Engine Exhaust Gas Emissions from Non-road Mobile Machinery

Effects of transient load conditions

Magnus Lindgren
Department of Biometry and Engineering
Uppsala

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Abstract

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Non-road mobile machinery is used for a range of different operations with varying engine load characteristics. Fuel consumption and emission amounts from such machinery are dependent on the operation performed. The engine load during many of the operations performed is largely transient in nature. However, the majority of fuel consumption and emission data reported in the literature are based on static measurements and engine load patterns not representative of the real use of the vehicle.

Emission data derived from the average annual use of agricultural tractors, based on three different typical areas of use, were compared with emission data obtained from European and international standards. The results showed that the emission amounts obtained from the engine load cycles stipulated in European and international emissions regulations were not representative of the real use of non-road mobile machinery.

Furthermore, operations can be performed using a variety of different approaches depending on factors such as vehicle characteristics and driver behaviour. An engine load simulation model was developed and the results from the simulations showed that by using different driving strategies or transmission characteristics, it was possible to affect the emissions for a specific operation without affecting the fuel consumption or duration of the operation.

Changes in engine speed and torque, *i.e.* transients, are likely to affect the fuel consumption and formation of emissions due to periods of time lags in *e.g.* the fuel injection system or the boost pressure. During normal operations a decrease in the fuel efficiency due to transient loads was detected. The fuel consumption increased with up to 13% compared with the corresponding steady-state condition. In order to study the effects of transients, special transient test cycles were developed. At high transients, both in engine speed and torque, the effects on fuel consumption and emissions were substantial. For example emissions of particulate matter increased with up to twelve times. Based on these results, a mathematical model was developed that could derive transient fuel consumption and emission amounts for all types of operations using a limited amount of emission measurements.

Keywords: engine, emission, exhaust gas, non-road mobile, agricultural tractors, construction equipment, transient, steady-state, diesel, fuel efficiency

Author's address: Magnus Lindgren, Department of Biometry and Engineering, SLU, P.O. Box 7032, SE-750 07 UPPSALA, Sweden. E-mail: Magnus.Lindgren@bt.slu.se

Sammanfattning

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Arbetsmaskiner används för en rad olika arbetsoperationer med varierande belastning på motorn. Bränsleförbrukning och avgasmängder eller emissioner är beroende av den arbetsoperation som utförs. Motorbelastningen är dessutom ofta av en transient natur. Ändock är majoriteten av den i litteraturen redovisade bränsleförbruknings- och emissionsdata baserad på statiska mätningar. Dessutom är, i de flesta fallen, de använda motorbelastningarna inte representativa för det verkliga arbetsområdet för motorerna.

Emissionsdata beräknad från tre olika belastningsfall jämfördes med data beräknad från belastningar enligt Europeiska och internationella standarder. De tre belastningsfallen var baserade på olika typiska årliga användningsområden för lantbrukstraktorer. Resultaten visade att emissionsmängderna erhållna via Europeiska och internationella standarder inte var representativa för de verkliga användningsområdena för arbetsmaskiner.

Dessutom kan en specifik arbetsoperation utföras på flera olika sätt beroende på t.ex. olika fordonsegenskaper eller körsätt. En simuleringsmodell över motorbelastning vid olika fältarbeten utvecklades och resultaten från simuleringar visade att det är möjligt att påverka emissionsnivåerna för ett specifikt arbete utan att påverka varken tid eller bränsleförbrukning.

Förändringar i motorns belastning dvs. varvtal och moment s.k. transienter påverkar med största sannolikhet bränsleförbrukning och emissionsbildning pga. olika fördröjningar i till exempel bränsleinsprutning eller turbotryck. Vid normala arbetsoperationer uppmättes en minskad bränsleeffektivitet på grund av transienta belastningar. Bränsleförbrukningen ökade med upp till 13% jämfört med motsvarande statiska förhållande. Speciella transienta testcykler utvecklades för att kunna studera effekter av transienter på bränsleförbrukning och emissioner. Vid kraftiga transienter i både varvtal och moment var effekterna på både bränsleförbrukning och emissioner påtagliga Bland annat ökade emissionerna av partiklar med upp till 12 gånger. Baserat på dessa resultat utvecklades en matematisk modell vilken, med ett begränsat antal mätningar, kunde beräkna bränsleförbrukning och emissionsmängder under transienta belastningar för alla typer av arbeten.

Keywords: motor, emission, avgaser, mobila maskiner, lantbrukstraktor, arbetsmaskiner, transient, statisk, verklig motorbelastning, diesel, bränsleeffektivitet

Adress: Magnus Lindgren, Institutionen för biometri och teknik, SLU, Box 7032, 750 07 UPPSALA, Sverige. E-mail: Magnus.Lindgren@bt.slu.se

“The improvement in city conditions by the general adoption of the motorcar can hardly be overestimated. Streets clean, dustless and odourless, with light rubber tired vehicles moving swiftly and noiselessly over the smooth expanse, would eliminate a greater part of the nervousness, distraction, and strain of modern metropolitan life.”

Conyngton, Thomas. (1899). Motor Carriages and Street Paving.
Scientific American Supplement 48

Content

Appendix

Paper I-VI

The present thesis is based on the following papers, referred to in the text by their Roman numerals. Published papers are appended and reproduced with kind permission of the publishers.

I. Hansson, P.-A. Lindgren, M. Norén, O. (2001). A comparison between different methods of calculating average engine emissions for agricultural tractors. Journal of Agricultural Engineering Research 80 (1), 37-43 doi:10.1006/jaer.2001.0710

II. Lindgren, M. Hansson, P.-A. (2002). Effects of engine control strategies and transmission characteristics on the exhaust gas emissions from an agricultural tractor. Biosystems Engineering 83 (1), 55 –65, doi:10.1006/bioe.2002.0099

III. Hansson, P.-A. Lindgren, M. Nordin, M. Pettersson, O. (2003). A methodology for measuring the effects of transient loads on the fuel efficiency of agricultural tractors. Applied Engineering in Agriculture, 19(3), 251-257

IV. Lindgren, M. Hansson, P.-A. (2004). Effects of transient conditions on exhaust emissions from two non-road diesel engines. Biosystems Engineering, 87(1), 57-66, doi:10.1016/j.biosystemseng.2003.10.001

V. Lindgren, M. Hansson, P.-A. Effects of transient conditions on particulate matter emissions from a non-road diesel engine. (Submitted to Transactions of the ASAE)

VI. Lindgren, M. A transient fuel consumption model for non-road diesel engines. (Submitted to Biosystems Engineering)

Note on the authorship of the papers

In Paper I, the planning of the measurements was performed by Hansson and Norén, the interpretation and analysis of different methods of calculating average engine emissions were performed by Hansson and Lindgren. Hansson performed the writing of the paper.

In Paper II, the planning of the model was performed by Lindgren and Hansson. Lindgren performed the construction of the model, the interpretation and analysis of the results and the writing, with revision by Hansson.

In Paper III, the planning of the investigation was performed by Hansson, Nordin and Lindgren, the construction of measurement equipment was performed by Nordin and Pettersson, and the recording of measurements was performed by Pettersson and Lindgren. The interpretation and analysis of the results and the writing were performed by Hansson and Lindgren.

In Paper IV, the planning of the investigation was performed by Hansson and Lindgren, and the measurements were performed by Lindgren. The interpretation and analysis of the results and the writing were performed by Lindgren, with revision by Hansson.

In Paper V, the planning of the investigation was performed by Hansson and Lindgren, and the measurements were performed by Lindgren. The interpretation and analysis of the results and the writing were performed by Lindgren, with revision by Hansson.

In Paper VI, the planning of the investigation was performed by Hansson and Lindgren. Lindgren carried out the construction of the model, the interpretation and analysis of the results and the writing, with revision by Hansson.

Introduction

The most common source of propulsion in non-road mobile machinery today, and in the foreseeable future, is the diesel engine. Non-road mobile machinery consists of several different types of machinery, such as agricultural and forestry tractors and construction equipment *e.g.* wheel loaders, excavators, articulated haulers and road maintenance equipment, which are used for a range of different operations with varying engine load characteristics. Furthermore, the engine load characteristics during many of the operations carried out using non-road mobile machinery, such as different loading and transport operations, are largely transient in nature.

The exhaust gas is composed of the three major components water vapour, carbon dioxide and nitrogen oxide. However, the exhaust gas also contains several combustion by-products such as particulate matter, carbon monoxide and various hydrocarbons that can be hazardous to the environment and toxic to humans (Heywood, 1988; Bosch, 1996). Some of the most prominent pollutants are particulate matter and nitrogen oxide.

Growing public concern with exhaust gas emissions arising from diesel engines has led to increasingly tighter regulations worldwide. In the European Union, the USA and Japan, there is an obligation for manufacturers to certify new engine models to emissions performance standards. The current emissions regulations within the European Union, the USA and Japan stipulate the use of a test cycle based on different static engine loads *i.e.* at constant engine speed and torque, identical to the C1 cycle in the ISO 8178-4 standard (ISO, 1996; EU, 2004a; USEPA, 2004).

Therefore, most engines and engine equipment are optimised for steady-state conditions. However, impending regulations specify the use of a new test cycle that includes transient engine loads. Transient conditions are likely to affect both fuel consumption and emission formation compared to steady-state conditions.

Fuel consumption and emission amounts are dependent on the actual operation performed including transient load conditions. Measurements of fuel consumption and emissions are normally conducted under static conditions. Furthermore, the engine load patterns used in most measurements are not representative of the real use of the vehicle. In order to obtain accurate fuel consumption and emission data, operation-specific engine load patterns should be utilised and measurements conducted under transient conditions.

During transient conditions, engine equipment such as the air induction system, *i.e.* turbocharger and the fuel injection system, suffers from time lags thus affecting the air-fuel ratio and the mixing conditions within the cylinder (Rakopoulos & Giakoumis, 1999; Benajes, Luján & Serrano, 2000; Bane, 2002). For example, the fuel supply system responds rather quickly to a change in operating conditions, whereas the turbocharger may need several seconds to reach a fully spooled state.

Good knowledge of the fuel consumption and emissions amounts under different engine load conditions and especially under transient conditions is of vital importance to both decision-makers and manufacturers of engines and engine components. Members of the European Union must report annual national anthropogenic emission inventories to, for example, the United Nations Framework on Climate Change. Furthermore, high quality data on fuel consumption and emission data are also important in life cycle assessment methodology. Errors in emission data due to limited amounts of data being available and/or transient effects influence the statistics and can lead to incorrect conclusions.

Objectives

The main objective of this thesis was to study the effects of different engine load characteristics on the fuel consumption and emission amounts from non-road mobile machinery.

Sub-objectives were to study the influence of driver behaviour and the system of transmission of power from engine to drive wheel on the fuel consumption and engine exhaust gas emissions (Paper II) and to analyse the effects of transients in engine speed and torque on the fuel consumption and emissions of carbon monoxide, hydrocarbons, nitrogen oxides and particulate matter from non-road diesel engines (Papers III to VI).

A further sub-objective was to analyse the effects of different engine load characteristics on the fuel consumption and engine exhaust gas emissions from non-road diesel engines with respect to the average annual use of the vehicle (Paper I).

Relationship between the papers

This thesis focuses on the fuel consumption and engine emissions from non-road diesel engines. Paper I deals with the average annual use of agricultural tractors and the resulting amounts of engine exhaust gas emissions. Emission data based on three different methodologies for estimating the annual use of the vehicle were compared with emission amounts obtained according to European and international emission standards. Paper I can be regarded as an introduction to the area of research and it is rather independent compared with the rest of the papers included in the thesis.

In Paper II, a simulation model of an agricultural tractor including some implements was presented. The simulation model was used for estimations of the effects of different control strategies and transmission characteristics on fuel consumption and emissions. A semi-static interpolation method was used to derive fuel consumption and emission data. The model was developed at a rather early stage in order to gain understanding of the processes affecting the fuel consumption and emission amounts during different operations and engine loads with non-road mobile machinery. Only a limited amount of data was available for non-road mobile machinery and no literature on the influence of transient engine loads on fuel consumption and emission amounts suitable for use in the present study was found. Such data are, however, now available in for example Papers III to VI.

In Paper III, the effects of transients are introduced for the first time and then further studied in Papers IV and V. Paper III deals with a methodology for estimating the effects of transient conditions on the fuel efficiency for whole operations, whereas Papers IV and V deal with the effects of discrete or isolated transients on fuel consumption and emissions. Fuel consumption and emissions of carbon monoxide, hydrocarbon and nitrogen oxide are included in Paper IV, while Paper V focuses on emissions of particulate matter.

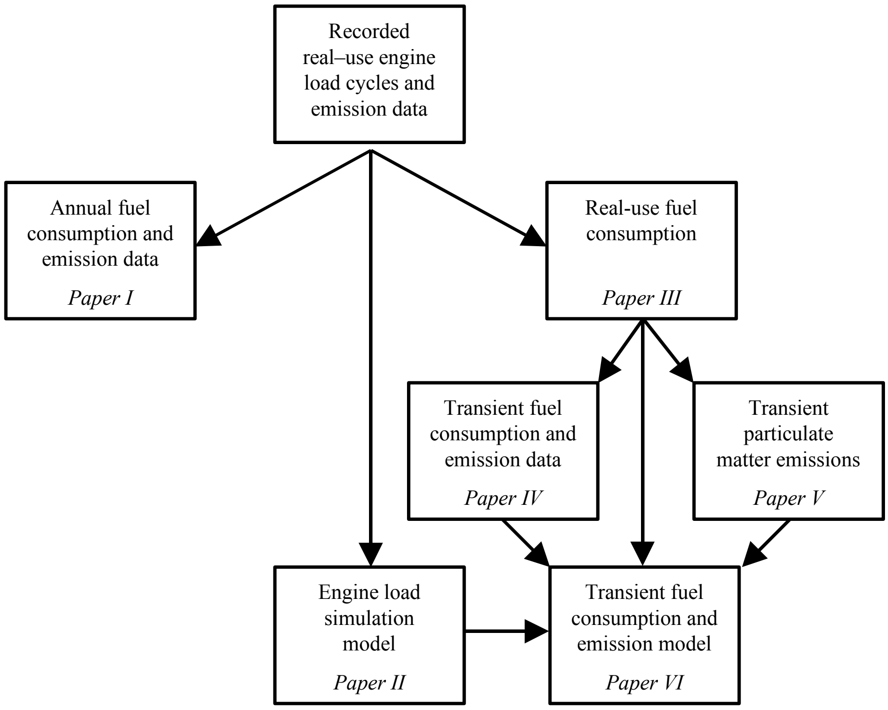

Fig. 1. Relationship between the individual papers included in this thesis.

The final paper, Paper VI, includes a presentation of a mathematical model for estimates of fuel consumption during transient conditions. This paper utilises the findings and results from Papers III to V and is implemented on the simulation model presented in Paper II.

Background

Busses, trucks, non-road mobile machinery and other types of heavy-duty vehicles are almost exclusively powered by diesel engines due to the high energy conversion and thus low fuel consumption. However, growing public concern with emissions from diesel engines, especially particulate matter and nitrogen oxides, and their impact on health and the environment has led to increasingly tighter regulations worldwide.

