Diesel Engine Transient Operation

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Principles of Operation and Simulation Analysis

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Preface

Traditionally, the study of internal combustion engines operation has focused on the steady-state performance. However, the daily driving schedule of automotive and truck engines is inherently related to unsteady conditions. In fact, only a very small portion of a vehicle's operating pattern is true steady-state, *e.g.,* when cruising on a motorway. Moreover, the most critical conditions encountered by industrial or marine engines are met during transients too. Unfortunately, the transient operation of turbocharged diesel engines has been associated with slow acceleration rate, hence poor driveability, and overshoot in particulate, gaseous and noise emissions. Despite the relatively large number of published papers, this very important subject has been treated in the past scarcely and only segmentally as regards reference books. Merely two chapters, one in the book *Turbocharging the Internal Combustion Engine* by N. Watson and M.S. Janota (McMillan Press, 1982) and another one written by D.E. Winterbone in the book *The Thermodynamics and Gas Dynamics of Internal Combustion Engines, Vol. II* edited by J.H. Horlock and D.E. Winterbone (Clarendon Press, 1986) are dedicated to transient operation. Both books, now out of print, were published a long time ago. Then, it seems reasonable to try to expand on these pioneering works, taking into account the recent technological advances and particularly the global concern about environmental pollution, which has intensified the research on transient (diesel) engine operation, typically through the Transient Cycles certification of new vehicles.

For a number of years now, the vast majority of diesel engines have been turbocharged and this trend is sure to continue. Although turbocharging the diesel engine is beneficial because it increases its (specific) brake power, and also because it provides better fuel economy and reduced CO_2 emissions, it is the turbocharged diesel engine that suffers way more than its naturally aspirated counterpart from poor transient response. This originates in what is known in the engine community as 'turbocharger lag', which is the key factor responsible for the slow speed response and heavy exhaust emissions. Consequently, the turbocharged diesel engine will be the focus of analysis in this book, with the behavior of its naturally aspirated counterpart highlighted only on specific aspects (*e.g.,* cold starting). As a matter of fact, the title of the book could well have read *Turbocharged Diesel Engine Transient Operation*.

Although there are many operating schedules experienced by diesel engines that can loosely be termed transient, we have focused on the most influential ones in terms of engine performance and exhaust emissions, namely load acceptance, acceleration and cold starting, as well as their combinations, most notably in the form of Transient Cycles.

Emphasis in the book is placed on the in-cylinder thermodynamic discrepancies, exhaust emissions and methods of improving transient response. However, it has been our intention to cover the subject from all relevant aspects; consequently, the interested reader will be able to find information on areas of 'lesser popularity' such as second-law (exergy, availability) analysis, compressor surging or crankshaft torsional deformation during transients. Although automotive applications are usually the main case studied, industrial or marine engines' transient response is dealt with too. Moreover, the analysis is thermodynamics rather than control oriented. Control matters are only briefly discussed, mainly in Chapter 6, where the effects of the various control strategies on the engine transient response improvement are pinpointed.

The book is organized as follows: in Chapter 1, an introduction to transient diesel engine operation is given, highlighting various typical load acceptance and acceleration schedules of both automotive and industrial/marine engines, and detailing the importance and different evolution pattern of transient operation compared with steady-state conditions. Chapter 2 describes the complex thermodynamic issues of transient operation (located in the fuel injection mechanism, heat transfer, combustion, air-supply and exhaust gas recirculation processes), starting, of course, with the discussion of the fundamental turbocharger lag problem. Chapter 3 focuses on the dynamic issues encountered during transients, namely friction overshoot, components stress and crankshaft torsional deformation. Chapter 4 discusses briefly the experimental procedure involved in transient diesel engine research, addressing in more detail the instantaneous particulate matter measurement techniques and the heat release analysis of transient pressure data. Chapter 5 deals with the very important aspect of exhaust emissions during transients, and Chapter 6 with the various methods developed over the years for reducing the turbocharger lag phase and improving transient response. Chapter 7 focuses on other aspects of transient conditions, namely cold starting, operation when the turbocharger compressor experiences surge and low-heat rejection engine operation. Chapter 8 provides an alternative coverage of the subject through the perspective of the second law of thermodynamics. Finally, Chapter 9 summarizes the various modeling approaches developed over the years for the simulation of transient operation.

