trees and residues from forestry and forest industry, municipal waste wood and paper, agricultural residues such as bagasse, straw, corn stover (stalk and leaves), purpose grown energy crops (farmed crops) such as willow, poplar, etc. The main constituents of this type of biomass are cellulose, hemicellulose and lignin, which present more difficult processing conditions for conversion to ethanol as they cannot so easily be transformed to fermentable sugars. Cellulose is long linear polymer of glucose (hexose) molecules in a crystal structure which provides rigidity in the plant. Hemicellulose is a heterogeneous and branched polymer (less degree of polymerisation than cellulose) made up predominantly pentoses as well as hexose sugars and forms a protective layer around the cellulose chain. Lignin is phenolic polymer that ensures strength and resistance against chemical and biological attacks in trees and plants. Remaining constituents are so called extractives (soluble in organic solvents) and inorganic ash.

Different lignocellulosic feedstocks vary in composition dependant on type, origin and growing conditions of the plants and range as percentage of the dry substance

- cellulose:	35-55 %
- hemicellulose:	5-30 %
- lignin:	15-30 %
- extractives and ash:	10 %

The ash content is usually only one or a few percent. Young plants (one year crops; straw, bagasse, corn stover, switchgrass) are usually richer in hemicellulose than woody short rotation plants (willow (Salix), poplar, aspen) hardwood trees and the ratio cellulose/hemicellulose is highest in older softwood trees (spruce, pine).

The basic problem in the use of lignocellulosic feedstocks for ethanol production is the difficulty in separation its constituents from each other with high yield and good economy for subsequent conversion to fermentable sugars. The first step is hydrolysis of the polymers (breaking the chains and incorporation of water) which can be illustrated in the overall reactions shown below for cellulose and hemicellulose respectively:

$$(C_6H_{10}O_5)_n + n^{\cdot}H_2O \;==\; n^{\cdot}\underset{\text{hexoses}}{(C_6H_{12}O_6)}$$

$$(C_5H_8O_4)_n + n^{\cdot}H_2O \;==\; n^{\cdot}\underset{\text{xyloses}}{C_5H_{10}O_5}$$

The polymers is very high in cellulose, thousand up to 15,000 i.e. a high degree of polymerisation, but lower, about a few hundred in hemicellulose. The hydrolysis means some mass gain to monomeric sugars, slightly higher for xyloses. In the subsequent fermentation both the hexose and pentose sugars show the same mass yield of ethanol, theoretically 51 %m. The total amount of convertible components obviously sets the limit the ethanol yield.

The hydrolysis reactions can be speeded up catalytically by acid or enzymes. The hemicellulose fraction is more readily hydrolysed than the cellulose and in the process layout this will be used to make the more difficult to hydrolyse cellulose polymer chains more available. Both must be used to achieve highest possible yield of ethanol. Wood feedstock was used before world war II in processes based on acid hydrolysis, but since the oil crises in the early 70-ies R&D began and has now resulted in mainly enzymatic hydrolysis of a broad range of lignocellulosic feedstocks.

The modern optimised processes, which have given the best results, use a two-stage approach to yield about 80 % conversion to ethanol of the theoretical potential. In the first step, pre-treatment of suitably disintegrated feedstock by steam at about 200^{o}C and lowered pH (by sulphur dioxide or sulphuric acid) is performed and is followed by simultaneous enzymatic saccharification and fermentation with micro-organisms (bacteria and genetically modified yeast strains) capable of converting both pentoses and hexoses and their oligomers to ethanol

(SSF). The process conditions and their severity are a balance between the wanted reaction to ethanol and degradation products, which may act as inhibitors for the ethanol formation. As the further up-grading to end product fuel ethanol involves distillation it is crucial for the economy the process to see the energy demands. The concentration of ethanol of the fermented beer will then be the most important factor.

Engineering studies

Engineering studies with complete mass and energy balances are required for calculation of economic feasibility of ethanol production from lignocellulosic feedstocks. The most comprehensive study has been accomplished by US DOE´s National Renewable Energy Laboratory (NREL) (ref. NREL 1999) and seems to have been used by most other studies later (incl. Eurocar/JRC/Concave -Well-to-Wheel May 2006).