In Europe, the USA and Japan, there is an obligation for manufacturers to certify new engine models to specific emission performance standards, such as ISO 8178 or ECE R49 (ISO, 1996; EEC, 2000). These standards are used for all types of non-road mobile machinery, but are not adapted to conditions typical for different operations with non-road mobile machinery, such as agricultural operations (Treiber & Sauerteig, 1991; Rinaldi & Näf, 1992). Due to the varying use of non-road mobile machinery, it is not possible to design one specific standardised test cycle valid for the average use of non-road mobile machinery (Renius, 1994; Hansson *et al*., 1998; Lindgren *et al*., 2002).

In this thesis, the definition of non-road mobile machinery includes mobile machines, transportable industrial equipment and vehicles with or without body work not intended for use as passenger or goods transport on the road and equipped with an internal combustion engine with compression ignition. Non-road mobile equipment is covered by directive 97/68/EC, last changed by Directive 2004/26/EC, and 74/150/EC, last changed by 2000/25/EC, and includes industrial drilling rigs, compressors, construction equipment *e.g.* wheel loaders, bulldozers, crawler tractors, crawler loaders, truck-type loaders, off-highway trucks, hydraulic excavators, *etc.* and agricultural and forestry tractors (EU, 1974; 1997; 2000; 2004b).

To deal with impending new regulations, knowledge concerning fuel consumption and emissions from diesel engines under different load conditions, especially transient conditions, is of vital importance to manufacturers of engines and engine components such as turbochargers and fuel injection systems. With good knowledge of the load characteristics on the engine, it is possible to optimise engine performance more effectively in order to minimise the fuel consumption and the overall emission amounts. Another reason to search for more knowledge on the effects of transient loads on fuel consumption and emission levels is that data not including these effects underestimate the real fuel consumption and amounts of emissions produced (Ullman *et al.,* 1999; Lindgren *et al.,* 2002).

High quality data on fuel consumption and engine exhaust gas emission amounts from non-road mobile machinery are also needed in the calculation of environmental loads related to different food production or construction systems, for example by the use of life cycle assessment (LCA) methodology. Errors in fuel consumption and emission data can lead to inaccurate conclusions (Hansson & Mattsson, 1999; Udo de Haes *et al.,* 2002; Wrisberg & Udo de Haes, 2002).

Moreover these errors influence the national emission inventories and statistics, since in both the EU and USA, emission data are used for estimating national anthropogenic emissions (USEPA 1997; EEA, 1999). Members of the European Union report annual national anthropogenic emissions inventories to the European Union and other international conventions such as the United Nations Framework on Climate Change (UNFCCC), and the United Nations Economic Commission for Europe (UNECE) Convention on Long Range Transboundary Air Pollution (CLRTAP) (UNFCCC, 1992; EU, 1999; UNECE, 2002). In order to harmonise the data delivered, members of the European Union are encouraged to use the Intergovernmental Panel on Climate Change (IPCC) Guidelines for National Greenhouse Gas Inventories when estimating and reporting their annual national inventories (IPCC, 1996).

Several computer programmes for estimating national emissions from road transport have been developed within the European Environment Agency, for example COPERT III, in order to assist the member states to develop consistent and comparable national emission inventories (EEA, 2000). The main use of COPERT III is to calculate national traffic emission estimates on an annual basis. Total emissions are estimated from emission factors and activity data in the form of vehicle kilometres. The emission factors are divided into different categories depending on driving conditions, *i.e.* urban, rural and highway, and combined with average speed.

Emission factors commonly used in calculation programmes for emission estimates are based on either emission measurements obtained during different stationary test cycles such as the European Stationary Cycle and ISO 8178 or transient test cycles such as the FTP-75 test cycle for cars and light duty trucks or simulations of road traffic and specific operations (Sjödin *et al.,* 2003). Development of emission factors based on emission measurements from transient loads in chassis or engine dynamometers are time-consuming and expensive as they are only representative of the actual operation performed. On the other hand, simulation models based on emission maps have the potential to adjust the emission factors according to the actual operation performed. However, most engine maps are developed from steady-state measurements and are thus unable to satisfactorily describe emissions from real use of the engine or vehicle. Transient load conditions are often estimated by a succession of steady-state modes, thus describing the instantaneous steady-state fuel consumption and emissions. The difference between calculated steady-state values and real-use values including transient effects can be substantial. Correction measures must be treated with caution and take the actual workload and changes in engine speed and torque into consideration. For example, within the ongoing EU-project ARTEMIS, transient correction functions are developed in order to increase the accuracy during emission estimates for different driving conditions.

Non-road mobile machinery is used for a variety of different operations and the real fuel consumption and emission amounts are dependent on the actual use of the engine. The engine load characteristics for a specific operation can be influenced by alternative driving techniques, by the design of the drive train and by the use of implements with different work capacities (Danfors, 1988; Johansson, Färnlund & Engström*,* 1999). It has been found that many drivers operate their vehicles in a lower gear than necessary, causing the engine to operate at high speed (Krisper & Schimmel, 1985; Schrock *et al.,* 1986; Grogan *et al.,* 1987). The proper selection of gear and thus engine speed depends on the skill of the operator.

Transients in engine speed and torque, *i.e.* changes in engine speed and torque, occur frequently during the normal operation of non-road vehicles. Lindgren *et al.* (2002) have shown the occurrence of transients for several operations with non-road vehicles, from relatively low transient operations such as pumping urine fertiliser to high transient operations such as loading with a wheel loader. Similar results have been reported by Starr, Buckingham & Jackson (1999) and Ullman *et al.* (1999).

Moreover, Lindgren *et al.* (2002) showed that during normal operation with a wheel loader, the emissions of carbon monoxide increased by more than 200% compared with steady-state conditions. One reason for this comparatively large difference was the failure of the turbocharger to reach a fully spooled state at low engine speed in combination with high loading torque even at steady-state conditions, which the semi-static calculation model used for carbon monoxide was unable to reproduce. During the same operation the fuel consumption and emissions of nitrogen oxide and hydrocarbon increased by 14, 16, and 60% respectively.

There are two major methods for measuring real-use fuel consumption and emissions from engines. Firstly, recording of fuel consumption and emissions during field trials, a fairly difficult and expensive method. Secondly, recording of engine load characteristic during field trials and using these as engine load input data in engine dynamometer tests. A system for recording engine load during field trials is described by Pettersson *et al*. (2002).

Field trials that include sampling of engine load characteristics and especially engine emissions are expensive and time-consuming, particularly if the effects of different types of drivelines and operation strategies are to be studied. Furthermore, it is almost impossible to measure and control all variables that influence the engine emissions in the field. However, computer simulation provides a suitable tool for studying this and indicates conceivable measures. In the automotive industry, drive train modelling and simulation have been necessary tools for efficient product development (Pettersson, 1997; Rubin, Munns & Moskwa, 1997). However, for the agricultural sector there are only a few (specially designed) models, and no literature was found on models for special agricultural operations.

Existing studies on specific vehicles have concentrated on fuel consumption and measures to improve fuel efficiency (Krisper & Schimmel, 1985; Schrock *et al.,* 1986; Grogan *et al.,* 1987; Danfors, 1988). Several studies predict engine performance from an empirical relationship between engine speed, engine torque and fuel consumption, *i.e.* given any two of the variables, the third can be derived (Harris, 1992). The connection between fuel consumption and emissions strongly depends on engine utilisation. Therefore, a reduction in fuel consumption can also reduce emissions, although this is not a matter of course. The formation of nitrogen oxide, for example, is strongly correlated to engine torque and combustion temperature, not primarily to fuel consumption.

Effects on human health and the environment

Exhaust gas from diesel engines contains several components that are potentially hazardous to the environment and that cause symptoms such as respiratory problems for humans at high levels of exposure. However, when evaluating spatial effects of emissions from engines, consideration must be given to ambient conditions, driving pattern and ageing of the engine.

The Swedish government has established 15 national environmental quality objectives in order for Sweden to be a driving force and a model of ecologically sustainable development (Swedish Government, 1998). Some of these directly affect emissions from non-road mobile machinery, namely: clean air; limited influence on climate change; natural acidification only; no eutrophication; high-quality groundwater; a non-toxic environment; and a good urban environment.

Emissions of nitrogen oxides and volatile organic compounds, primarily hydrocarbons, form ozone and other oxidants such as peroxy-acetyl-nitrates under the influence of sunlight (Bosch, 1996). The reactivity is dependent on the concentrations in the atmosphere and on meteorological conditions. Emissions of nitrogen oxides also contribute to the eutrophication of soil and water. The effects of eutrophication include oxygen deficiency in water, decreased growth of forests and increased nitrogen content in groundwater (SEPA, 1993). Moreover, nitrogen oxides, particulates, ozone and carcinogenic substances such as different hydrocarbons, which are all frequently occurring in diesel engine exhaust gases, affect the cleanliness of the air and thus represent a health risk to humans, animals, plants and cultural assets.

Emissions of carbon dioxide, methane and nitrous oxide contribute to the increase in average global temperature and consequently affect the climate. Carbon dioxide is the most important compound, while methane and nitrous oxide are highly potential greenhouse gases, although sparsely occurring in diesel engine exhaust.

Acidification due to sulphur oxides is dependent on the sulphur content of the fuel and as Swedish environmental class 1 diesel fuel consists of maximum 10 ppm sulphur, the problem is limited. On the other hand, long-range transboundary emissions of sulphur oxides originating from combustion of fuels with a higher sulphur content are an important contributor to the overall emission amounts. For example, European non-road diesel consists of maximum 350 ppm sulphur. In 2009, a maximum sulphur limit of 10 ppm will be introduced within the European Union for both non-road diesel engines and heavy-duty truck and bus engines. Acidification due to nitrogen oxides is also a major problem. In Europe, non-road mobile machinery accounts for approximately 20% of the total emissions of nitrogen oxides (EEA, 2004).

Slowly biodegradable substances such as the polycyclic aromatic hydrocarbons present in diesel engine exhaust may accumulate in plants, animals and humans (SEPA, 1996; 2000). Accumulating organic toxic substances are especially harmful to top consumers in the food chain, *e.g.* humans.

In densely populated areas air pollution, primarily exhaust gas emissions but noise as well, sometimes reach levels hazardous to humans. Diesel exhausts have been shown to be potentially carcinogenic up to definitely carcinogenic for humans depending on organisation and risk evaluation model. For example, the International Agency for Research on Cancer (IARC) has declared that diesel exhaust gas emissions are probably carcinogenic for humans (IARC, 1989). Most human hazard assessment studies have been conducted on rats. However, there is no generally accepted method for transferring results obtained from studies on rats to humans.

Valberg & Watson (1996) question the usefulness of rat bioassay-based estimates of human risk. Studies have shown that rats grow tumours when exposed to diesel exhaust gas, while hamsters or mice chronically exposed to diesel exhaust do not produce tumours (Valberg & Watson, 1996). Furthermore, Morgan, Reger & Tucker (1997) strongly doubt the relationship between increased risk of cancer and diesel exhaust gas exposure.

Particulate matter emissions, which are considered by many to be the most important emission from diesel engines as regards human risk, consist of hundreds of different compounds adsorbed to the surface of a nucleus, which itself consists of a volatile fraction. Many of these compounds, such as benzene, 1,3-butadiene and formaldehyde, are regarded as mutagenic or carcinogenic (Neumann, 2001; Kawaga, 2002). Particulate matter is generally believed to cause decreased lung functions and increased respiratory symptoms such as chronic bronchitis, asthma and cardiovascular diseases (Siegmann, Scherrer & Siegmann, 1999; Kawaga, 2002; Morawska & Zhang, 2002).

In the urban environment, internal combustion engines are the major contributor to emissions of ultra-fine particles *i.e.* particles with a diameter of less than 100 nm (Morawska & Zhang, 2002). A high occurrence of symptoms typical of respiratory disease has been reported among people living in the vicinity of highly trafficked roads. Furthermore, studies have shown that inert particles may cause lung cancer (McClellan, 1986; Mauderly, 1997; WHO, 1999). Siegmann, Scherrer & Siegmann (1999) showed that ultra-fine particles could give rise to cardiopulmonary diseases and premature deaths because such particles can penetrate into the alveolus of the lungs due to their small size.

Morgan, Reger & Tucker (1997) demonstrated that eye and nose irritation, chest tightness and wheezing were common symptoms in people exposed to high concentrations of nitrogen dioxide and particulate matter. Exposure to carbon monoxide, a colourless, tasteless, odourless and non-irritant gas, can reduce the availability of oxygen to body tissues, even at rather low concentrations (Raub, 1999). Carbon monoxide absorbed into the bloodstream forms a slowly reversible complex with haemoglobin, thus decreasing the oxygen-carrying capacity of the blood. At high concentrations, exposure to carbon monoxide can result in death.

Emission regulations

Emissions from engines to be installed in non-road mobile machinery, as defined above, are regulated in Europe through Directive 97/68/EC, last changed by Directive 2004/26/EC and Directive 2000/25/EC (EU, 1997; 2000; 2004b). These Directives cover emissions of gaseous and particulate pollutants from internal combustion engines with compression ignition to be installed in non-road mobile machinery and agricultural and forestry tractors with a rated engine power of 18-560 kW. Emissions of gaseous and particulate pollutants from engines intended to power agricultural and forestry tractors are covered by Directive 2000/25/EC. The regulated emissions are carbon monoxide (CO), hydrocarbon (HC), nitrogen oxide (NO_x) and particulate matter (PM).

The legislation was introduced in two steps, stage I and stage II respectively. Each stage is divided into different categories depending on rated engine power, with different implementation dates and requirements as shown in Table 1. Agricultural and forestry tractors are regulated by a corresponding standard, as shown above, but with different implementation dates.

Table 1. *Implementation dates and emission limits for the European emission regulations, Directive 97/68/EC and 2000/25/EC*

Implementation date	Net power kW	CO g kWh^{-1}	HC g kWh^{-1}	NO_x g kWh^{-1}	PM g kWh^{-1}
Stage I					
1999.01/2001.07[a]	130≤P≤560	5.0	1.3	9.2	0.54
1999.01/2001.07[a]	75≤P<130	5.0	1.3	9.2	0.70
1999.04/2001.07[a]	37≤P<75	6.5	1.3	9.2	0.82
Stage II					
2001.01/2002.07[a]	130≤P≤560	3.5	1.0	6.0	0.2
2003.01/2003.07[a]	75≤P<130	5.0	1.0	6.0	0.3
2004.01/2004.01[a]	37≤P<75	5.0	1.3	7.0	0.4
2001.01/2002.01[a]	18≤P<37	5.5	1.5	8.0	0.8

[a] Agricultural and forestry tractors, Directive 2000/25/EC.

Furthermore, the legislation stipulates the use of a steady-state test cycle identical with the C1 cycle in the ISO 8178-4 standard. The C1 steady-state test cycle consists of eight modes of different engine speed and torque loads, see Table 2. Emissions are recorded for each individual mode and weighted together, according to different weighting factors, into one value for each emission.