This book is the outcome of many years of research on the subject and it is intended to serve as a reference for engineers and researchers but it should also be useful to (post-graduate) students as a supplementary text. It is expected that the reader is already familiar with (basic) aspects of internal combustion engine operation. Consequently, when dealing, for example, with the combustion development during transients, only a brief reminder is provided at the beginning of the section concerning some fundamental features of (steady-state) combustion, and afterwards we focus on the discrepancies and special behavior noticed during transients that diversify the operation from steady-state conditions. Wherever possible, we provide experimental results to support our analysis. There are a few points, however, where this was not feasible due to lack of relevant experimental work (*e.g.,* transient operation when the turbocharger compressor experiences surge or crankshaft transient torsional deformation).

At this point, we would like to express our thanks to the various publishers and companies, who have granted permission to reproduce figures and photos from their publications. In particular, the assistance of Messrs Jörg Albrecht of MAN Diesel SE, Elmar Gasse of Daimler AG, Chris Nickolaus of Cambustion Ltd, John Zambelis of Isuzu Motors Greece, Martin Stenbäck of Lysholm Technologies AB, Günther Krämer of BorgWarner Turbo Systems and Masayasu Kondo of Mitsubishi Heavy Industries Europe Ltd is greatly appreciated.

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Contents

Notation

a	Specific availability/exergy (J/kg)
A	Availability/exergy (J), or surface area (m^2)
b	Piston acceleration (m/s^2)
B	Bearing loading (N)
c_p	Specific heat under constant pressure (J/kg K)
c_v	Specific heat under constant volume (J/kg K)
d	Diameter (m)
D	Cylinder bore (m)
D_p	Particle diameter (m)
E	Energy (J)
f	Friction coefficient (Ns/m), or frequency (Hz)
F	Force (N)
g	Specific Gibbs free enthalpy (J/kg), or gravitational acceleration (9.81 m/s^2)
G	Mass moment of inertia (kg m^2), or Gibbs free enthalpy (J)
h	Specific enthalpy (J/kg), or heat transfer coefficient (W/m^2K), or oil film thickness (μm)
i	Gear ratio
I	Irreversibility (J), or light intensity (W/m^2)
k	Crankshaft stiffness coefficient (N m/rad), or thermal conductivity (W/m K), or extinction coefficient (m^{-1}), or spring stiffness (N/m)
K	Whitehouse–Way combustion model preparation rate constant, or dissociation constant
L, ℓ	Length (m)
m,M	Mass (kg)
$\dot{m}$	Mass flow-rate (kg/s)
n	Number
N	Engine speed (rpm)
O	Opacity (%)
p	Pressure (Pa or bar)

Q	Heat (J)
r	Pressure ratio, or crank radius (m)
R_{mol}	Universal gas constant (8,314 J/kmol K)
R_s	Specific gas constant (J/kg K), or swirl ratio
Re	Reynolds number
s	Specific entropy (J/kg K)
S	Piston stroke (m), or entropy (J/K)
t	Time (s)
T	Absolute temperature (K)
u	Specific internal energy (J/kg), or speed (m/s)
U	Internal energy (J)
V	Volume (m^3), or vehicle velocity (kph or mph)
V_h	Cylinder swept volume (m^3)
W	Work (J)
$\dot{W}$	Power (J/s)
x	Mole fraction, or displacement (m)
y	Governor control lever position (mm)
z	Fuel pump rack position (mm), or injector nozzle holes