NREL studies a big stand alone (grassroots) plant with renewable corn stover or switchgrass from large nearby farmed crops but uses property data of yellow poplar hardwood (which may represent other short rotation forestry crops such as willow) as shown in the table below.

Table A1.1 Feedstock Composition

Component	% Dry Basis
Cellulose (Glucan)	42.67
Xylan	19.05
Arabinan	0.79
Mannan	3.93
Galactan	0.24
Acetate (as groups in hemicellulose)	4.64
Lignin	27.68
Ash	1.00
Total	100.00
Moisture (as received)	47.90

The theoretical conversion yield with this feedstock is 483 litres of ethanol corresponding to 53% on energy basis based on an assumed energy content of 19.2 MJ/kg dry feedstock (not given in the report). Actual yield figures will include energy for power and steam needed to operate the plant.

In the table below the achieved and expected future process conditions used in the study and the results have been listed. The base case refers to the status end 1998, the industry best with near term optimisation and the others include improvements following R&D work and use of genetically improved feedstock (2015) with higher cellulose and hemicellulose content (80%).

Table A1.2 Conditions

CASE	Base	Industry best	2005	2010	2015
Pretreatment: xylan to xylose, %	75	85	85	85	85
SSF-Hydrolysis/fermentation time, days	7	7	3.5	2	2
SSF " " temp., ^oC	30	30	55	55	55
Conversion: cellulose to glucose, %	80	80	80	90	90
" : glucose to ethanol, %	92	92	92	95	95
" : xylose to ethanol, %	85	85	85	95	95
" : other C5 to ethanol, %	0	85-90	85-90	95	95
Losses, %	7	7	5	3	3
Operating time, hours per year	8406	8406	8406	8406	8478

Engineering calculations for a stand alone plant based on a feedstock input of 2,000 dry tones/day and the conditions in the table above resulted in production data shown below. Fuels for supplying steam and power to operate the plant are dewatered lignin product, evaporated liquid slop, and biogas (methane) from the anaerobic waste water digestion. Fuels appear in excess to the needs and are converted to power for sale. Enzyme production is accomplished internally within plant from part of the feedstock.

Table A1.3 Production base data

CASE	Base	Industry best	2005	2010	2015
Feedstock input, tonnes DS/year	700,500	700,500	700,500	700,500	706,500
Added feedstock as fuel, tonnes DS/year	--	--	--	--	97,140
Ethanol (absolute), m^3/year	197,575	222,555	235,425	273,275	331,185
Excess electricity, GWh/year	92	156	174	88	--
Specific data					
Ethanol, litres/tonne feedstock	282	318	336	390	412
Excess electricity, kWh/tonne feedstock	130	220	250	130	--
Ethanol, % of theoretical	58	66	70	81	81.5
Energy yields, % of feedstock LHV					
Ethanol (absolute), %	31.0	34.9	36.9	42.9	45.3
Electricity in excess, %	2.5	4.2	4.7	2.4	--
Total energy yield, %	**33.5**	**39.1**	**41.6**	**45.2**	**45.3**

The improvements involve use of more efficient micro-organisms, more efficient enzyme production, improved conversion yields, speedier conversion, better recovery/lower losses, and improved energy efficiency. The ethanol production thereby increases nearly 40 % to 390 litres/tonne dry feedstock. This is about 81% of the theoretical corresponding to nearly 43 % of the energy content of the feedstock (LHV-basis) together with a small excess of electricity, totalling the energy yield to about 45 %. Somewhat higher ethanol production will be achieved with feedstock of higher content of convertible cellulose/hemicellulose but the plant then needs additional fuel to make it energetically self-sustained and the total energy utilisation will not improve. A lignocellulosic feedstock with 67% convertibles is rather high considering an average of all types of feedstocks, which may be in the range 60-62 %. Such feedstocks may result in higher ratio of surplus fuel to ethanol.