Table 2. *Engine load and weighting factors stipulated by the ISO 8178 C1 regulation*

Mode	Engine speed	Torque %	Weighting factor
1	rated	100	0.15
2	rated	75	0.15
3	rated	50	0.15
4	rated	10	0.10
5	intermediate	100	0.10
6	intermediate	75	0.10
7	intermediate	50	0.10
8	low idle	0	0.15

However, during real use of the engine the load characteristics are of a transient nature. Transient engine loads occur during all kinds of operation and especially during different non-road operations. When developing new emission regulations, consideration should be given to the real use of the engine *e.g.* heavy-duty diesel engines should be tested according to a transient test cycle, the European transient test cycle (ETC) based on measurements of the real use of heavy-duty diesel engines (EU, 1999).

The same trends can be seen for non-road mobile machinery where a proposal, COM(2002) 765, for amending Directive 97/68/EC has been developed (EU, 2002). The above-mentioned proposal resulted in a new directive in the beginning of 2004, Directive 2004/26/EC (EU, 2004a; b). The new legislation will be introduced in two steps, stage III and stage IV. However, due to requirements imposed on engine manufacturers and the use of low-sulphur diesel fuel, emission limitations in stage III will be implemented in two separate parts, stage III A and stage III B. Stage III A will use the non-road steady cycle (NRSC) identical to the ISO 8178 C1 steady-state test cycle and the limit values of gaseous pollutants should virtually be the same as the USA regulations tier 3, see Table 3 (EU, 2004b, USEPA, 2004). However, the net power engines in the powerband 18 to 37 kW will be changed to 19 to 37 kW in order to harmonise with regulations in the USA. Emission limits and implementation dates for stage III A are presented in Table 3.

Table 3. *Implementation dates and emission limits for stage III A in Directive 2004/26/EC*

Implementation date	Net power kW	CO g kWh^{-1}	Sum of HC and NO_x g kWh^{-1}	PM g kWh^{-1}
2006.01	130≤P≤560	3.5	4.0	0.2
2007.01	75≤P<130	5.0	4.0	0.3
2008.01	37≤P<75	5.0	4.7	0.4
2007.01	19≤P<37	5.5	7.5	0.6

For stage III B a transient test cycle, the non-road transient cycle (NRTC), is proposed for measurements of particulate matter emissions only. The NRSC will still be used for measurements of gaseous pollutants. However, the manufacturer can choose to use the NRTC in stage III A and for the gaseous pollutants in stage III B and in the forthcoming stage IV as well. A technical review has to be carried out before the end of 2007 for amending the implementations dates and emission limits for PM and NO_x as there are some uncertainties regarding the cost effectiveness of the respective after-treatment equipment. Emission limits and implementation dates for stage III B and stage IV are presented in Table 4.

Table 4. *Implementation dates and emission limits for stage III B and IV in Directive 2004/26/EC*

Implementation date	Net power kW	CO g kWh^{-1}	HC g kWh^{-1}	NO_x g kWh^{-1}	PM g kWh^{-1}
Stage III B					
2011.01	130≤P≤560	3.5	0.19	2.0	0.025[a]
2012.01	75≤P<130	5.0	0.19	3.3	0.025[a]
2012.01	56≤P<75	5.0	0.19	3.3	0.025[a]
2013.01	37≤P<56	5.0	4.7[b]		0.025[a]
Stage IV					
2014.01	130≤P≤560	3.5	0.19	0.4	0.025[a]
2014.10	56≤P<130	5.0	0.19	0.4	0.025[a]

[a] Non-road transient test cycle.
[b] Sum of HC and NO_x.

The NRTC was developed by the US EPA in co-operation with authorities and industry in both the USA, Japan and the European Union and is based on real-use engine load characteristics for several different operations and vehicles. The operations and vehicles studied are presented in Starr, Buckingham & Jackson (1999) and Ullman *et al.* (1999) and represent typical American operations with non-road mobile machinery. The NRTC is also being proposed for use in the coming US EPA standards for Tier 4.

Compared to heavy-duty diesel engines, regulatory limits on emissions of gaseous and particulate pollutants for non-road mobile machinery are less strict. Regarding emissions of NO_x, regulated values are at least 20% higher depending on engine size, while the corresponding differences for PM are at least 100% for corresponding implementation dates. However, the engines are tested according to different test cycles and the recorded emission levels are not directly comparable but give an indication of the differences in engine exhaust gas regulations.

The diesel engine

The diesel engine is named after its inventor, Rudolf Diesel, who was born in Paris, France, in 1858 and educated at Munich Polytechnic, where he was an outstanding student and graduated with the highest marks ever awarded. During his employment as a refrigerator engineer at a manufacturer of steam and ice machines in Sulzer, Swiss, Diesel came in contact with ammonia as a working fluid in a closed system. This knowledge of thermodynamic work cycles proved to be of vital importance during the development of the diesel engine.

Unlike other contemporary developers of internal combustion engines *e.g.* Lenoir, Otto, Langen and Ackroyd, Rudolf Diesel based his work on thermodynamics and the Carnot cycle. Diesel's first attempt at constructing a diesel engine was based on high compression pressures, up to 300 bar, isothermal combustion and a theoretical efficiency of 75%. However, the isothermal combustion was not practically viable.

In 1893 Rudolf Diesel published his text "Theory for the construction of a rational thermal engine to replace the steam engine and other internal combustion engines currently in use" and clearly stated his intention of developing an engine to replace the steam engines of the time. In the same year, Diesel presented the first prototype engine, which was not functional. However, in February 1894 there was a breakthrough in the history of the diesel engine, when the engine was self operated for a whole minute. About a year later the efficiency was about 17% and in 1897 Diesel could present his first successful engine, often referred to as the first diesel engine.

The world's first diesel engine powered vehicle was a non-road mobile machine, a power plough presented by Benz-Sendling, later Daimler-Benz, in 1922. Since then the diesel engine has continued to develop and today features turbocharging, turbocompound, electronically controlled injection system *etc*. Diesel engines are the predominant power source in both non-road mobile machinery and in other heavy-duty applications.

Most diesel engines operate according to a 4-stroke process with heterogeneous mixture formation and spontaneous ignition. During the intake stroke, pure air is fed to the cylinder through the inlet manifold. The amount of air is dependent on engine load conditions and thus inlet and exhaust valve timing, boost pressure *etc.*

Often, diesel engines used in non-road mobile machinery are supercharged. In the most common supercharger used in non-road diesel engines, the energy is taken from the exhaust gases through a turbine and converted to mechanical work. Mechanical work is transferred to a compressor that compresses the induction air *i.e.* a turbocharger. In order to avoid unnecessary material stress in turbocharged diesel engines, maximum permissible peak pressure is limited through lower compression ratios (Bosch, 1996).

The cylinder air is compressed during the compression stroke and thus raises the air temperature above the fuel's ignition point. At the end of the compression stroke, fuel is injected into the cylinder at high pressure. The injected fuel atomises into small drops that penetrate and mix with the compressed air. Some of the mixed fuel vaporises, which causes a drop in both temperature and pressure, and thus this is an endothermic process.

The period between start of injection and start of combustion is called ignition delay. The ignition delay is dependent on a number of factors, including ignitability of the fuel *i.e.* cetane number, compression ratio, boost factor, fuel injection pump characteristics, induction air temperature and component temperature. During the ignition delay, portions of the fuel mix with the high-temperature air into combustible limits. The air-fuel ratio, lambda, varies from pure air beyond the fuel spray propagation to pure fuel in the spray core. Combustion is possible at a narrow range between lambda 0.3 to 1.5. Stoichiometric air-fuel ratios correspond to lambda 1. The combustion of the portion of fuel that has already mixed with air within flammability limits during the ignition delay is characterised by rapid combustion and pressure rise and thus high heat-release. For several reasons, it is desirable to keep the ignition delay as short as possible.

Combustion of the remaining portion of the fuel is dependent on turbulence and diffusion of air to the fuel-rich regions of the combustion chamber. The mixing of air and fuel is enhanced by controlled air pattern *i.e.* swirl. The swirl depends on both the shape and character of the induction system and on the shape of the combustion chamber. The diffusion-controlled combustion is characterised by a rather slow-burning flame and pressure rise.

During the expansion stroke, chemical energy bound in the fuel is converted into heat through combustion. Heated air and combustion products expand through an approximately adiabatic process, pushing on the piston and doing work. At the end of the expansion stroke, about 50° before bottom dead centre, the exhaust valve opens so that blowdown can assist in expelling the exhaust gases. Some of the energy remaining in exhaust gases can be used to drive a turbine, exhaust gas turbocharger or turbocompound. The last stroke of the 4-stroke engine is the exhaust stroke where the combustion camber is emptied of combustion gases by moving the piston upwards, back to the top dead centre. At the end of the exhaust stroke the inlet valve opens, thus causing a valve overlap period.

Emission formation

The combustion products resulting from complete combustion of a hydrocarbon-based fuel should theoretically consist of water and carbon dioxide (CO_2). However, the exhaust gas from a diesel engine consists of unburned hydrocarbons, partially burned hydrocarbons including carbon monoxide, particulate matter, thermal crack products and derivatives and other combustion by-products apart from water and CO_2.

Combustion by-products derive from the fuel, the lubricating oil or the intake air. Fuel and oil impurities and additives give rise to combustion by-products, *e.g.* sulphur in the fuel gives rise to sulphur oxides. The intake air consists of about 2/3 nitrogen gas, which during combustion partially reacts to form nitrogen oxides. Nitrogen oxides and hydrocarbons when exposed to sunlight can give rise to photochemical smog and ozone.

Carbon monoxide in the exhaust is formed during oxygen deficiency, which can occur locally in the cylinder. Furthermore, CO_2 forms CO at high temperatures by dissociation. However as a diesel engine operates with excess air, emissions of CO are rather limited and not regarded as a great problem.

The most important diesel engine exhaust gas emissions are NO_x and PM. The dominant source of NO_x is oxidation of atmospheric nitrogen. Nevertheless, fuels with a high content of nitrogen can contribute a considerable proportion of the total emissions of nitrogen oxides. Nitrogen oxides are composed of nitric oxide (NO) and nitrogen dioxide (NO_2), a reddish-brown poisonous and reactive gas. Nitric oxide is the dominant fraction by mass. Other nitrogen oxides are not covered by the emission regulations in spite of the fact that some of them are harmful, *e.g.* nitrous oxide.

The extended Zeldovich mechanism describes the principal reactions for the formulation of NO, which is briefly described below (Lavoie, Heywood & Keck, 1970; Bowman, 1975). Firstly, NO and nitrogen (N) is formed from atmospheric molecular nitrogen (N_2) and oxygen (O). Secondly, N and molecular oxygen (O_2) react and form NO and O. Thirdly, NO and hydrogen (H) are formed through a reaction of N and hydroxide (OH).

The formation of NO from atmospheric nitrogen is enhanced by temperature, almost in an exponential manner. Furthermore, high pressure and high oxygen content in the cylinder further accelerate the process. Nitrogen dioxide is formed through additional oxidation of NO. The chemical equilibrium of the NO_2/NO ratio should be low during combustion. However, NO formed within the flame zone transforms rather quickly to NO_2. Unless the reaction freezes by mixing with cooler gases, which is the case in diesel engines, the NO_2 formed returns to NO. About 10-30% of the total NO_x emission from diesel engines is in the form of NO_2, depending on engine speed and torque.

The formations of NO_x are enhanced by high temperature, pressure and oxygen content, which all are characteristics of the pre-mixed combustion in diesel engines. The maximum NO_x concentration within the cylinder arises shortly after the time of peak pressure. Combustion gases formed during the pre-mixed combustion are further compressed as the rest of the fuel combusts, whereby the temperature of the combustion gases increases and thus increases the formation of NO_x. As the expansion stroke proceeds, temperature decreases as a result of the expansion itself and of mixing with cooler gases, which freezes the formation of both NO and NO_2.

The amount of NO_x in the exhaust is approximately proportional to the amount of fuel injected (Heywood, 1988). However, the ignition delay is also important for the NO_x formation rate, due to the proportion of the pre-mixed combustion. Both injected quantity of fuel and the extent of the ignition delay are dependent on driving behaviour, not least during transient conditions.

Particulate matter (PM) consists of a carbon-based nucleus with absorbed compounds and is often divided into an insoluble fraction and a soluble fraction. Particulate matter primarily forms in the fuel-rich regions in the core of the fuel spray. As soon as the combustion starts, PM concentrations rise rapidly. However, once injection ends and the local air-fuel ratio increases due to mixing, oxidation of PM decreases the overall concentration. The majority of the PM formed in the early stage of the combustion process is oxidised before the end of the expansion stroke.

Small nuclei form in the fuel-rich core of the spray and the formation rate is enhanced by high temperature through pyrolysis of fuel molecules. At low temperatures, it is primarily heavy aromatics and highly unsaturated aliphatic compounds that contribute to the formation of nuclei. At intermediate temperatures, nuclei are predominantly formed through combustion under oxygen deficiency. At extremely high temperatures, nuclei probably form through a process involving carbon vapour. However, such high temperatures do not occur in diesel engines.

The nucleation gives rise to a large number of small but light particles, which chiefly gain in mass by surface growth. Surface growth involves incorporation of gas-phase hydrocarbon compounds to the nuclei. Individual nuclei or particles collide with each other and merge into larger units *i.e.* agglomeration. Agglomeration causes a decrease in the number of particles and a corresponding increase in size. Different types of agglomeration take place in the formation of particulate matter. At an early stage small nuclei collide and form a new single particle, which continues to increase in mass by surface growth. Once surface growth has ceased, individual particles merge into chainlike clusters of discrete particles.

Hydrocarbons (HC) in the exhaust gas are a result of either unburned or partially burned fuel or oil hydrocarbons (Bosch, 1996). Depending on engine operation conditions, different HC compounds are found in the exhaust. The chemical composition of the fuel, *e.g.* aromatics and olefins, also influences the composition of the exhausts.

Processes for the formation of HC emission can be divided into two main paths, formation of HC emission during the ignition delay and during the combustion. Some of the fuel injected during the ignition delay mixes beyond the ignition limit and is thus overlean. Overlean mixtures are too lean to autoignite or support a propagating flamefront. Only rather slow reactions oxidise this fraction of the fuel, which results in thermal crack products and derivatives. Hydrocarbon emissions due to overleaning during the ignition delay depend on the amount of fuel injected, turbulence of the combustion air *i.e.* swirl, mixing rate and other in-cylinder conditions such as temperature.

Another source of HC emissions during the ignition delay is locally over-rich mixtures, *i.e.* the core of the fuel spray. Over-rich mixtures can mix with air and completely combust later in the combustion process. However, combustible mixtures can give rise to emissions of HC through bulk quenching. Quenching occurs if the flame loses heat, *e.g.* due to cool cylinder walls. Furthermore, engine oil on the cylinder walls absorbs some of the fuel, which thus does not take part in the combustion. Absorbed fuel is released later during the expansion stroke and thus contributes to the HC emissions.