Greek symbols

α	Thermal diffusivity (m^2/s)
β	Connecting rod angle (deg or rad)
γ	Ratio of specific heat capacities c_p/c_v
Δp	Pressure drop (Pa or bar)
ε	Angular acceleration (s^{-2}), or second-law efficiency (%), or aftercooler effectiveness (%)
η	Efficiency (%)
θ	Bearing loading angle (deg or rad), or grade angle (deg or rad)
λ	Relative air–fuel ratio, or ratio of crank radius to connecting rod length
μ	Chemical potential (J/kg), or dynamic viscosity (N s/m^2)
ρ	Density (kg/m^3)
σ	Crankshaft stress (N/m^2)
τ	Torque (N m), or time constant (s)
τ_{id}	Ignition delay (ms)
φ	Crank angle (deg)
Φ	Fuel–air equivalence ratio
ψ	Flow availability/exergy (J/kg)
ω	Angular velocity (s^{-1})

Subscripts

o	Reference conditions
2p	Two-pulse (governor)
a	Ambient conditions, aerodynamic

AC	Aftercooler
b	Boost
bl	Blow-by
br	Brake, or break-up
c	Coolant
C	Compressor
ch	Chemical
cl	Clearance
coupl	Coupling
cv	Control volume
cyl	Cylinder
d	Duration, or downstream
D	Demand, or damping
dyn	Dynamometer
e	Engine
em	Exhaust manifold
eq	Equivalent
f	Fuel, or frontal area
fb	Fuel burning
fi	Fuel injected
fl	Flywheel
fr	Friction
fw	Flyweight
g	Gas
gov	Governor
gr	Gravitational, or grade
i	Any species, or injected
im	Inlet manifold
in	Inertia
ind	Indicated
ins	Insulation
irr	Irreversibilities
is	Isentropic
kin	Kinetic
l	Reciprocating masses
L	Loss, or load
m	Mechanical, or molar
N	Nozzle
p	Particle, or periodic
r	Rotating masses
rod	Connecting rod
s	Start, or swirl
S	Stiffness
sc	Scavenge
se	Sensing element
st	Steady-state
sup	Supporting

rest	Restoring
T	Turbine
TC	Turbocharger
thr	Thrust
tot	Total
tr	Traction, or trapped
u	Upstream
v	Vapor
V	Vehicle
var	Various
vol	Volumetric
w	Wall, or work
wgv	Waste-gate valve

Superscripts

o	True dead state
ch	Chemical
tm	Thermomechanical
w	Water

Abbreviations

°CA	Degrees crank angle
A/C	Aftercooler or aftercooled
AFR	Air–fuel ratio
BDC	Bottom dead center
bmep	Brake mean effective pressure (bar)
bsfc	Brake specific fuel consumption (g/kWh)
BSN	Bosch smoke number
CAN	Controller area network
CARB	California Air Resources Board
CI	Compression ignition
CNG	Compressed natural gas
CO	Carbon monoxide
CPC	Condensation particle counter
CVS	Constant volume sampling
DI	Direct injection
DMA	Differential mobility analyzer
DMS	Differential mobility spectrometer
DOC	Diesel oxidation catalyst
DPF	Diesel particulate filter
ECU	Engine control unit
EGR	Exhaust gas recirculation
ELPI™	Electrical low-pressure impactor
EM	Electric motor