The power production is in the NREL-study achieved with back pressure steam turbine/generator with condensing tail turbine. The ratio power to steam is in the base case only about 0.27. The study mentions the possibility of using an IGCC-system for improved efficiency but this is not pursued. As the size of the power central is moderate, the effect on total efficiency would not be very great in spite of the rather sizeable ethanol plant. In a much larger system of many ethanol plants, central surplus fuel handling for higher power production and higher efficiency, the IGCC might be an option. The EUCAR-study a power production efficiency of 0.33 has been used.

The lignocellulose to ethanol plant is reality a multi-product plant as excess of solid fuel (or electricity) is unavoidable. This raises the question of how to allocate the energy used to operate the plant. Simple crediting of the by-product energy in accounting of the ethanol energy balance (as in the EUCAR-study) is hardly correct as the most energy demanding sub-processes are required for ethanol.

A Swedish study (SNRA 2001:85) allowing somewhat higher ethanol yield but the poorer feedstock indicates 41 % yield of ethanol (LHV-basis) or 370 litres/tonne feedstock DS and 49 % including the excess lignin fuel. Reasonable allocating is estimated 1.15 MJ/MJ ethanol and 0.45 MJ/MJ lignin fuel.

LCA-considerations - WTT

In the entire chain for production and use of a motor fuel several steps are involved. For ethanol from lignocellulosic feedstocks the full energy chain constitute the following steps

- Primary feedstock exploitation/recovery
- Feedstock transport to conversion plant
- Conversion to motor fuel
- Motor fuel distribution
- Conversion in vehicle engine
- Conversion to transport work.

The four first steps constitute the "well-to-tank part" (WTT) part of the full chain are summarised here. It is common to express energy demanded in MJ/ $MJ_{ethanol}$ product.

In the first step energy inputs are at silviculture and collection of forestry residues or at cultivation and harvesting of farmed (short rotation forestry) crops in form of fertilisers and motor fuel (diesel oil) for machinery used. Step two involves feedstock preparation and efficient haulage considers a 150-300 km road roundtrip. Conversion to ethanol is covered in the previous section above. The final step includes distribution to the dispensing stations in a manner similar to the present handling of petrol. In the table below data for waste forestry residues from the EUCAR/JRC/Concawe WTT-report, App.1 and 2., May 2006, (based on the NREL best industry case) is shown as well as data from two Swedish studies (SNRA 2001:85, futuristic case and LoF, early case). Comparisons between different reports are difficult as the presumptions made are not entirely the same and case configurations differ. Obviously a great number configurations and scenarios can be designed.

Table A1.4 Energy Use in Various Steps from Lignocellulose Biomass, MJ/MJ$_{ethanol}$ and efficiency

Step Source	Feedstock recovery	Feedstock transport	Conversion	Distribution	TOTAL WTT		**Efficieny** dim.less
EUCAR	0.08(f)	0.04(f)	1.80(b)	0.03(f)	1.80(b)	0,15(f)	**0.34**
SNRA	0.06(f)	0.03(f)	1.15(b)	0.012(f)	1.15(b)	0.10(f)	**0.444**
LoF	0.054(f)	0.015(f)	1.38(b)	0.01(f)	1.38(b)	0.08(f)	**0.445**

(f) refers to fossil energy, (b) to bio-energy

The conversion to ethanol step dominates in the full chain and the small differences in data from the other steps depending on less defined configurations are not very important. Farmed wood feedstock is usually presumed to demand somewhat more energy than in silviculture. Fossil resources such as motor fuels could, of course, be replaced by bio-fuels in the future.

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Potential use of naphtha by-product

1.1 Potential of refining naphtha to petrol

1.1.1 Naphtha share of the FT-products

The share of naphtha (FTN) from a GTL plant that would be optimized for diesel production (FTD) has been calculated in the BLGMF II study as 34% with 67% as diesel (on an energy basis). The diesel fraction yielded a boiling point curve with T95% as 285 ^{o}C. Thus, the naphtha share would be taken as approximately C_{5+} of hydrocarbons up to 180 ^{o}C.

Should a higher T95% be taken, e.g. 320 ^{o}C, then the split would be about 70/30 (on an energy basis). However, the naphtha fraction would remain. According to Statoil the split could be as high as 80/20 (on an energy basis) for a dedicated diesel GTL plant [1].