Fuel injected during the combustion can also give rise to emissions of HC through locally over-rich regions and bulk quenching. Another important source of HC is the fuel remaining in the nozzle sack. This fuel enters the combustion chamber late in the expansion stroke when the pressure has reduced and thus oxidises rather slowly due to the low temperature and limited time. Emissions of HC, which are an indication of combustion efficiency, vary greatly with engine load. Both local overleaning and undermixing *i.e.* over-rich mixtures, can significantly contribute to the total HC emissions during hard driving conditions such as heavy transients.

Engine characteristics during transient conditions

In previous research, several factors have been identified to explain how fuel consumption and emissions at transient conditions differ from those at steady-state conditions. One of these factors is the turbo-lag. It has been found that turbochargers have time lags of several seconds (Benajes, Luján & Serrano, 2000; Bane, 2002). Bane (2002) studied the behaviour of a turbocharger during trapezoidal variations in engine load. The engine load was oscillated with different periods from 20% full power to 100% full power while maintaining a constant engine speed of 1500 min^{-1}. The turbocharger was unable to respond to the variations in engine speed and torque during oscillations faster than a 4-s period. Benajes, Luján & Serrano (2000) found time lags of approximately 5 s before the final boost pressure was reached after an increase in engine torque from idle to full load with fixed engine speed.

Rakopoulos & Giakoumis (1999) showed that the response time for a mechanical governor, and thus the fuel pump rack position, to reach steady-state conditions after a steep increase in engine speed was considerable. While the engine speed was changed from 1500 min^{-1} to 1300 min^{-1} in 0.08 s, the governor needed more than 5 seconds to reach its final steady-state conditions. However, the drop in engine speed was very rapid and consequently severely stressed the governor *i.e.* a large change in spring deformation. Moreover, a change in engine speed means that the inertia of the engine changes as well. Under normal driving conditions, the engine makes use of more energy to increase the engine speed than is used during a decrease in engine speed *e.g.* braking.

Long-term thermodynamic transient responses, *i.e.* non-periodic temperature oscillations, have been studied by Rakopoulos, Mavropoulos & Hountalas (1998). During transient conditions in engine speed or torque, the turbocharger and the mechanical governor reached steady-state conditions in about 5 seconds and thus a rather stable air-fuel ratio. However, according to Rakopoulos, Mavropoulos & Hountalas (1998) the structural temperature needs several hundred seconds to reach a thermodynamic equilibrium, while the brake mean efficient pressure and in-cylinder temperature reach fairly steady-state conditions in a few seconds.

Most of the work on quantifying emissions during transient operation deals with either comparison of specific emissions and fuel consumption between steady-state cycles and transient cycles (Samulski & Jackson, 1998; Swain, Jackson & Lindhjem*,* 1998), or transient emission prediction models without any aim to specifically quantify the transient effects (Ouenou-Gamo, Ouladsine & Rachid*,* 1998; Ramamurthy *et al.,* 1998; Traver, Atkinson & Atkinson*,* 1999). The limited research reported in the literature has concentrated on engine acceleration and engine torque increase (Arcoumanis, 1992; Arcoumanis, Megarits & Bazari*,* 1994). However, Callahan *et al.* (1985a; 1985b; 1987) have studied the effects of discrete transients in engine speed and torque on the emissions from a naturally aspirated diesel engine, and found rather large differences between transient and static emission amounts.

Materials and methods

This thesis is based on engine load characteristics, fuel consumption and emission data recorded with several different methodologies applied on various vehicles and engines. Methodologies for emission measurements on individual engines in test benches, measurement systems for recording engine load characteristics and vehicle operation conditions in field are described below. Furthermore, models used and analysis conducted are also explained, together with the vehicles and operations studied.

Emissions measurements

Fuel consumption and engine exhaust gas emissions data, *i.e.* CO, HC and NO_x, were either obtained from the manufacturers or measured within project-EMMA at the Swedish Machinery Testing Institute located in Umeå, Sweden. Emissions were measured according to a 20-mode steady-state test cycle based on the 8-mode international ISO 8178 test cycle and extended with 12 additional modes (Fig. 2). All measurements were conducted in accordance with the ISO 8178 regulation (ISO, 1996). Particulate matter emissions were only measured as an average value for the whole 20-mode test cycle and thus not included in the steady-state analysis of operation-specific emission data, except for a few isolated cases (one engine) where PM emissions were analysed for all modes individually.

Emission measurements during transient conditions were conducted at the Swedish Machinery Testing Institute. Emissions of CO, HC, NO_x and PM were recorded and analysed in accordance with the ISO 8178 regulation (ISO, 1996). During transient conditions a partial flow of the engine exhaust gases was diverted to a specially designed heated steel container, which caused an average retention time of 90 s before sampling. Samples for analysis were taken continuously from the container until steady-state conditions were obtained. Measurements according to a 20-mode steady-state test cycle were also conducted as references. The engine test facilities and measurement systems are described in greater detail in Wetterberg *et al.,* (2002) and Paper IV.

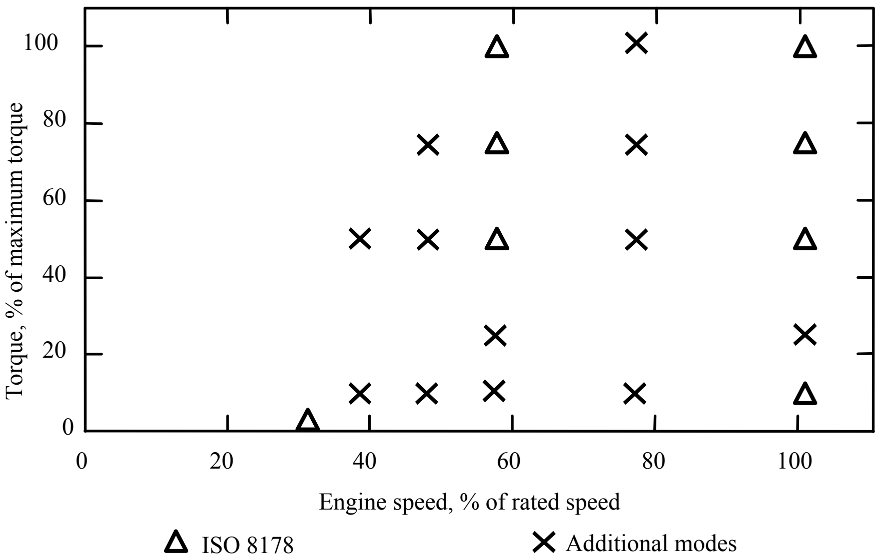

Fig. 2. The 20-mode steady-state test cycle based on ISO 8178 and extended with 12 additional modes.

Field measurement system

All of the vehicles studied were equipped with measurement systems for recording of engine load characteristics. The system included sensors for recording parameters such as engine speed, fuel consumption, fuel temperature, inlet air temperature, and vehicle speed through radar. A data logger mounted in the cabin of the vehicle sampled all sensors with a frequency of 1 Hz.

The system for measurement of fuel consumption is illustrated in Fig. 3. The transducer gave 250 pulses ml^{-1} with an accuracy of 0.5% and the measurement range was 0-40 l h^{-1}. A single pipe system was used to measure the fuel consumption excluding the leakage from the injectors and the pump. The leakage flow was returned to a compartment within the flow sensor. The fuel flow supplying the engine was taken from either the compartment or from the tractor’s diesel tank. Only the fuel flow directly from the diesel tank was measured. A more detailed description of the measurement system is given in Lindgren *et al.* (2002) and Pettersson *et al.* (2002).

The Valtra 6650 agricultural tractor used was also equipped with sensors for measuring engine torque directly on the engine axle. The construction of the tractor provides the potential to measure the torque loading of the engine at the axle between the engine and the transmission. Two strain gauge transducers were glued to the transmission input axle and connected to a full Wheatstone bridge to provide temperature compensation. Telemetric equipment from Datatel was used to transmit the signal to a receiver at the tractor body. The sender at the rotating part was supplied with power using an inductive system. In the measurements reported, the signals from the transducers were logged with a sample frequency of 200 Hz. However, before sampling, the signal was filtered with an analogue low-pass filter at 100 Hz. The mounting of the measurement equipment and the calibration are described in more detail in Nordin (2001).

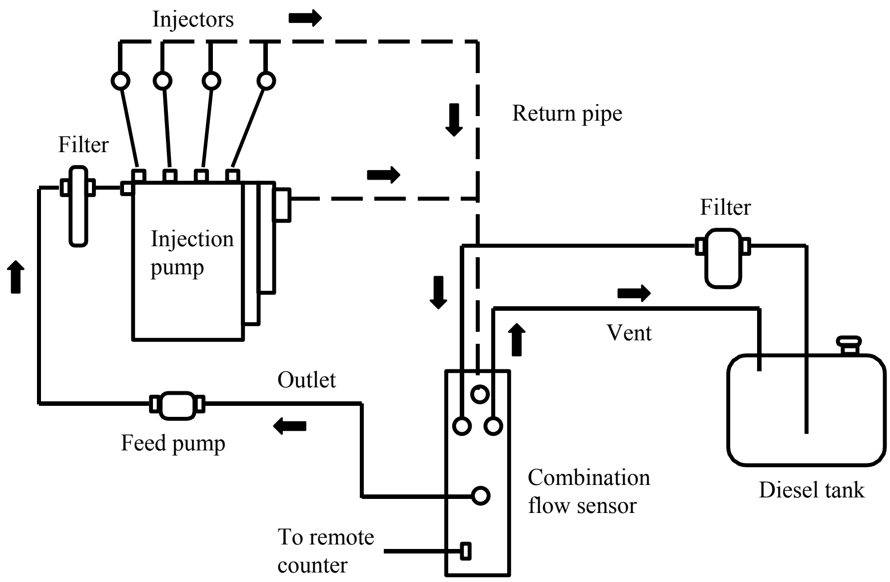

Fig. 3. Fuel flow in a fuel system equipped with a combined flow sensor for measurement of fuel consumption.

Analysis of field data

Based on static fuel consumption and engine load data, *i.e.* engine speed and torque recorded in test bench measurements, an expression expressing the engine torque as a function of engine speed and fuel consumption was developed. The proposed model was similar to that presented by Jahns (1983) and consisted of nine coefficients specific to the engine:

$$f = c_1 \times n + c_2 \times n^2 + c_3 \times n^3 + \left(c_4 \times n + c_5 \times n^2 + c_6 \times n^3\right) \times \tau + \left(c_7 \times n + c_8 \times n^2 + c_9 \times n^3\right) \times \tau^2$$

where: τ was engine torque in Nm, n was engine speed in min^{-1}, f was steady-state fuel consumption in $l\ h^{-1}$ and c_1 to c_9 were engine-specific coefficients. The coefficients c_1 to c_9 were found for each engine by least-squares regression.

Furthermore, matrices of emissions of CO, HC and NO_x for all possible steady-state combinations of engine speed and torque data for each engine, within the operational range of the engine, were developed by two-dimensional interpolation based on static emissions characteristics for each engine. Operation-specific emission data were calculated using time series of recorded engine speed data and calculated engine torque data as inputs in the matrices describing engine exhaust gas emissions. The validity of the semi-static method has also been confirmed by Hansson *et al.* (1998) and Cornetti *et al.* (1988) for operations mainly including rather slow variations in engine load.

Vehicles and operations studied

Eight different non-road mobile machines were studied over a total of 49 different types of operations. The field trials were conducted from 1999 to 2002 and included several repetitions of each type of operation. One individual operation for each type of operation and vehicle was chosen as representative. The selection was based on statistical methods and calculated the fit of the individual operation to the sum of all operations of that specific type. The vehicles studied were:

- Valtra 6600 agricultural tractor,
- Valtra 6650 agricultural tractor,
- Case IH MX 270 agricultural tractor,
- Massey Ferguson 7254 combine harvester,
- Volvo L50C wheel loader,
- Volvo L70 wheel loader,
- Volvo A25 articulated hauler,
- Volvo EW150C wheeled excavator.

A short description of each vehicle and the individual operations performed, including implement used, follows below.

The Valtra 6600 agricultural tractor was equipped with a 4-cylinder turbocharged engine, 420 DS, from Sisu-diesel with a rated power of 75 kW. The vehicle was delivered in 1996, before the implementation of stage I of the emission regulation Directive 2000/25/EC (EC, 2000). The following operations were recorded for the Valtra 6600 agricultural tractor:

- Baling with a trailed baler,
- Collection and transport of logs with a forest trailer,
- Filling of urine manure with a hydraulic pump mounted on the tractor,
- Harrowing with a 7 m wide 70-tine harrow in autumn,
- Harrowing with a 7 m wide 70-tine harrow in spring,
- Harrowing with a 7 m wide 85-tine harrow in spring,
- Mowing with a 2.1 m wide precision chopper,
- Mowing with a 2.8 m wide mower conditioner,
- Ploughing with a 5 furrow semi-mounted non reversible plough,

- Ploughing with a 4 furrow reversible plough,
- Rolling with a 6 m wide roller,
- Sowing with a 3 m wide seed drill,
- Spreading of mineral fertiliser with a fully mounted centrifugal spreader,
- Spreading of urine manure with a 15 m^3 capacity band spreader with trailing hoses, 12 m working width
- Transport on road with a trailer,
- Turning of hay with a 5 m wide tedder,
- Wrapping with a trailed bale wrapper.

The Valtra 6650 agricultural tractor was equipped with a 4-cylinder turbocharged engine, 420 DWRE, from Sisu-diesel with a rated power of 81 kW. The vehicle was delivered in 2000, and the measured amounts of emissions from the engine were less than the regulated levels in stage I according to Directive 2000/25/EC (EU, 2000). The following operations were recorded for the Valtra 6650 agricultural tractor:

- Filling of urine manure with a hydraulic pump mounted on the tractor,
- Front end loading,
- Harrowing with a 8 m wide four axle 72-tine harrow,
- Mowing with a 2 m wide mower conditioner,
- Ploughing with a 5 furrow semi-mounted reversible plough,
- Ploughing with a 4 furrow reversible plough,
- Ploughing with a 3 furrow reversible plough,
- Rolling with a 6 m wide roller,
- Sowing with a 3 m wide seed drill,
- Spreading of semi-liquid manure with a 11 m^3 capacity manure spreader, 10 m working width,
- Spreading of solid manure with a 8.3 m^3 capacity manure spreader, 6 m working width,
- Spreading of urine manure with a 15 m^3 capacity band spreader with trailing hoses, 12 m working width,
- Transport of manure to field,
- Transport on road with a 17 tonne trailer.

The Case IH MX 270 agricultural tractor was equipped with a 6-cylinder turbocharged and intercooled 24 valve engine, with a rated power of 240 kW. The vehicle was delivered in 2002, and the measured amounts of emissions from the engine were less than the regulated levels in stage II according to Directive 2000/25/EC (EU, 2000). The following operations were recorded for the Case IH MX 270 agricultural tractor:

- Ploughing with a 8 furrow semi-mounted non reversible plough,
- Sowing with a 6 m wide seed drill.

The Massey Ferguson 7254 combine harvester was equipped with a 18 feet cutterbar and a 6-cylinder engine, 634 DSBIEL, from Sisu-diesel with a rated power of 162 kW. The vehicle was delivered in 2002, and the measured amounts of emissions from the engine were less than the regulated levels in stage I according to Directive 97/68/EC (EU, 1997). The following threshing operations were recorded for the Massey Ferguson 7254 combine harvester:

- Barley,
- Oats,
- Rapeseed,
- Winter wheat.