EMG	Electric motor-generator
EoI	End of injection
EPA	Environmental Protection Agency
ESC	European Steady-state Cycle
ETC	European Transient Cycle
EU	European Union
EUDC	Extra Urban Driving Cycle
EVC	Exhaust valve closing (°CA)
EVO	Exhaust valve opening (°CA)
FGT	Fixed geometry turbine
FID	Flame ionization detector
fmep	Friction mean effective pressure (bar)
GVWR	Gross vehicle weight ratio
HC	Hydrocarbons
HEV	Hybrid-electric vehicle
HP	High-pressure
HRR	Heat release rate
HSDI	High speed direct injection
I	Intercooler
ICE	Internal combustion engine
IDI	Indirect injection
imep	Indicated mean effective pressure (bar)
isfc	Indicated specific fuel consumption (g/kWh)
ISG	Integrated starter-generator
IVC	Intake valve closing (°CA)
IVO	Intake valve opening (°CA)
LEV	Low emission vehicle
LFG	Landfill gas
LHR	Low-heat rejection
LHV	Lower heating value (kJ/kg)
LII	Laser induced incandescence
LP	Low-pressure
mph	Miles per hour
NEDC	New European Driving Cycle
NMHC	Non-methane hydrocarbons
NRTC	Non-road Transient Cycle
NYCC	New York City Cycle
OBD	On board diagnostics
PAHs	Polycyclic aromatic hydrocarbons
PFSS	Partial flow sampling system
P-I-D	Proportional-integral-differential
PM	Particulate matter
PRA	Piston ring assembly
PSZ	Plasma spray zirconia
RME	Rapeseed methyl ester
RoHR	Rate of heat release
SCR	Selective catalytic reduction

SMD	Sauter mean diameter (μm)
SMPS™	Scanning mobility particle sizer
SN	Silicon nitride
SOC	State of charge (%)
SOF	Soluble organic fraction
SoI	Start of injection
TDC	Top dead center
TEOM®	Tapered element oscillating microbalance
THC	Total hydrocarbons
VGT	Variable geometry turbine
WHTC	World-wide Harmonized Transient Cycle

1

Transient Operation Fundamentals

1.1 Introduction

The most attractive feature of the compression ignition (diesel) engine is its excellent fuel efficiency, which can surpass 40% in vehicular applications and even 50% in large, two-stroke units of marine propulsion or electrical generation. Consequently, vehicles equipped with diesel engines achieve much lower specific fuel consumption and reduced carbon dioxide emissions than their similarly rated spark ignition counterparts over the entire operating range, resulting also in considerable money savings over the vehicle's lifetime. Moreover, diesel engines are characterized by low sensitivity in terms of air–fuel ratio variations, absence of throttling, high torque and high tolerability in peak cylinder pressures and temperatures that favors the application of various supercharging schemes.

Table 1.1. Characteristics of diesel engines for heavy trucks from 1930 to 1996 (reprinted with permission from Hoepke [1])

Maximum Power (kW)	70–110	107–132	140–154	283–304	309–441
Maximum Torque (Nm)	520–1010	620–1050	680–760	1285–1300	1850–2700
Displacement Volume (L)	8.5–16.6	8.3–11.6	9.7–12.7	11.5–18.3	12.1–18.3
Year	1930s	1955/6	1960	*ca.* 1985	1996 (Euro II)

For a number of years now, the vast majority of compression ignition engines is turbocharged and aftercooled; Table 1.1 shows the remarkable increase in engine power of heavy trucks during the last 70 years. Quadrupling of power output has been achieved without significant increase in displacement volume and with only modest increase in engine speed (mainly owing to improvements in the combustion process and fuel quality).

Instead, the main portion of the increase in engine output has been the result of super- or turbocharging, which has enabled more powerful yet smaller and more compact engines to be fitted to vehicles (downsizing). Turbocharging the diesel engine is beneficial because it not only increases its (specific) brake power, but also because it provides better fuel economy and reduced CO_2 emissions – due to leaner operation, increased mechanical efficiency and positive pumping work – and, in some circumstances, reduced exhaust gas and noise emissions. As a result, the turbocharged diesel engine is nowadays the most preferred prime mover in medium and medium-large unit applications, *i.e.,* truck driving, railroad locomotives, non-road mobile machinery, ship propulsion, electrical generation (Figure 1.1). Moreover, it continuously increases its share in the highly competitive automotive market. In fact, diesel-engined vehicles have already ensured a market share that is on a par with gasoline-engined ones across much of the European Union, they are beginning to gain considerable attention in the US and this trend is sure to continue.[1]