1.1.2 Isomerization process

The naphtha is highly paraffinic and because of very low octane number would not be used for isomerization to petrol (FTP) because the refineries values better economic use as steam cracker feed and to produce olefins [1], [2], [4]. In fact, the yield is 10% better than for conventional naphtha.

However, should the naphtha be used for upgrading to petrol a standard catalytic isomerization process would be used, e.g. UOP Penex process (see Figure 1), which has been commercial since 1958 [3]. The Penex process for light naphtha isomerization is currently operating in more than 120 units worldwide (and at three Swedish refineries).

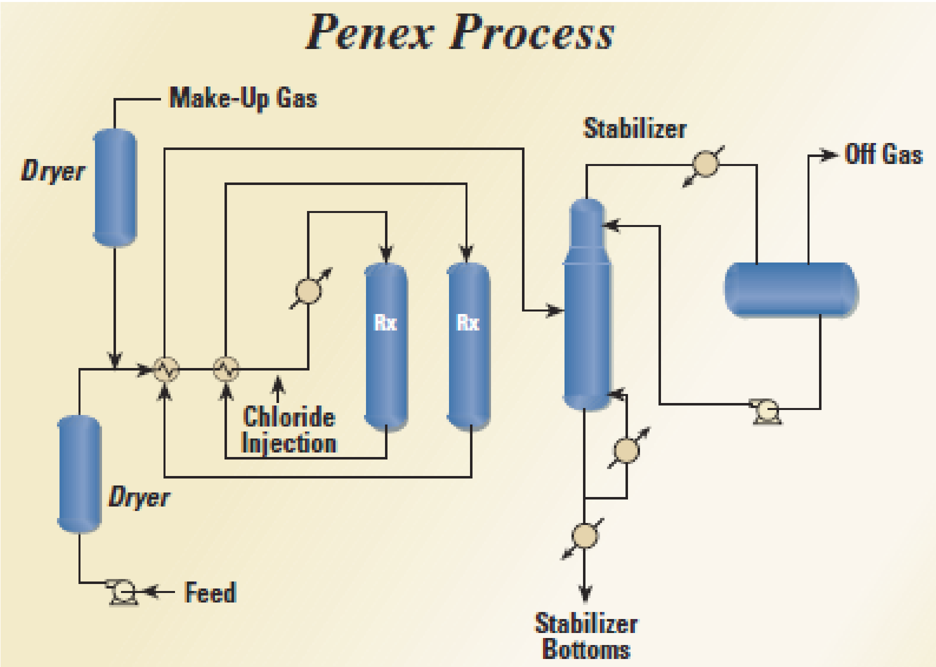

Figure 1. *Simplified Penex Process Flow Sheet, UOP © 2006.*

For best quality of the petrol product only the C_5/C_6 components should be used [4]. However, UOP states that the costly pre-fractionation column can be avoided and even the C_6 cyclics and C_{7+} could be used as feed.

However, the risk when using higher hydrocarbons is coking [1]. Therefore, the C_5/C_6 would be recommended as feed. Alternatively, high carbons such as C_7 would be cracked with about 25% yield along with isomerisation [4].

The isomerate yield is said to be 99+ vol%, thus, the selectivity is very high. If n-paraffins are not isomerised the process can be combined with molecular sieves that capture the n-paraffins and recycle them. The yield of FT-naphtha would possibly be 95+ vol% when considering C_5/C_6.

For typical feeds the product RONC would range 83 to 86 on single pass and with recycling as high as 93 RONC [3]. However, it can not be stated with certainty the yield of naphtha to petrol as a component analysis must be made because the range of octane numbers for the hydrocarbons is substantial, e.g. C_6 ranges 30-105.

The isomerisation product octane number depends on the feed octane number. For example, the naphtha at Preem refinery is isomerised to octane number 82 for C_5 and 77 for C_6 and 60 for C_7. The combined end-product has octane number of 86-87 which is then mixed with hydrocarbons of very high octane number, e.g. alkylates and MTBE (however costly and sparse on the market).