The Volvo L50C wheel loader was equipped with a turbocharged 4-cylinder engine, TD 40 GFE, from Perkins with a rated power of 75 kW. The vehicle was delivered in 1999, and the measured amounts of emissions from the engine were less than the regulated levels in stage I according to Directive 97/68/EC (EU, 1997). The following operations were recorded for the Volvo L50C wheel loader:

- Application of road shoulder for edge support,
- Loading at a terminal,
- Loading of sand,
- Loading of snow,
- Mowing of roadside vegetation.

The Volvo L70 wheel loader was equipped with a turbocharged and intercooled 6-cylinder engine, TD 63 KDE, with a rated power of 96 kW. The vehicle was delivered in 1999, and the measured amounts of emissions from the engine were less than the regulated levels in stage I according to Directive 97/68/EC (EU, 1997). The following operations were recorded for the Volvo L70 wheel loader:

- Loading of gravel to a grading sieve,
- Loading of gravel to trailers at a quarry,
- Mowing of roadside vegetation.

The Volvo A25C articulated hauler was equipped with a turbocharged 6-cylinder engine, TD 73 KCE, with a rated power of 187 kW. The vehicle was delivered in 1998, before the implementation of stage I of the emission regulation, Directive 97/68/EC (EC, 1997). The following operations were recorded for the A25C articulated hauler:

- Transport of sand within construction site,
- Transport of soil over difficult terrain, and
- Transport on road with no load.

The Volvo EW150C wheeled excavator was equipped with a turbocharged 4-cylinder engine, TD 40 KEE, from Perkins with a rated power of 78 kW. The vehicle was delivered in 1998, and the measured amounts of emissions from the engine were less than the regulated levels in stage I according to Directive 97/68/EC (EU, 1997). The following operations were recorded for the Volvo EW150C wheeled excavator:

- Digging, levelling, etc with a rototilt bucket.

A more detailed description of the vehicles used and the operations studied is given by Lindgren *et al.* (2002).

Average annual engine exhaust gas emissions

Average annual engine exhaust gas emissions for one engine, Valtra 420 DS, were calculated using five different methodologies. Two of the methods were based on different emission standards frequently used for non-road mobile machinery, ISO 8178 and ECE R49 (ISO, 1996; EEC, 2000). Two other methods were based on the average annual use of an agricultural tractor, one representing the average sized tractor at the farm and the other representing the largest tractor at the farm, chiefly used for heavy soil cultivation operations such as ploughing and harrowing (Renius, 1994). The fifth and last method, the Deutz five-point test cycle, was based on five combinations of engine speed and torque assumed to be representative of agricultural conditions. The method was initially used by Klöcker-Humbolt-Deutz AG, developed by Renius & Bumcke (1980) and described by Welschof (1981) and Renius (1985). The methodology and calculation are described in greater detail in Paper I.

Engine load simulation model

A simulation model describing requisite speed and torque at the engine as a result of the transmission of forces and motion from tractor and implement is presented in Paper II. Fuel consumption and engine exhaust gas emissions were calculated as a function of simulated engine speed and torque data. The model was based on Newton's second law to determine the speed of the tractor v_v in m s^{-1} in relation to obtainable motive power and actual workload, as shown below:

$$m\dot{v}_v = F_e - \left(F_i + \Sigma F_{mr} + \Sigma F_p + F_a + F_r + F_d\right)$$

where: m was the mass of the tractor and implement in kg and useful force of propulsion on drive wheels F_e in N. Resisting forces were divided into different categories, namely: inertial force F_i, motion resistance of drive wheels ΣF_{mr}, pull force due to slope ΣF_p, air resistance of tractor F_a, motion resistance of implement F_r, and draft force of implement F_d, all in N.

Two different types of agricultural operations were simulated, transport and soil cultivation, *i.e.* harrowing. For each operation different engine control strategies and transmission characteristics were simulated. For example, restrained throttle setting during acceleration, selection of main gear during soil cultivation, changed field speed, changed size of implement, reduced final gear ratio, increased transmission efficiency, and the use of a continuously variable transmission adjusted for different constant engine speeds.

Effects of transient engine load conditions

Fuel efficiency in real driving conditions

Time series of engine speed, torque and fuel consumption were recorded for four different operations with varying degree of transient conditions, *i.e.* fast variations in engine speed and torque, with the Valtra 6650 agricultural tractor equipped with strain gauge transducers glued to the transmission axle. The first operation recorded, a transport operation with a 17 000 kg trailer on a flat surface, consisted of rather static conditions *i.e.* only small variations in engine speed and torque. The other three operations recorded, acceleration with a 17 000 kg trailer from stationary to approximately 35 km h^{-1}, on-farm driving without any external load on the tractor, and front end loading of gravel, represented typical agricultural operations with increasing proportions of transient loads.

Semi-static fuel consumption data *e.g.* as if the transient effects were neglected, were calculated from the recorded time series of engine speed and torque data using the semi-static method described above. The effect of transient conditions on the fuel consumption for the whole operation was calculated by comparison of the recorded fuel consumption with the calculated semi-static fuel consumption. The methodology is described in greater detail in Paper III.

Fuel consumption and emissions

Test cycles describing a specific change in engine speed or torque were developed in order to study the effects of transient loads, *e.g.* different rate of change in engine speed and torque, independently of each other. The test cycles were developed in order to form a complete cycle *i.e.* the last point of engine speed and torque in the test cycle was equivalent to the first point. An example of a test cycle over a positive transient in engine torque is shown in Fig. 4. A more detailed description of the test cycles is presented in Paper IV.

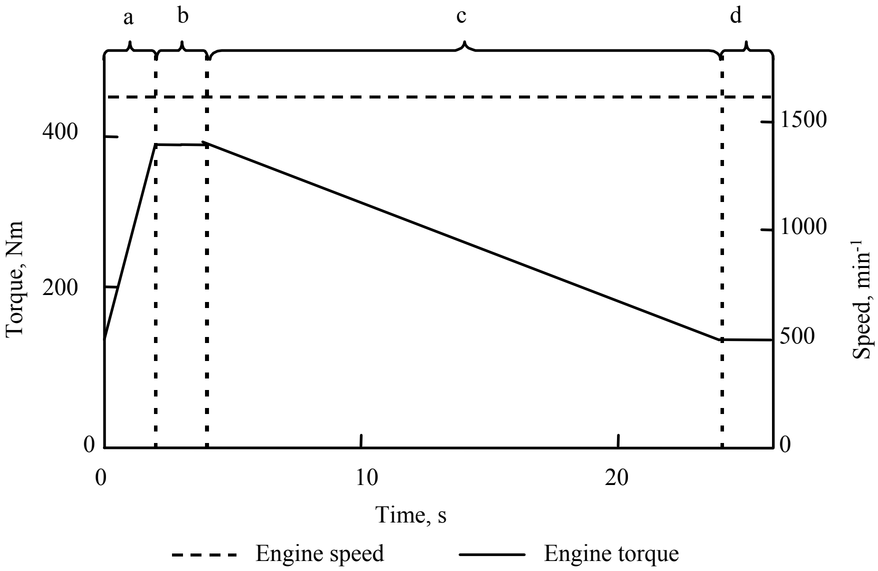

Fig. 4. Synthetic test cycle for a positive transient in engine torque and fixed engine speed: a, positive torque transient zone; b and d, steady-state zones; c, negative torque transient zone.

The absolute rate of change over the studied transient segment was varied between 10 and 40% s^{-1} depending on engine tested and emission recorded. The absolute rate of change over the other transient change section was kept constant at 2% s^{-1}. The rate of change was derived as normalised change of engine speed or torque per second as shown by:

$$\mathrm{d}n = \frac{n_i - n_{i-1}}{n_{rated} - n_0} \Big/ (t_i - t_{i-1}) \times 100$$

$$\mathrm{d}\tau = \frac{\tau_i - \tau_{i-1}}{\tau_{\max}} \Big/ (t_i - t_{i-1}) \times 100$$

where: dn and $d\tau$ were rate of change in engine speed and torque in % s^{-1}, n was engine speed in min^{-1} with subscripts i, $i\text{-}1$, *rated* and *0* representing engine speed at current time, engine speed at previous time-step, rated speed and low idle speed respectively, τ was torque in Nm with subscript *max* representing maximum available torque, and t was time in seconds with subscripts as above.

Four different categories of transient operations were studied: increase in engine speed, increase in engine torque, decrease in engine speed and decrease in engine torque. During transients in engine speed, the engine speed varied from 30% of rated speed to 70% of rated speed. The corresponding variations in engine torque were from 30% of maximum available torque to 70% of maximum available torque. Each category of the transient load change was repeated with three different levels of the constant variable, *i.e.* fixed engine speed during transients in torque and fixed torque during transients in engine speed.

Fuel consumption and emissions of CO, HC, NO_x, and PM were calculated using the semi-static method. The semi-static method was based on matrices describing steady-state fuel consumption and emissions of CO, HC, NO_x, and PM for all possible combinations of engine speed and torque within the operational range of the engine and used engine speed and torque data recorded in the test cycles as inputs. The effects of transients in engine speed and torque on the fuel consumption and emissions of CO, HC, NO_x, and PM were calculated as:

$$x = \left(\frac{E_M - E_C}{E_C}\right)\left(\frac{t_{cycle}}{t_{transient}}\right) + 1$$

where: x was the transient effect in g g^{-1} at steady-state, E_M was measured transient fuel consumption and emissions in g $cycle^{-1}$, E_C was calculated steady-state fuel consumption and emissions in g $cycle^{-1}$, t_{cycle} was the total cycle time in s, and $t_{transient}$ was the duration in seconds of the transient zone *i.e.* section a in Fig. 4.

The differences between measured fuel consumption and emission amounts and calculated fuel consumption and emission amounts were assumed to derive to the transient zone solely, hence the time quotient. At a rate of change of 0% s^{-1} *i.e.* steady-state conditions, measured fuel consumption and emissions were defined to be equal to calculated fuel consumption and emissions thus causing the ratio to be equal to unity.

The effects of transient conditions on the fuel consumption and emissions were studied on two engines, a TD 63 KDE engine originally mounted in a Volvo L70 wheel loader and a 420 DWRE engine originally mounted in a Valtra 6650 agricultural tractor. The TD 63 KDE engine was fitted with an in-line injection pump with a plunger and barrel assemblies with a lower helix only, thus causing the pumping to begin at the same stroke travelling independently of fuel quantity injected (Bosch, 1996). In contrast to an in-line injection pump, the delivery of fuel to the cylinders in a distributor pump always ends at the same stroke travelling independently of fuel quantity injected (Bosch, 1996). The distributor pump in the 420 DWRE engine was equipped with a centrifugally-controlled timing device for correction of start of delivery. Moreover, the distributor pump was also equipped with a boost control device, a boost pressure actuated diaphragm mechanism that adjusts the maximum fuel delivery from the pumping element relative to the boost pressure. The in-line pump was without corresponding characteristics. However, the in-line pump changed the injection characteristics during low loads and speeds.

Fuel consumption model

Based on the data from Papers III to V, a model was developed that expresses the fuel consumption as a function of engine speed n in min^{-1}, engine torque τ in Nm, and the rate of change in engine speed dn and torque $d\tau$ in s^{-1}. The transient fuel consumption model was based on two parts, a semi-static part and a transient correction part as shown by:

$$Z_t = Z_s(n,\tau)\times(1+R_t(n,\tau,dn,d\tau))$$

where: Z_t was transient fuel consumption in g h^{-1}, Z_s was semi-static fuel consumption in g h^{-1} and R_t was a dimensionless transient correction function. Transient fuel consumption data were derived for each time step and then integrated over time to obtain an accumulated value. In the first part, fuel consumption was calculated as if the transient effects were neglected, using engine speed and torque data as inputs to a matrix describing steady-state fuel consumption for all possible combinations of engine speed and torque. In the second part, a correction factor due to transients in engine load was calculated as the sum of the individual correction factors due to transients in engine speed and torque, firstly independently of each other and secondly as a synergism of the two as shown by:

$$R_t = q_n(\tau,dn)+q_\tau(n,d\tau)-cq_n(\tau,dn)q_\tau(n,d\tau)$$

where: q_n was a dimensionless correction factor due to transients in engine speed, q_τ was a dimensionless correction factor due to transients in engine torque, and c was a model-specific coefficient.

The proposed model was calibrated against time series of recorded engine speed, engine torque and fuel consumption data for four different operations with varying degrees of transient conditions described in Paper III. Furthermore, one complete engine load cycle with the 420 DWRE engine, describing a front end loading operation, was distinguished and repeated in an engine test bench for validation. In order to further validate the proposed model an additional engine was tested, a TD 63 KDE engine. The TD 63 KDE engine was tested according to another load cycle describing a loading operation at a quarry.

Recorded fuel consumption was compared with calculated fuel consumption including transient effects for both engines. Moreover, emissions of CO, HC, and NO_x were measured over the validation test cycles for both engines and the proposed model was employed, with the same coefficients, on exhaust gas emissions.

Results and discussion

Operation-specific fuel consumption and emission data

Recorded driving patterns were analysed with respect to fuel consumption and emissions at static engine load using the semi-static method described above. An example of an engine load pattern recorded during a ploughing operation with the Valtra 6650 agricultural tractor is given in Fig. 5.

The recorded operation included driving on headlands, turning *etc.,* everything that occurred during the operation. The recorded engine load pattern clearly shows different engine characteristics such as the brake torque curve. Figure 5 also indicates the type of operation performed *e.g.* average engine power utilisation, changes in engine torque and speed.

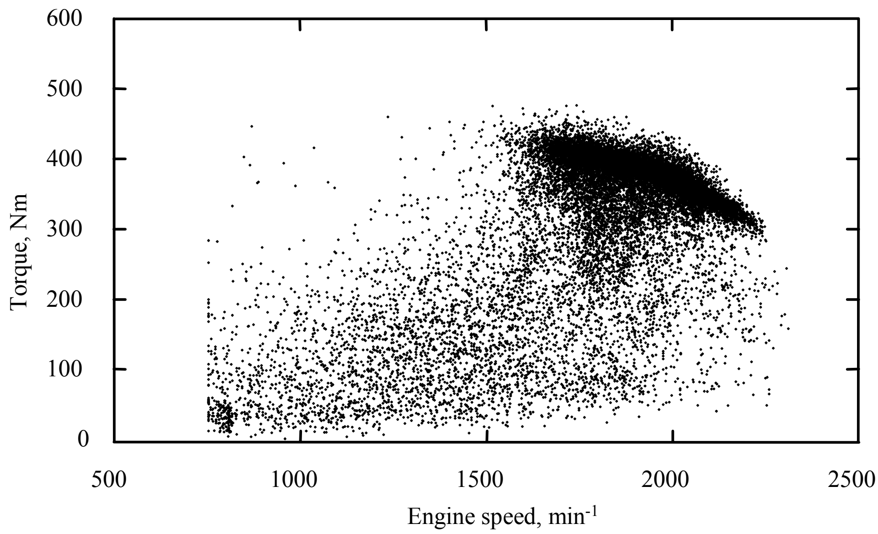

Fig. 5. Recorded engine load pattern during ploughing with Valtra 6650 agricultural tractor.