During recent decades, the increasingly stringent exhaust emission regulations have dominated the (automotive) industry, and forced manufacturers to new developments. For diesel engines, the emphasis is on reducing emissions of nitrogen oxides (NO_x) and particulate matter (PM), due to the toxicity of the inhaled nanoparticles and because these pollutants are typically higher than those from equivalent rated, port-injected gasoline engines equipped with three-way catalysts. Unfortunately, there is a trade-off between NO_x and PM reduction, resulting in a complicated control strategy requiring complex after-treatment systems. Sophisticated, high-pressure common rail injection systems, exhaust gas recirculation (EGR), or selective catalytic reduction (SCR), multi-valve configurations with variable valve timing, variable geometry turbochargers (VGT), exhaust after-treatment systems with particulate traps or urea-based deNO_x are among the measures applied for reduction of pollutant emissions and fuel consumption. Moreover, carbon dioxide (CO_2) emissions are becoming increasingly important owing to their connection with global warming; limiting CO_2 production can be achieved, primarily, through improvements in fuel economy and use of biofuels. Today's diesel-engined automobiles not only demonstrate greater fuel efficiency than ever before, but they also achieve emission levels at least 50% lower than those of a few years ago.

Unsurprisingly, the various technological advances mentioned above have also led to a significant increase in the complexity and cost of the engine and its control system, and this trend is sure to continue. For example, the Euro 5 details require improvements in PM emissions of 80% compared with Euro 4 (see also Appendix A, Table A.1), which will reflect in an increased cost of the order of €400 per (light-duty) vehicle or up to €3000 per heavy-duty vehicle.

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[1] According to the European Automobile Manufacturers' Association (ACEA), diesel-engined cars accounted for 53.3% of total new car registrations in the EU in 2007. By 2017, North America and Asia combined are predicted to account for 45% of global annual demand for diesel-engined light vehicles, compared with only 25% in 2007, according to JD Power and Associates, a global marketing information services firm.