1.1.3 Yield of naphtha as petrol components

For naphtha, all hydrocarbons from C_5 up to boiling point 160 ^oC could be upgraded [4]. The hydrocarbons that would be isomerised but not meet required octane number would have to be converted to aromatics, where there is a petrol maximum limit of 35% vol. There is also a question on keeping the Reid Vapour Pressure (RVP) as the isomerised naphtha would raise the RVP and heavier components would have to be blended in to maintain the petrol specification.

Essentially all naphtha carbons could be upgraded, but at higher cost and to some extent energy intensive with steam used for heating the processes and fractionation (the isomerisation process in itself requires little utilities). Some uncertainty remains in the realistic possibilities of obtaining required octane numbers and problems with increased RVP etc.

Taking the above numbers and assuming the naphtha is used as is with components below 160 ^oC, then about 66 MW of naphtha would be used as feed. With an assumed yield of about 95% and remaining low octane number hydrocarbons aromatised the FTP product would be roughly 63 MW[1] or 17% of the input biomass (with no regard taken of heat input for the Penex process).

Alternatively, taking only the C_5/C_6 components the yield would be possibly higher, 99% however a pre-fracticonation column must be included. The share would tentatively be 50% of the total naphtha and consequently, the yield to FTP would then be roughly 41 MW or 11% of the input biomass (with no regard taken for additional heat consumption in destillation column or Penex process).

[1] In this case, the additional heat consumption would be relatively and comparatively small.

In conclusion, the naphtha yield as petrol components FTP would range 41-63 MW in the BLGMF II calculated case or 11-17% of input biomass energy (378 MW).

The final energy balance without regard for additional heat consumption would be from 100% biomass energy, 43% (or 45% with 320 ^{o}C) FTD (diesel) and 17% as highest FTP (petrol) with remaining 40% losses.

1.2 Potential and efficiency of BIGCC

1.2.1 Biomass IGCC process description

On behalf of Nykomb Synergetics Carbona prepared a conceptual economic analysis of the biomass IGCC considering Carbona's technology and USA conditions [5]. The analysis considered both condensing and cogen application of IGCC. The cogen application concentrated on application in the pulp and paper industry. The plant size was between 15 and 100 MW electrical capacity. The results of the analysis are documented in this report.

The biomass based IGCC concept of Carbona employs air blown gasification and hot gas clean-up (see Figure 2). The fuel feeding system includes feeding lines based on lock-hoppers which pressurize the fuel up to gasifier pressure. The biomass fuel is fed by screw feeder into the gasifier reactor. The gasifier system is a pressurized fluidized bed gasifier. The gasification medium is air and a small amount of steam depending on feedstock moisture content.