Recorded engine load patterns are in agreement with those presented by Hansson, Norén & Bohm (1999) and Hansson *et al.* (1998). However, data presented by Hansson, Norén & Bohm (1999) and Hansson *et al.* (1998) were based on sample times of 7.5 s, while the present study was based on sample times of 1 s. The higher resolution results in a better estimate of real-use engine load especially during transient operations.

Operation-specific fuel consumption and emission data for the Valtra 6600 agricultural tractor are presented in Table 5. Data on average engine power and cultivated area per hour, where required, are shown in the same table.

The results presented can easily be recalculated to g l^{-1} or g MJ^{-1} fuel consumed by use of the density (814 g l^{-1}) and energy content of diesel (35.15 MJ l^{-1}). For several of the agricultural operations presented, data can be expressed as g ha^{-1}, which is a useful quantity when comparing different working strategies, for example different sizes of implements.

Table 5. *Operation-specific fuel consumption and emission data for typical operations with a Valtra 6600 agricultural tractor*

Operation	Power kW	Work rate ha h^{-1}	Fuel kg h^{-1}	CO g h^{-1}	HC g h^{-1}	NO_x g h^{-1}
Baling	42	-	9.5	22.9	9.3	369
Filling urine manure	11	-	4.1	40.4	8.8	140
Harrowing, autumn [a]	53	4.3	12.1	28.9	11.1	404
Harrowing, spring [a]	54	4.3	12.0	28.5	11.1	403
Harrowing, spring [b]	57	8.0	12.8	27.0	10.2	469
Precision chopper	74	0.9	17.3	30.8	10.6	618
Mower conditioner	59	2.5	12.9	25.8	9.4	512
Ploughing, 5 furrow	51	1.1	11.6	27.5	9.4	449
Ploughing, 4 furrow	54	0.7	12.3	28.0	9.4	453
Rolling	16	3.7	4.4	20.3	5.6	201
Sowing	55	2.4	12.1	24.7	9.1	489
Spreading, mineral	16	11.0	4.4	32.3	7.2	181
Spreading, urine	28	4.4	7.0	25.0	8.6	267
Transport, forest trailer	1	-	3.0	36.1	6.2	148
Transport, on road	26	-	6.4	33.3	7.9	257
Turning of hay	24	2.2	6.2	25.8	7.9	241
Wrapping	20	-	5.0	22.3	5.9	236

[a] 70 tine; [b] 85 tine.

The variation in engine load between different operations was substantial *e.g.* from the forest trailer operation, which was mainly accomplished at low idle up to the heavy mowing operations with the precision chopper. However, the engine power of the Valtra 6600 agricultural tractor was rather limited compared to the power requirement of the precision chopper as shown by the large power utilisation, *i.e.* 74 kW on average of 75 kW available.

Operation-specific fuel consumption and emission data for the Valtra 6650 agricultural tractor are presented in Table 6, which also contains data on average engine power and work rate.

As with the Valtra 6600 agricultural tractor, the variation in engine load was substantial. Furthermore, the variation in engine load over time also differed between different operations from fairly static operations such as spreading of urine or manure and many other PTO-driven operations to operations with highly varying engine load, such as front end loading or several transport operations.

Table 6. *Operation-specific fuel consumption and emission data for typical operations with a Valtra 6650 agricultural tractor*

Operation	Power kW	Work rate ha h^{-1}	Fuel kg h^{-1}	CO g h^{-1}	HC g h^{-1}	NO_x g h^{-1}
Filling urine manure	18	-	5.9	28.0	9.5	102
Front end loading	13	-	3.8	22.6	6.6	77
Harrowing	72	4.9	14.8	99.8	10.3	716
Mowing	27	1.6	6.8	24.3	9.1	172
Ploughing, 5 furrow	80	0.65	17.2	126	11.0	810
Ploughing, 4 furrow	80	0.80	17.5	133	11.1	833
Ploughing, 3 furrow	77	0.58	16.5	103	11.2	750
Rolling	30	5.8	6.9	17.2	7.5	216
Sowing	33	2.0	7.6	21.2	8.4	243
Spreading, semi-liquid	43	6.9	10.3	29.5	11.4	310
Spreading, solid	48	2.4	10.5	33.6	10.4	385
Spreading. urine	27	6.7	6.7	23.5	8.5	181
Transport, manure	64	-	13.3	87.5	10.8	596
Transport, on road	58	-	12.3	62.5	10.9	496

Operation-specific fuel consumption and emission data for the Case IH MX 270 agricultural tractor are presented in Table 7. Data on average engine power and cultivated area per hour, where required, are also shown in Table 7.

Table 7. *Operation-specific fuel consumption and emission data for typical operations with a Case IH MX 270 agricultural tractor*

Operation	Power kW	Work rate ha h^{-1}	Fuel kg h^{-1}	CO g h^{-1}	HC g h^{-1}	NO_x g h^{-1}
Ploughing	169	1.9	34.4	109	11.7	1202
Sowing	114	6.6	24.3	79.0	11.2	679

The Case IH MX 270 agricultural tractor was equipped with an engine that fulfilled the stage II regulation according to Directive 97/68/EC as can be seen on the low emissions per mass of fuel consumed or average engine power. Compared to the Valtra 6650 agricultural tractor (stage I) and the Valtra 6600 agricultural tractor, emissions of both HC and NO_x were lower for the Case IH MX 270 agricultural tractor as well as the specific fuel consumption, *i.e.* g kWh^{-1}.

Average engine power, work rate, fuel consumption and emission data for the Massey Ferguson 7254 combine harvester are presented in Table 8.

Table 8. *Operation-specific fuel consumption and emission data for typical operations with a Massey Ferguson 7254 combine harvester*

Operation	Power kW	Work rate ha h^{-1}	Fuel kg h^{-1}	CO g h^{-1}	HC g h^{-1}	NO_x g h^{-1}
Barley	78	1.6	22.9	211	12.6	739
Oats	72	1.8	21.7	191	12.1	675
Rapeseed	72	2.4	21.2	175	11.5	680
Winter wheat	102	2.1	27.8	337	14.7	1045

The variation in engine load between threshing of different crops was small. However, the power demand of the threshing of winter wheat was somewhat higher compared to the others.

For the construction equipment, work rate is not included in the results as rather limited amounts of data were obtained on parameters such as average level of load of buckets or trailers. Data on average engine power, fuel consumption and emissions for different typical operations with the Volvo L50C wheel loader and the Volvo L70 wheel loader are presented in Tables 9 and 10, respectively.

The fuel consumption and emission data given in Tables 9 and 10 for the Volvo L50C and Volvo L70 wheel loaders, respectively, correlate rather well, except for emissions of NO_x. However, the low power loading operation at a terminal and the mowing operation, which utilised a rather large proportion of the available engine power, diverged to some extent.

Table 9. *Operation-specific fuel consumption and emission data for typical operations with a Volvo L50C wheel loader*

Operation	Power kW	Fuel kg h^{-1}	CO g h^{-1}	HC g h^{-1}	NO_x g h^{-1}
Loading, terminal	16	3.6	18.7	3.6	112
Loading, sand	27	6.0	22.7	4.2	175
Loading, snow	20	4.5	19.7	3.8	135
Mowing	24	5.2	17.8	3.9	151
Road shoulder	13	2.7	15.0	3.1	83

Table 10. *Operation-specific fuel consumption and emission data for typical operations with a Volvo L70 wheel loader*

Operation	Power kW	Fuel kg h^{-1}	CO g h^{-1}	HC g h^{-1}	NO_x g h^{-1}
Loading, grading sieve	23	4.9	18.0	3.7	220
Loading, trailer	29	6.1	20.3	4.0	265
Mowing	55	11.4	25.1	6.1	473

Table 11 shows fuel consumption and emission data for typical transport operations in different terrains with a Volvo A25 articulated hauler. Corresponding data for a digging operation with a Volvo EW150C wheeled excavator are presented in Table 12.

Table 11. *Operation-specific fuel consumption and emission data for typical operations with a Volvo A25 articulated hauler*

Operation	Power kW	Fuel kg h^{-1}	CO g h^{-1}	HC g h^{-1}	NO_x g h^{-1}
Transport, site	76	19.4	65.0	38.6	628
Transport, terrain	44	11.4	64.7	22.2	358
Transport, on road	90	20.7	95.9	26.8	727

Table 12. *Operation-specific fuel consumption and emission data for typical operations with a Volvo EW150C wheeled excavator*

Operation	Power kW	Fuel kg h^{-1}	CO g h^{-1}	HC g h^{-1}	NO_x g h^{-1}
Digging	49	10.3	32.5	8.5	433

Data presented in Tables 5 to 12 represent typical individual operations with different non-road mobile machinery. The average annual use of the vehicles and thus average annual fuel consumption and emissions amounts depends on factors such as farm size, production policy and on the total pool of machinery at the farm or construction site. Moreover, the operation-specific fuel consumption and emission amounts presented here clearly diverge from those obtained at different standardised emission regulations such as the ISO 8178 or ECE R49. For example for the Valtra 6650 agricultural tractor, emissions of NO_x were almost twice as high for the ploughing operation compared with the standard according to the 8-mode ISO 8178 C1. Moreover, no consideration was given to the engine's characteristics at high load variations *i.e.* transients. Many operations, especially with agricultural tractors, only include rather slow variations in engine speed and torque and thus can be analysed with the semi-static method described above. For operations including fast variations in engine speed and torque, such as different loading operations and some transport operations, the effects of transients will be considerable and should be included.

Average annual engine exhaust gas emissions

In Paper I, average annual emissions for a 420 DS engine originally mounted in a Valtra 6600 agriculture tractor were estimated using three different methodologies and compared with the widely used standards, ISO 8178 and ECE R49.

The comparison between the two standardised methodologies and the three methodologies based on average annual use of a tractor resulted in considerable differences in emitted amounts of both specific and absolute emissions of CO, HC and NO_x as shown in Figs. 6 and 7.

For a medium-sized tractor partly used for low-load operations such as front end loading and harvesting operations including hay-making, average annual specific emission values were up to 60% higher compared to the values obtained according to the ISO 8178 standard. The Deutz five-point cycle and the average annual use of a large tractor more frequently used for heavy soil cultivation operations showed similar values, although the differences were less pronounced. Specific emissions of NO_x were rather insensitive to the different weighting methodologies used, the difference between the highest (ECE R49) and the lowest (Deutz five-point) being 12%.

The results show that the standards, especially ISO 8178, stipulated for use according to the existing non-road mobile machinery emission regulation, underestimate average specific emissions of both CO and HC.

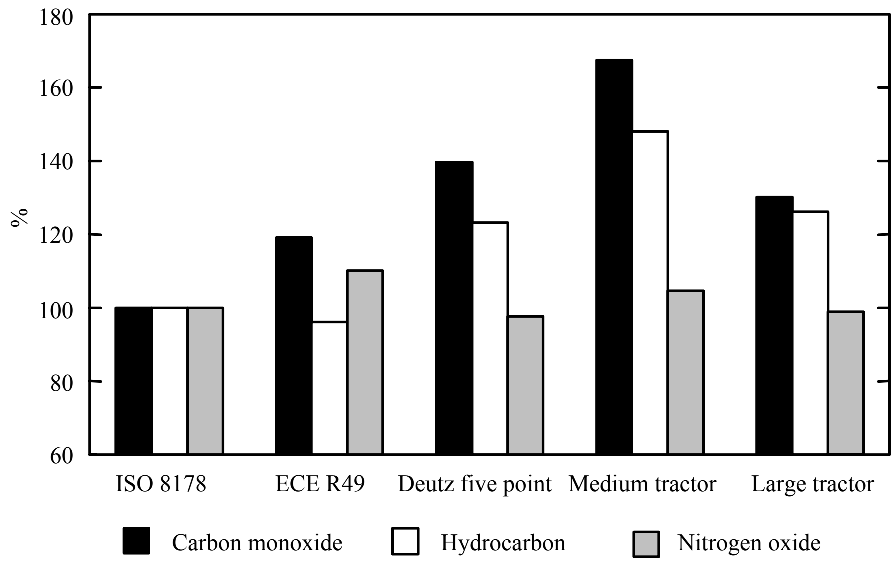

Fig. 6. Calculated average annual specific emissions relative to the ISO 8178 standard.

When comparing absolute emissions, the results were completely different. Emissions of HC for the three methodologies used was close to the value given by the ISO 8178 standard, while ECE R49 resulted in a rather low value. The most pronounced differences were shown in emissions of NO_x where all the three methodologies used resulted in considerably lower values compared with the two standards, down to less than 70%.

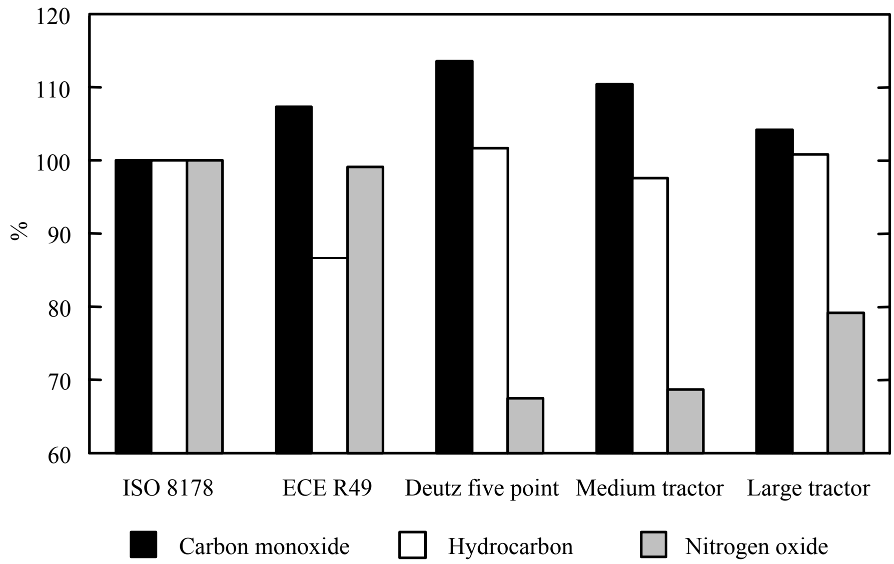

Fig. 7. Calculated average annual absolute emissions relative to the ISO 8178 standard.

Furthermore, the methodologies used showed that both average specific and absolute emissions were rather sensitive to real use of the tractor. The results indicate that it is not possible to design one set of weighting factors representative for all types of tractors, even less for all types of non-road mobile machinery. Due to the very high variation in both fuel consumption and emission amounts between different operations, the use of data from different standards such as the ISO 8178 or ECE R49, as for example in national emission inventories or life cycle assessment analyses, may give misleading results. Operation-specific emission data used in this study were based on one specific operation assumed to be representative for that type of operation. However, a specific operation can be executed in a variety of different ways depending on driver, vehicle, transmission, implement *etc.*, thus resulting in different emissions amounts.