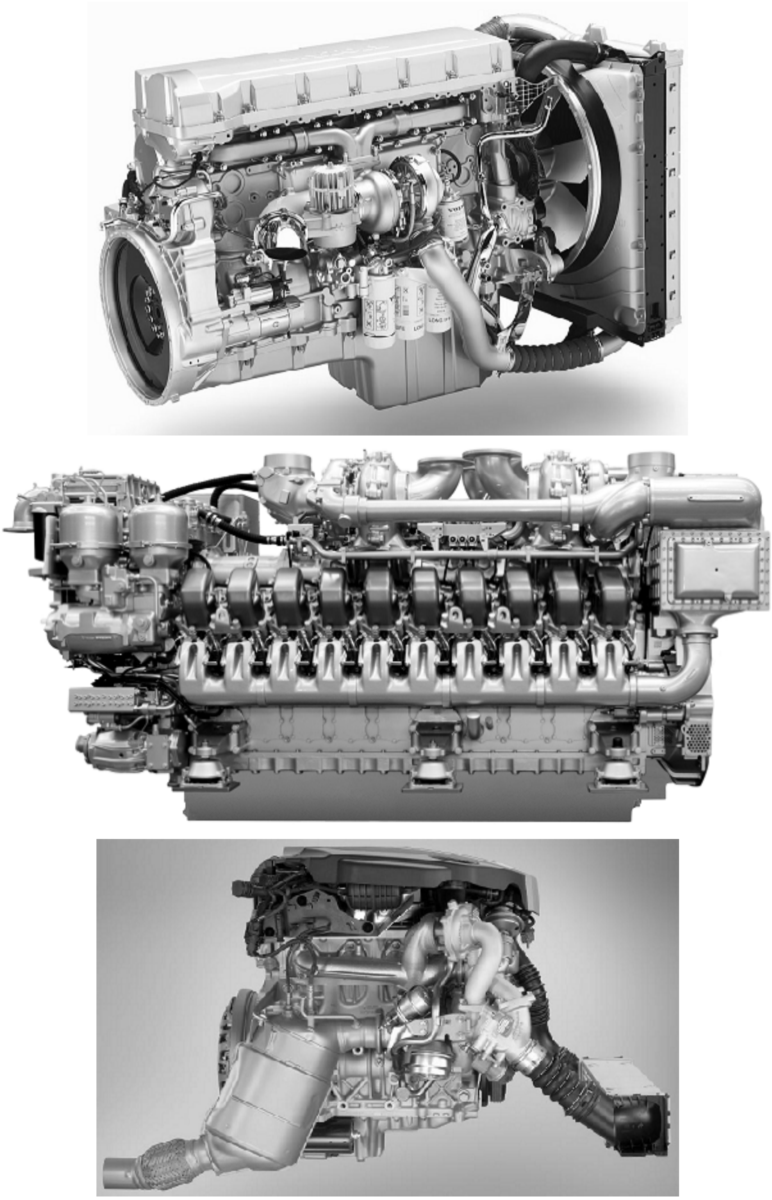

Figure 1.1. Modern, turbocharged, four-stroke, DI, diesel engines. *Upper:* Six-cylinder, variable geometry turbocharged and aftercooled, truck diesel engine of 16.1 L displacement volume, with electronic unit injectors and SCR for reduced NO_x emissions, producing up to 485 kW (image provided by Volvo Trucks). *Middle:* Sequentially turbocharged, and aftercooled, 20-cylinder, marine diesel engine MTU 20V 4000 with four turbochargers, producing 4.3 MW (Copyright Tognum AG). *Lower:* Four-cylinder, two-stage turbocharged and aftercooled, passenger vehicle diesel engine of 2.0 L displacement volume with common rail injection system (2,000 bar), EGR and diesel particulate filter (courtesy of BMW AG)

The level of particulates are set at 5 mg/km for passenger vehicles and 0.03 g/kWh for heavy-duty engines, which effectively means that a DPF (diesel particulate filter) would be mandatory downstream of the oxidation catalyst; more injections will be required during a cycle, especially post-injections in order to help regeneration of the DPFs. Similar improvement will be required from Euro V to Euro VI level, since for heavy-duty engines a reduction of the order of 66% for both NO_x and PM emissions will most probably be legislated (equally strong are the improvements required by the US EPA and the Japan Ministry of Environment). More than in the past, combination of both internal engine measures and efficient exhaust after-treatment devices will be required. Since the number of engine and turbocharger controllable components rises continuously (see also Figure 1.3 later in the section), complicated and sophisticated control algorithms are needed to achieve optimum performance. It is universally accepted today that great benefits in the analysis and control of internal combustion engines will (and have actually) come from optimizing the interaction of all the associated engine sub-systems by detailed *simulation models* of the engine processes (Chapter 9). The latter will also pave the way for more advanced engine concepts and alternative combustion systems, such as homogeneous charge compression ignition (HCCI) and low-temperature combustion (LTC).

Traditionally, the study of internal combustion engines operation has focused on the steady-state performance, with minor, if any, attention paid to the unsteady-state or more accurately termed *transient* operation. However, the majority of daily driving schedule involves transient conditions. In fact, only a very small portion of a vehicle's operating pattern is true steady-state, *e.g.,* when cruising on a motorway. Historically, however, the research on transient diesel engine operation was initiated from the observation by engine manufacturers in the 1960s that when highly-rated, medium-speed diesel engines are employed in sudden 0–100% step load changes, severe difficulties are encountered, even leading to engine stall. In recent years, it is the global concern about environmental pollution that has intensified the respective studies; particulate, gaseous and noise emissions typically go way beyond their acceptable values following the extreme, non-linear and non-steady-state conditions experienced during dynamic engine operation. A few representative results follow: cold- or warm-start emissions from heavy-duty diesel engines have been found to exceed up to 15 times their steady-state values; 50% of NO_x emissions from automotive engines during the European Driving Cycle stem from periods of acceleration, whereas instantaneous particulate matter and NO_x emissions during load increase transients have been measured to be 1 to 2 orders of magnitude higher than their respective quasi-steady values (Figure 1.2 [2–5]).