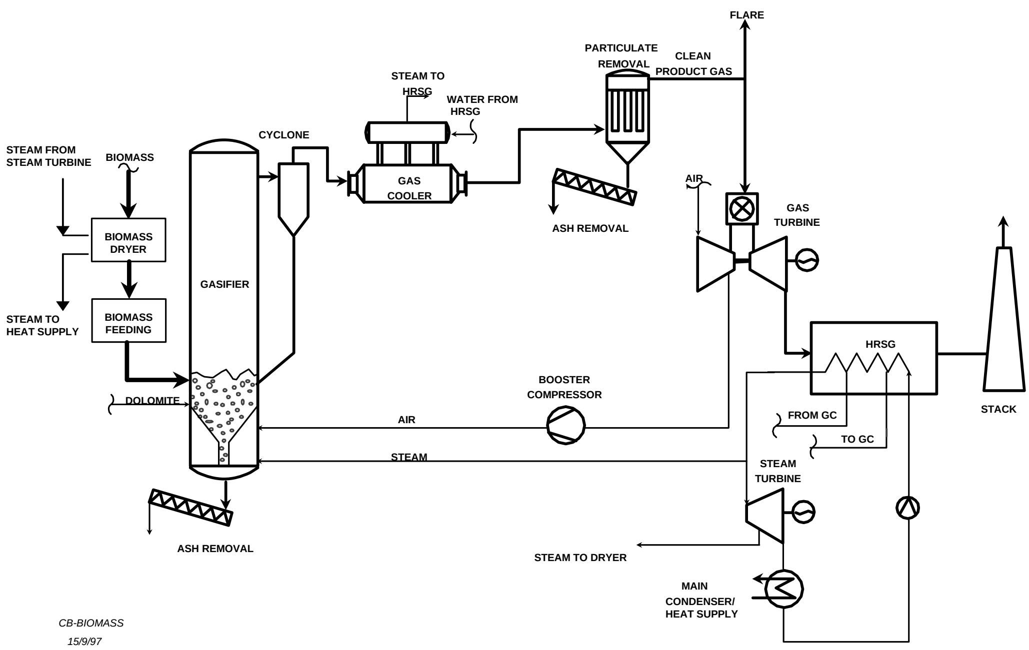

Figure 2. *Simplified IGCC Process Flow Sheet, Carbona © 2006.*

The gasification air is extracted from the gas turbine and after passing through the booster compressor/heat exchanger system it is fed into the gasifier. The product gas is a typical low-calorific gas of 4 to 6 MJ/m^3 LHV (100-150 Btu/scf). It contains combustible gas components like carbon monoxide (9-18%v), hydrogen (9-15%v) and methane (4-8%v) and inert gas components (nitrogen, carbon dioxide and water vapour). The product gas impurities are ammonia and a small amount of sulphur gases, light tars and vapour-phase alkalines as well as dust particles.

The product gas treatment of the simplified IGCC-process includes two steps: gas cooling and filtering. The product gas exiting the gasifier cyclone is cooled from freeboard temperature to 450-550 °C (840-1020 °F) in a fire tube type gas cooler. The gas cooler generates saturated steam. The dust content of the cooled product gas is removed in a filter unit, which operates at gasifier pressure and at 450-550 °C (840-1020 °F) temperature. The clean product gas is directed to the control valve of the gas turbine.

The Gas Turbine Plant includes the gas turbine package of usual components. Heavy duty, single shaft industrial gas turbines of high efficiency, specific power and exhaust temperature are preferred for IGCC-application. The gas turbine is commercial equipment which is modified for low calorific value gas combustion and IGCC-application.

The product gas of 450-550 °C (840-1020 °F) temperature is fed to the gas turbine through the control valve system. The combustion chamber of the gas turbine is modified for low calorific value gas combustion (4-6 MJ/m3 i.e. 100-150 Btu/scf) and low ammonia to NOx conversion. About 10% of the total air flow is extracted after the last stage of the compressor and used as gasification agent. The extracted air is returned to the combustion chamber as a part of product gas.

Steam Cycle of the IGCC-plant is similar to the steam cycle of conventional combined cycle plants including heat recovery steam generator (HRSG) steam turbine and condensate and feed water system. Both in condensing and cogeneration applications the integration of gasification plant (drying, gas cooler, etc.) into the steam cycle provides numerous possibilities to establish different, efficient process configurations.

1.2.2 Biomass IGCC efficiency

The gas turbine is the determining component in the IGCC process. The dimensioning and performance of the IGCC-process depends on the gas turbine technical parameters. The heat demand of the gas turbine determines the size and capacity of the Gasification Plant, while the exhaust parameters of the gas turbine determine the size and performance of the steam cycle.

When selecting a suitable gas turbine type for an IGCC application the following technical and strategic aspects have to be considered:

- the selected gas turbine type has to be a high efficiency, high specific power design,
- single shaft, heavy duty gas turbines are preferable (not sensitive of varying fuel flow and pressure),
- the selected gas turbine type has to be suitable for IGCC combined cycle application and site circumstances.

Considering the above listed requirements and the IGCC plant size requirements from 10 to 100 MW the following gas turbines were selected:

Nuovo Pignone PGT10 is a 10 MW gas turbine with single shaft arrangement and silo type combustor system. It has a redesigned upgraded version (PGT10B).

General Electric PG5371P gas turbine is a 26 MW machine with single shaft arrangement and can type burners. Being an older gas turbine type it has lower efficiency compared to the other machines considered in this study. It was selected since in this specific size range there are not so many machines available for low calorific gas application and General Electric approved that F5P would be available for this purpose if necessary.