Engine load simulation model

The average power demands for the transport and soil cultivation operations simulated in Paper II were 49 kW and 59 kW, respectively. Simulated fuel consumption and emission amounts were in agreement with recorded data for the corresponding operations reported in Table 6, with consideration given to ambient conditions, *e.g.* soil condition, speed of the vehicle, work rate, and load. However, both the recorded and simulated operations represent typical use of agricultural tractors but with different external loads and utilisation of implements. Fuel consumption and emission data for the reference scenario, without any modifications, are presented in Table 13.

Table 13. *Fuel consumption and emission amounts from the reference scenarios*

Operation	Fuel $kg\ h^{-1}$	CO $g\ h^{-1}$	HC $g\ h^{-1}$	NO_x $g\ h^{-1}$	PM $g\ h^{-1}$
On-road transport	12.3	55.3	22.4	393	16.7
Soil cultivation	14.0	40.2	17.8	564	14.9

The simulation of different engine control strategies and transmission characteristics showed that it is possible to affect the amount of emissions to a considerable extent without affecting the duration or fuel consumption of the operations. This also shows that fuel consumption alone is not a good parameter for estimating the resulting emissions for a specific operation, except for fuel-related emissions such as carbon dioxide or sulphur oxide.

Simulated changes can be divided into two different categories: one that requires changes in the mechanical construction of the tractor, the other that requires only a change in the driver's behaviour or the addition of fairly simple computer software.

An example of a change in the mechanical construction of the tractor is the use of a continuously variable transmission (CVT). A CVT is a potential tool to influence both the fuel consumption and emission amounts for a specific operation without affecting the work rate, *e.g.* the time for accomplishing a specified operation. A CVT can be adjusted so the most favourable combination of engine speed and torque can be attained for each type of engine load. Moreover, a CVT is, theoretically, able to change gear ratio during transient conditions and keep the optimal engine power trajectory relatively constant over time and thus minimise the transient effects on the engine (Pfiffner, Guzzella & Onder, 2003). The simulations showed that by the use of a CVT it was possible to reduce the fuel consumption and emissions of CO and HC without affecting the duration of the operation. However, emissions of NO_x increased with decreasing engine speed for both the transport and soil cultivation operations.

Another rather simple mechanical change of the transmission was to reduce the final gear ratio in order to reduce the engine speed at maximum permissible velocity. Such a change decreased the emissions of both CO and HC while emissions of NO_x increased. Fuel consumption and emissions of PM were only slightly affected. However, a change in the final gear ration did not affect the fuel consumption or emission amounts for other operations that take place in lower gears only.

The driver can also affect the maximum engine speed during transport operations by reducing the speed of the vehicle. However, in most cases a reduced vehicle speed only has little effect on the resulting fuel consumption and emission amounts for the whole operation but the total time taken to perform the operation increases with decreasing velocity.

The biggest challenge for the driver is to optimise work rate and at the same time minimise fuel consumption and emissions. The results from the simulations show that it is important to use implements that are correctly dimensioned to the power of the tractor. An undersized implement resulted in decreased work rate, *i.e.* increased time for the operation and a minor decrease in emissions of CO and NO_x. Fuel consumption was only slightly affected by a change in size of the implement, *i.e.* work width.

The work rate for a rather small implement can be increased by increased velocity. By increasing the velocity a larger proportion of the engine rated power can be utilised without affecting the fuel consumption for the operation. However, increased velocity causes increased emissions of CO, NO_x and PM. Furthermore, most soil cultivation operations have a narrow field-operation velocity range within which the operation generates desirable results.

Another conceivable method for the driver to affect the fuel consumption and emission amounts for a specific operation is the selection of gear and thus engine speed. Considerable amounts of fuel can be saved by selecting a higher gear ratio and reducing engine speed – 'shift-up, throttle-back'. This has also been confirmed by Tinker (1993) and Grogan *et al.* (1987). However, the simulation results showed that emissions of CO, NO_x, and PM increased with decreased engine speed and increased engine torque.

The choice of control strategy must be a question of priorities since it is clear from the results that it is normally not possible to obtain reduced emission amounts of NO_x and at the same time decrease the emission amounts of CO and HC. Furthermore, the simulation model was based on fuel consumption and emission data measured at steady-state conditions and thus no consideration was given to the engine characteristics at transient conditions. However, the only simulated modification that potentially affected the transient load on the engine was the use of a CVT. The results of the simulated modifications in control strategies have partly been confirmed by Lindgren *et al.* (2002). The authors studied the effects of different gear ratios, work widths and field velocities on the time, work rate, fuel consumption and emissions for a specific operation.

Effects of transient engine load conditions

Fuel efficiency in real driving conditions

Fuel efficiency for four operations with varying proportions of transient loads, *i.e.* variations in engine speed and torque, was studied in Paper III. The effects of transient loads on the fuel consumption varied from less than 0.3% for the fairly static transport operation up to approximately 13% during front end loading. The two other operations studied, acceleration with a heavy trailer and on-farm driving without any external load on the tractor, resulted in a decreased fuel efficiency of approximately 3 and 7%, respectively.

For operations only including rather slow changes in engine speed and torque, such as static transport or acceleration with a heavy trailer, the effects on fuel efficiency were rather limited. However, operations including fast variations in engine load, for example on-farm driving or front end loading, resulted in increased fuel consumption compared to the corresponding steady-state based fuel consumption.

The equipment for measuring torque was found to be very reliable and to have high accuracy. Time series of recorded engine speed and torque data for different typical operations can be used in various other applications, for example in modelling highly detailed instantaneous fuel consumption and emission data including transient effects. Furthermore, the equipment developed may be used for improving the transmission, engine control equipment, turbocharger, *etc.* with the aim of reducing the effects of transients. However, the equipment was rather complicated to mount and maintain, since parts of the tractor had to be disassembled in order to access it.

Fuel consumption and emissions

The effects of individual transients in engine speed or torque on fuel consumption and emissions of CO, HC, and NO_x from two non-road engines, a Valtra 420 DWRE and a Volvo TD 63 KDE, were studied in Paper IV. In Paper V the effects of individual transients on the emission of PM for the TD 63 KDE engine were studied. The results indicated that the air-fuel ratio was important for the fuel consumption and emission formation during transients. This has also been found under steady-state conditions by Taylor (1985) and Heywood (1988).

The fuel consumption and formation of emissions from internal combustion engines are intimately coupled with the conditions within the cylinder during combustion, *e.g.* air-fuel ratio, mixing, temperature, and how the fuel distribution changes with time (Heywood, 1988). During transients in engine speed and torque, the initial in-cylinder conditions resemble those of the preceding engine load conditions. However, depending on time delays the parameters react differently to a change in load conditions, *e.g.* the time delay for injected amount of fuel is less than the time delay for the boost pressure. This has also been confirmed by Rakopoulos, Mavropoulos & Hountalas (1998), Rakopoulos & Giakoumis (1999), Benajes, Luján & Serrano (2000) and Bane (2002).

The two engines studied were equipped with different types of injection system, which resulted in different effects of transient conditions. The boost control device reduced the amount of fuel delivered during accelerations from low engine loads with the 420 DWRE engine and thus decreased the amounts of fuel injected during positive transients in engine speed, as indicated by the results. The fuel consumption decreased down to approximately 75% of the fuel consumption during the corresponding steady-state condition. Engine TD 63 KDE was not equipped with a corresponding device and the amount of fuel injected increased accordingly, by about 10%. However, the in-line pump changed the injection characteristics during low loads, resulting in a minor decrease in fuel consumption for the 100 Nm transient in engine speed for engine TD 63 KDE. An example of the effects of transients in engine speed on the emissions of CO, HC, NO_x and PM for the TD 63 KDE engine is shown in Fig. 8.

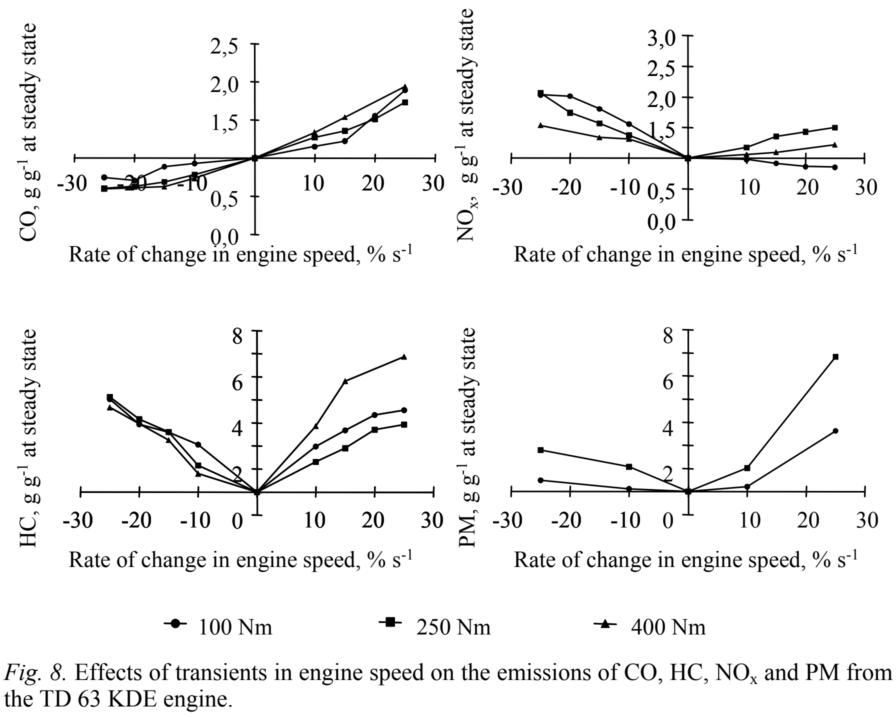

Fig. 8. Effects of transients in engine speed on the emissions of CO, HC, NO_x and PM from the TD 63 KDE engine.

At positive transients in engine torque, fuel consumption increased for both engines, by about 50% for the 420 DWRE engine and 20% for the TD 63 KDE engine. Compared with the transients in engine speed, the influence of the boost control device was limited. However, incomplete combustion due to low air-fuel ratio and locally over-rich regions in combination with a relatively low boost pressure probably increased the fuel consumption during positive transients in engine torque. Moreover, more heat from the combustion was needed to evaporate the injected fuel and increase the structural temperature. This could also be seen in the up to 150% increase in emissions of CO. Furthermore, a low boost pressure applies less work on the piston.

During negative transients in engine speed *i.e.* decrease in engine speed with maintained engine torque, fuel consumption increased with up to 100% compared to the corresponding steady-state conditions, a more pronounced increase for the 420 DWRE engine. The effects increased with increasing rate of change in engine speed. Similar results on the fuel consumption were obtained during negative transients in engine torque.

In diesel engines, torque is decreased by reducing the fuel supply but this study shows that a decreased fuel efficiency has the same effect. Rakopoulos & Giakoumis (1999) and Benajes, Luján & Serrano (2000) have shown that the response time in fuel metering on a change in fuel pump rack position is less than the corresponding response time for the turbocharger. The effects were even larger for the 420 DWRE engine, probably due to the more advanced fuel pump, *e.g.* boost control device and timing device. Furthermore, the high air supply and thus mixing rate in combination with the high structural and in-cylinder temperature were likely to decrease the fuel efficiency. To counteract the decreased fuel efficiency, more fuel was injected in order to maintain the desired engine speed and torque. The fuel consumption during negative transients in engine torque increased by approximately 80% and 40% for the 420 DWRE engine and the TD 63 KDE engine, respectively, compared with the corresponding steady-state conditions.

The effects of transient conditions on the emissions of NO_x were similar to the effects of transient conditions in fuel consumption. Emissions of NO_x increased during both positive and negative transients in engine torque, up to almost 200% during positive transients in engine torque for the TD 63 KDE engine. A similar increase in emissions of NO_x was shown during negative transients in engine speed for the 420 DWRE engine. During positive transients in engine speed the emissions of NO_x decreased to approximately 60% of the amounts at the corresponding steady-state conditions for the 420 DWRE engine and increased by 50% for the TD 63 KDE engine except during low engine torque where there was a minor decrease.

During negative transients in both engine speed and torque, emissions of CO decreased compared to the corresponding steady-state conditions thus indicating a more complete oxidation process. However, at the same time emissions of HC increased considerably, probably as a result of non optimal air-fuel mixing conditions and thus locally too rich or too lean mixtures. This indicates poor fuel efficiency as could be seen from the effects of transient conditions on the fuel consumption. Furthermore, overpenetration and bulk quenching of the flame were also likely to occur during transient conditions. Emissions of HC were shown to be extremely sensitive to transient conditions, especially for the TD 63 KDE engine where the emission amounts increased by more than 700% compared to the corresponding steady-state conditions.

Particulate emissions at transient conditions were only studied for one engine, the TD 63 KDE engine (Paper V). The combustion conditions during positive transients in both engine speed and torque are characterised by a rather low air-fuel ratio as the increase in injected amounts of fuel occurs more rapidly compared to the corresponding increase in boost pressure, *i.e.* turbolag. Those conditions are favourable for the formation of particulate matter emissions (Benjaes, Luján & Serrano, 2000). Other mechanisms that affect the formation of particles are overlapping of adjacent fuel sprays and impingement of liquid fuel on the cylinder walls at high loads (Heywood, 1988; Gomes & Yates, 1992). Under light loads, overlean mixtures increase the condensation and adsorption of unburned fuel on particles due to swirling (Gomes & Yates, 1992). The results showed that PM emissions increased considerably during positive transients, by about a factor of seven during positive transients in engine speed and up to more than a factor of ten at high rates of changes in engine torque.

During negative transients, a high air-fuel ratio in combination with high swirl promotes a more complete oxidation of particles formed early in the combustion process, as could be seen in the decreased amounts of PM emissions during negative transients in engine torque compared with the corresponding steady-state conditions. However during negative transients in engine speed, emissions of PM increased compared with the corresponding steady-state conditions, thus indicating that the formation and oxidation of PM are more complex than just air-fuel ratio and mixing conditions. One significant difference between negative transients in engine speed and torque is the change in internal motion of inertia of the engine. Due to the inertia, less energy is required to keep the engine torque at a constant value during negative transients in engine speed. Moreover, the mechanical governor is also affected by the change in engine speed and thus influences the formation of PM emissions. The surplus of fuel injected is likely to contribute to increased pyrolysis of fuel and condensation and adsorption of hydrocarbons on particles.

Fuel consumption model

A transient fuel consumption and emission model was developed in Paper VI, based on measurements and results from Papers III to V and Lindgren *et al.* (2002). Moreover, the proposed model was applied on the mechanistic engine load simulation model presented in Paper II.

The coefficient c was included in order to explain possible synergy effects due to changes in both engine speed and torque simultaneously. The coefficient was changed from 0 to 4 in steps of 1, and the best overall fit to the recorded data was obtained with c equal to 2.

The contribution from the transient correction function to the calculated overall fuel consumption during low transient operations, *i.e.* the static operation, was negligible. However, during operations with higher transient portion the transient fuel consumption model gave a considerable contribution to the resulting fuel consumption. In all measurements conducted, the transient fuel consumption model resulted in a better estimate of the real fuel consumption compared to the steady-state model alone.