The latter remark highlights in the most explicit way the considerably differentiated evolution pattern of transient response from the respective steady-state operation, and it will be discussed in detail in Chapters 2 and 5. Acknowledging the above-mentioned findings, various legislative Directives in the European Union, Japan and the USA, have drawn the attention of manufacturers and researchers all over the world to the transient operation of (diesel) engines, in the form of Transient Cycles Certification for new vehicles [6, 7].

Moreover, transient operation is inherently characterized by quick changes in the operating conditions, which can prove particularly demanding in terms of engine response and in the reliability of fuel pumps and governors/controllers, so that proper interconnection is required between engine, governor, fuel pump, turbocharger and load through an appropriate control strategy. The latter is required in order to avoid over- or under-speeding, overshoot of exhaust emissions or (significant) departure from the acceptable fuel consumption levels.

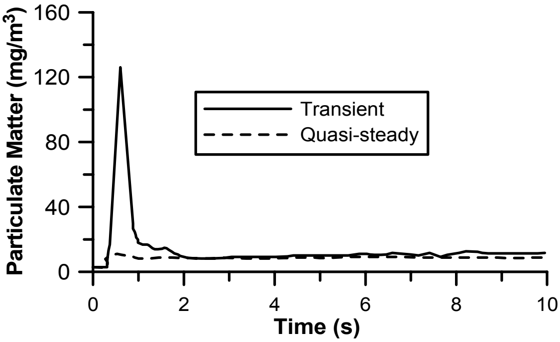

Figure 1.2. Transient *vs*. steady-state PM emissions during a typical load increase event of a turbocharged diesel engine (reprinted with permission from SAE Paper No. 2006-01-1151 [4], © 2006 SAE International)

Traditionally, the control optimization is undertaken during the design stage for steady-state operation, with the calibrated parameters, *e.g.,* injection strategy (pre-, main- and post-injection scheme, rate, timing and pressure of injection), VGT vanes position, boost pressure, EGR valve position *etc.,* stored in 3-D maps with respect to engine rotational speed and load/fueling. The main objective of the diesel engine control system is then to provide the required torque with minimum fuel consumption and exhaust emissions. During transients, the behavior of the engine control system depends upon the specific application (vehicle, ship propulsion, locomotive traction, electrical generation, *etc.*), which imposes different requirements, compared with steady-state operation, with 'driveability'[2] issues playing now a primary role. For example, in electrical generation applications, zero speed droop is required together with short recovery period after the new load has been applied for the base units, as well as rapid start-up for the stand-by ones, whereas in vehicles, fast acceleration is pursued combined with limited smoke emissions. In locomotive applications, the major challenge is to ensure that the electrical system and the diesel engine both function as a synchronized unit and undesirable phenomena such as under-speeding or excessive smoke emissions are avoided.

The comprehensive open- or closed-loop control systems employed gather the signals from the various sensors located on the engine, fuel pump and turbocharger (Figure 1.3), process them based on look-up tables (steady-state maps) or, better

[2] Thereinafter meaning rapid engine torque and speed response during acceleration of vehicular engines, as well as avoidance of over- or under-speeding (stall) and successful load acceptance with small speed droop and short recovery period for industrial/marine engines.

still, model-based control theory with, for example, feed-forward control, and eventually determine, via actuators, the optimum position of the various valves, vanes, *etc*. In order to realize such meaningful analysis of engine performance under transients, accurate analysis of the relevant experimental data is also needed that can only be accomplished using complicated and expensive test facilities equipped with electronically controlled dynamometers and accurate and fast response exhaust emission analyzers (Chapter 4).