General Electric PG 6561B gas turbine is a 39 MW machine with single shaft arrangement and can type burners. It is a well proven and widely used gas turbine. Carbona worked with GE on this machine (bids available) and GE tested Carbona's gas in the combustion system of the F6B.

Westinghouse W251B12 gas turbine is a 50 MW machine with single shaft arrangement and can type burners. It is a well proven gas turbine. Carbona worked with Westinghouse on this machine (bids available) and Westinghouse tested Carbona's gas in the combustion system designed for the W251B for low calorific value applications.

General Electric PG 6101FA gas turbine is a 70 MW machine with single shaft arrangement and can type burners. It is a new development of high efficiency. It is used in IGCC applications (Pinon-Pine project/USA, process is similar to Carbona's process) with air blown gas. GE approved that the F6FA gas turbine can be used with Carbona's gas.

The power plant calculations were executed by using Carbona's process calculation code. The process performances are summarized in the following Table:

Table A2.1. *Results on biomass IGCC efficiency.*

Gas turbine type[a]	**F6FA**	**W251B11**	**F6B**	**F5P**	**PGT10**
Condensing IGCC					
Fuel consumption, tph	103.7	70.7	60.4	44.9	18.4
IGCC net-power, MW	115.9	72.0	62.1	42.0	18.1
IGCC net heat, MMBtu/h	0	0	0	0	0
Heat Rate (HHV), Btu/kWh	9359	10 276	10 165	11 180	10 632
Cogen IGCC					
Fuel consumption, tph	103.7	70.7	60.4	44.9	18.4
IGCC net-power, MW	97.0	61.3	52.2	35.8	15.4
IGCC net heat, MMBtu/h	326.3	207.1	175.6	118.3	52.2
Heat Rate (HHV), Btu/kWh	7544	8340	8393	9414	8760

Notes:

[a] The gas turbine types have been in some cases enhanced but to minor extent and some remained during the decades thanks to reputation as "work horses".

The heat rate is in both condensing and cogen cases the heat rate of power generation on HHV basis. The heat rate of the F5P gas turbine based process is higher since it is a lower efficiency gas turbine. The heat rate of power generation of the cogen IGCC was calculated considering a surrogate heating boiler of 85% efficiency.

There is also possibility of cofiring in a natural gas fired combined cycle plants that typically have efficiency ranging 44-55%. In the case of fully biomass fuelled IGCC up to 200 MWth capacity this efficiency are between 42-50% [6].

In conclusion, the biomass to electricity efficiency ranges 43-49% with assumptions: 8000 h/a, wood waste of 50% moisture AR, condensing power. The most reliable and typical gas turbines with highest efficiencies are the General Electric, LM2500 with and the General Electric, F6FA with 49%, respectively.

1.3 Naphtha efficiency in combined cycle

Today, the development of advanced gas turbine cycles have been to some extent reversed as the prices of natural gas on a global scale have risen significantly and the new efficient turbines have been found to be rather sensitive to disturbances, e.g. subject to greater fouling sensitivity [7].

Naphtha is used as fuel all over the world and the difference in efficiency versus natural gas which is dominating fuel for stationary gas turbines can be accounted for the atomization energy and the pump energy to reach inlet pressure. Altogether, the difference is almost negligible.

The size determines the efficiency and for a biomass to liquid plant, it would be more efficient to transport the fuel to a large power plant in order to accommodate for the large efficiency that can be obtained. The transport cost is small in comparison. However, should the plant be located close to the BTL plant, a smaller gas turbine can only be used.

Smaller gas turbines are seldom packaged as combined cycle plants, however, and range typically 20-60 MW, e.g. the General Electric LM2500 with 35 MW and net plant efficiency of 50.4%, thus yielding a fuel consumption capacity of 69 MW (suiting the BLGMF II plant naphtha production).

Middle-size turbines have an electric efficiency (net plant) of about 50-54%, e.g. General Electric LM6000 (net 106 MW) and only the very large turbines, e.g. General Electric 9H (net 480 MW) can reach 60% electric efficiency [8].

In conclusion, naphtha can be used without modifications (with option of blended with conventional naphtha) in a condensing mode combined cycle with typically a net electric efficiency of 54% or for very large turbines as much as 60%.

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