The transient fuel consumption model, calibrated against fuel consumption data only, was employed both on another engine, the TD 63 KDE engine, and on emissions of CO, HC, NO_x, and PM. The transient model gave a better estimate of the real fuel consumption and emissions compared with the steady-state model only. However, for CO emissions from the TD 63 KDE engine, both steady-state and transient calculated amounts showed a major divergence from the recorded amounts. At low engine speeds in combination with high loading torque, the TD 63 KDE engine suffered from low air supply, thus causing high levels of CO emission of about 240 g h^{-1} even at steady-state, which the steady-state based model was unable to reproduce. At those engine loads the air utilisation was almost 100%.

The average difference between recorded and calculated emissions based on steady-state measurements was only about 30% for both the front end loading operation with the Valtra 6650 agricultural tractor, engine 420 DWRE and for the material moving operation with the Volvo L70 wheel loader, engine TD 63 KDE. However, emissions of CO for the TD 63 KDE engine were not included in this comparison. When the transient fuel consumption model was used, the average difference between recorded and calculated emission amounted to less than 7% for the front end loading operation and approximately 4% for the material moving operation, not including emissions of CO.

When the engine speed and torque data obtained from the engine load simulation model were utilised, the difference between fuel consumption based on steady-state measurements and fuel consumption estimated with the transient fuel consumption model was 2.4% and 5.3% for the soil cultivation and the transport operation, respectively. Based on recorded fuel consumption for similar engine load characteristics studied in Paper III, *i.e.* the acceleration operation and on-farm transport operation, calculated transient fuel consumption appeared to be accurately estimated.

General discussion

As cars and light duty vehicles have improved as regards fuel consumption and emissions of CO, HC, NO_x, and PM, the focus has shifted towards heavy-duty vehicles and especially non-road mobile machinery. However, fuel consumption and emissions from non-road mobile machinery are a complex area. In order to conduct projects within this subject, knowledge concerning the non-road mobile machinery sector and specific machinery is needed, as well as expert knowledge about engines, combustion systems, emission sampling and emission characteristics. This has also been recognised by Oscarsson, Grennfelt & Olsson (2000). This thesis was partly carried out within a Swedish project called EMMA, which is an example of a collaboration that brought the requisite knowledge together (Hansson *et al.,* 2002).

Reliable engine load, fuel consumption and emission data are a requirement in the development of new engines, fuel injection systems and after-treatment equipment for exhaust gases, as well as in the development of national emission inventories. This thesis presents average engine load, fuel consumption and emission data for typical operations using a variety of different non-road mobile machinery. In addition, methodologies for highly detailed studies of engine load variations are described, together with calculation models for fuel consumption and emissions that include the effects of transient loads.

For example, electronically controlled injection systems offer some potential advantages in the reduction of emissions and are becoming more and more popular. However, these systems are dependent on high quality sensors with high resolution, for example for engine speed and load, and on good control algorithms to be effective under transient load conditions. The methodology presented in Paper III could be used to evaluate the performance of the control algorithm of electronically controlled injection systems.

Furthermore, the high correlation between fuel consumption and emission amounts calculated according to the model presented in Paper VI and the data reported in Paper III and by Lindgren *et al.* (2002) indicate that the model is reliable. The methodology used is a rather straightforward way to calculate transient emissions. However, the emission characteristics used in the model are based on a specific engine. With a more advanced measurement system, for example online measurements of emissions, similar emission characteristics could be obtained using a rather simple and inexpensive approach. Based on recorded emission characteristics, including fuel consumption, operation-specific fuel consumption and emissions data can be estimated for all kinds of normal operations without the need for costly and time-consuming emission measurements in engine dynamometers. The only thing needed apart from emission characteristics is information regarding engine speed and torque for the operation in question. Nowadays, several non-road mobile machinery record data concerning operation conditions such as fuel consumption and engine speed for engine control or maintenance purposes.

Real-use fuel consumption and emissions from non-road mobile machinery are dependent on the actual operation performed and thus engine load. Manufacturers of engines and engine components often have a tendency to optimise the engines according to current emission regulations, thus causing the engines to be optimised for another area of use than the predominant engine utilisation for the machine in question. Test cycles not adapted to the real use of the machinery may cause the engines to consume unnecessary amounts of fuel and to emit large quantities of engine exhaust gas emissions when operated according to another engine load not covered by the emission regulations. Therefore, increased knowledge concerning engine load patterns for all kinds of non-road mobile machinery is important in order to design accurate test cycles valid for the real use of the engine. Moreover, in order to facilitate access to other markets, test cycles need to be harmonised. For example the proposed transient test cycle for non-road mobile machinery, NRTC, has been developed by the environmental protection agency in the USA in co-operation with authorities and industry in the USA, Japan and the European Union.

The coming transient test procedure will direct the interest from static engine load characteristics to transient, as can be seen in cars, light-duty vehicles and heavy-duty vehicles. Manufacturers of engines, transmissions and engine equipment such as turbochargers, fuel injection systems and electronic engine control units must adapt the combustion characteristics, the course of injection, response times *etc.* in order to comply with the transient and more stringent emission regulations.

The NRTC represents a rather modest transient operation and will initially be used for measurements of particulate matter emissions only. It will be introduced in stage III B. However, by the manufacturer can opt to use the NRTC also for measurement of gaseous emissions of CO, HC, and NO_x. Future emission regulations are likely to stipulate the use of a transient test procedure for the measurements of emissions of CO, HC, and NO_x as well. Engines optimised for a transient test cycle will probably result in reduced amounts of emissions even during other types of transient load conditions thus leading to lower total emissions amounts compared with the non-road mobile machinery of today.

However, it could be questioned whether the engine load pattern in the NRTC is valid for all types of operations with non-road mobile machinery covered by the directive or even representative for the typical use of a specific non-road mobile machine. It is obvious that the engine load pattern for a generator or an arc welder diverges from that of a wheel loader or an articulated hauler and thus generator sets are exempted from the transient test procedure described in Directive 2004/26/EC (EU, 2004b).

Moreover, parts of the transient test cycle were based on agricultural operations with a 190 kW tractor performing soil cultivation operations on a 1 km wide field. Four passes were conducted with each implement and regarded as ‘normal’ operation with the implement tested. The average European tractor has about half the rated engine power of that used in the preliminary study of the construction of the NRTC. However, the engine load pattern for a specific operation is rather independent of size of the tractor *i.e.* rated engine power, on condition that the implement is correctly dimensioned to the engine rated power. Nevertheless, the typical field width is less than 1 km, resulting for example in more headland driving. Furthermore, the emission regulations only apply to the engine whereas the emissions for a specific operation are a result of the use of the whole machine, including losses for example in transmission or hydraulics.

In order to comply with future emission regulations, the use of aftertreatment equipment to reduce emissions of both PM and NO_x will most likely be needed. Engine technology and technology for post-engine treatment of exhaust gases developed for heavy-duty vehicles cannot directly be applied on engines to be installed in non-road mobile machinery without considerable adjustments. The injection of a reducing agent during operation in a selective catalytic reduction system, *i.e.* NO_x-reduction, or the regeneration of diesel particulate filters is dependent on engine load characteristics. For example, continuous regeneration of particulate filters is based on high exhaust gas temperatures in order to oxidise particulate matter captured in the filter. However, during many operations with non-road mobile machinery, exhaust gas temperatures are lower than the ignition temperature of the filter. A particular problem could be operation during transient conditions. Many non-road mobile machines are used for operations that are largely transient in nature.

Transient conditions result in a highly varied load on the aftertreatment equipment due to non-optimal combustion conditions. The effects of the transient loads are caused by several factors. One major factor is that the injection system is not able to optimise the amount of fuel injected when the engine speed or load is varied. Often, too much fuel is injected in relation to the amount of air available for optimal combustion. This is especially the case for engines with turbochargers and with mechanically controlled injection systems.

Engines could also be equipped with *e.g.* a boost controller that decreases the quantity of fuel injected during low boost pressures and thus the duration of the periods when the amount of air available for combustion is not sufficient for optimal burning of the amount of fuel injected. It is reasonable to expect that the effects of the transient loads would be even higher on engines without this device. However, since the design and dimensioning of boost controllers, turbochargers and after-coolers show large variations between different engines, there is a need to study the behaviour of all these devices with transient loads. It may also be possible to use the methodology described in Paper III to improve them by decreasing the transient effects. Similarly, it would be possible to study the influence of various drive train and transmission design features on the transient effects.

Another possible method to reduce the effects of transient conditions on fuel consumption and emissions could be to limit the occurrence of transients during normal operation, for example by alternative hydraulic systems or the recovering of potential or kinetic energy. A system for the recovering of potential energy from the lowering of the arm system on hydraulic lifting machines has been developed by Bruun and tested at the SP Swedish National Testing and Research Institute (Hoseini, 2000). This system could also be developed in order to recover the energy from retardation, for example during a loading operation with a wheel loader, and thus decrease the amount of fuel necessary for the operation. Such a system not only has the potential to decrease the total fuel consumption but also to decrease the amount of transients, as stored potential or kinetic energy can be used together with the engine power during for example acceleration or a lifting operation. Based on engine load spectra for the operations recorded within this thesis and the effects of transients, it would possible to identify the operations and vehicles that could be equipped with a corresponding energy recovery system and to estimate their effects.

However, the most efficient method to reduce the fuel consumption and emissions for a specific operation would be to omit the operation if the desired result of the operation can be obtained in another way. In agricultural soil cultivation systems, several different operations are combined in order to obtain the desired result. Depending on the operations combined, fuel consumption decreased by more than 20% in a study by Norén e*t al.* (1999) comparing a traditional soil cultivation system including ploughing with a reduced soil preparation system without ploughing. On a corresponding systems level, reduction of the total fuel consumption and emissions for a construction site could be possible. Moreover, by optimising the utilisation of the total machinery system and minimising of the periods of idling *etc.*, it would be possible to save considerable amounts of fuel.

Future research

This study has identified several subjects for future research, of which some are presented below.

Fuel consumption and emission formation are dependent on the fuel used. The chemical composition, heat of vaporisation, viscosity *etc.* of alternative fuel differs from that of conventional diesel fuels, thus resulting in completely changed chemical and thermodynamic processes in the combustion chamber. Non-regulated emissions, for example specific hydrocarbon compounds, are especially likely to result in considerable differences when the composition of the fuel is changed. During transient conditions the differences would probably be even larger.

Emissions of for example formaldehyde, acetaldehyde, propene, 1.3-butadiene, di- and polyaromatic hydrocarbons are potentially hazardous for humans. However, these compounds are not regulated today and only a limited amount of studies have been conducted, especially for non-road mobile machinery. The influence of the composition of the fuel and the effects of engine load including transient conditions need to be further studied.

Several studies have shown that particulate matter emissions could give rise to negative effects on human health and even cause premature deaths. Most epidemiological studies conducted recently have focused on particle size and size distribution, especially ultra-fine particles, *i.e.* particles smaller than 100 nm. However, the standard method for measuring particulate matter emissions from diesel engines provides mass concentration. Therefore, more research is recommended to identify the particle sizes and size distribution in the exhaust from non-road mobile machinery, with emphasis on real-use engine loads including transient conditions.

The influence of different supercharging, injection and operating systems on the fuel consumption and emissions during transient conditions should be further studied. Does for example mechanical supercharging give any advantages during transient conditions compared with conventional turbocharging?

New engines apply electronically controlled injection systems, which offer the potential to control start of injection, injection rates, *etc.* However, these systems rely on accurate and fast sensors for engine load estimates and on good control algorithms. The performance of electronically controlled injection systems needs to be analysed, particularly during transient conditions,.

Reliable estimates of annual emissions from the total non-road mobile machinery sector are important for national and international emission inventories. In this thesis, operation-specific emission data for several non-road mobile machinery are presented. However, data concerning the quantity and average annual use of non-road mobile machinery are limited and more research within emissions statistics is recommended.

A road or building can be constructed using various methods with different non-road mobile machinery requirements. Grain can be produced using different soil preparation methods and thus draught requirements. Studies of entire systems of non-road mobile machinery applications need to be performed in order to analyse their effects and potential.

Manufacturers of engines are obliged to certify new engines according to different emission regulations and the development of emissions standards and new engines indicates that less pollutants should be emitted. However, such engines are installed in different machines, which may have a poor efficiency in converting engine power to useful work. The influence of optimising the whole engine-machinery system needs to be further investigated.

The function of diesel particle filters is closely related to the particulate matter load on the filter and on the temperature of the gases reaching the filter due to regeneration. The engine load spectra for different operations with non-road mobile machinery are strongly varying and thus particulate matter load and exhaust gas temperature also vary greatly. Time series of engine load data for typical operations are scarcely reported in the literature. However, such data would be of vital importance for the development and adaptation of diesel particle filters for non-road mobile machinery.

Exhaust gas aftertreatment technologies, especially particle filters, are currently being applied, chiefly through retrofit, on many non-road mobile machines. Due to adverse conditions during several operations with non-road mobile machinery, the equipment used may be damaged and the function reduced. The conditions for retrofit of aftertreatment technologies on non-road mobile machinery require further investigatory work.

The development of aftertreatment technologies for exhaust gases has focused on the direct effects on the composition of the exhaust. Knowledge is limited as regards the environmental effects in a lifecycle perspective from raw material acquisition to final disposal including processes, transportation, manufacture and use of aftertreatment equipment.

Conclusions

The difference in engine load and thus fuel consumption and emissions between different operations with non-road mobile machinery is considerable. Moreover, emissions measured according to different standards, *i.e.* ISO 8178 or ECE R49, are not representative of the average annual use of non-road mobile machinery. Therefore, it is generally not possible to design one standardised engine load pattern representative of all types of operations and machinery.

It is possible to influence the fuel consumption and emission amounts for a specific operation by using different driving strategies or transmission characteristics without affecting the duration of the operation. However, this must be a question of priorities, since it is normally not possible to reduce both fuel consumption and emissions of CO, HC, NO_x, and PM at the same time

The methodology presented based on engine torque measurements during real use appears to be a conceivable approach to estimate the effects of transient load conditions on the fuel consumption during different operations. When tested on different operations, the measurements showed that the average fuel efficiency decreased with increasing proportion of transient load conditions.

Depending on engine characteristics including fuel and air supply systems, the effects of transients in engine speed and torque differ. Through different design and construction of turbochargers, boost control devices and injection pumps, it is possible to influence the fuel consumption and emissions from engines to be installed in non-road mobile machinery.

Transients in engine speed and torque have a major effect on fuel consumption and emissions of CO, HC, NO_x, and PM. Transient conditions give rise to substantially increased amounts of emissions. For example in this study, PM emissions increased by a factor of more than ten compared to those in the corresponding steady-state conditions.

With a limited amount of measurements in an engine dynamometer, it is possible to estimate the fuel consumption and emissions for all types of operations and the use of a specific engine with high accuracy. The model proposed was tested on different engines and operations and resulted in reliable estimates of both fuel consumption and emissions.

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