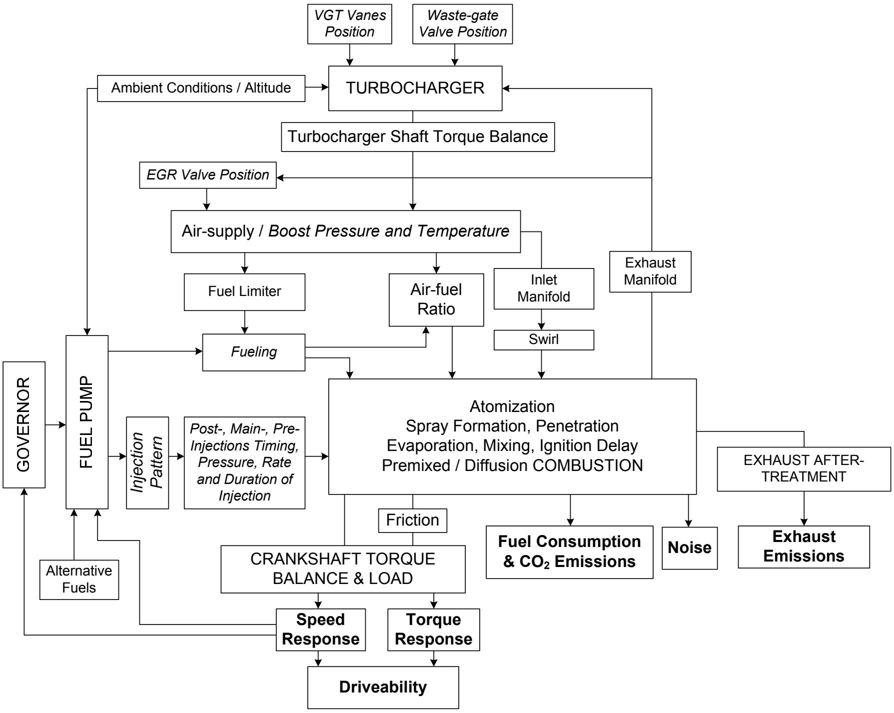

Figure 1.3. Simplified diagram showing some major air-supply and fueling controllable inputs (italics) and engine/vehicle outputs (bold), highlighting the complexity of a modern diesel engine powertrain

The fundamental aspect of transient condition lies in its operating discrepancies compared with steady-state operation (*i.e.,* operation at the same engine speed and fuel pump rack position of the respective transient cycles). Wheareas during (fully warmed-up) steady-state conditions, crankshaft rotational speed and fueling, hence all the other engine and turbocharger properties remain practically constant, during transient operation, both the engine speed and the amount of injected fuel change continuously. Consequently, the available exhaust gas energy varies, affecting turbine enthaply drop and, through the turbocharger shaft torque balance, the air-supply and boost pressure to the engine cylinders are influenced (Section 2.1). However, due to various dynamic, thermal and fluid delays in the system, mainly originating in the turbocharger principle of operation and its dymanic moment of inertia, combustion air-supply is delayed compared with fueling, eventually affecting torque build-up (driveability) and exhaust emissions. The most notable feature of transient turbocharged diesel engine operation that determines its whole operating profile is the *turbocharger lag*. Turbocharger lag is usually realized as slow response rate after a load or speed increase transient event as well as in the form of black smoke emissions coming out of the exhausts of diesel-engined vehicles. Unfortunately, there is a trade-off between exhaust smoke and engine response since the fuel–air ratio at which smoke becomes a problem is substantially lower than that at which maximum torque would be produced by the engine. During the early cycles of a transient event and until the air-supply and engine torque have been built-up efficiently, the following phenomena may be noticed that drastically differentiate the operation from steady-state conditions

- instantaneous torsional deformations in the driving system of the mechanical fuel injection pump leading to incomplete combustion;
- retardation of the dynamic injection timing;
- excursion of the fuel–air equivalence ratio to higher than stoichiometric values;
- increase in the average diameter of the fuel droplets leading to increased jet penetration;
- influence in mixture formation resulting in poorer mixing;
- increased ignition delay and hard combustion course;
- deflections of the crankshaft due to great accelerations/decelerations, resulting in an increase of mechanical friction;
- sharp thermal and mechanical gradients in the cylinder walls and valves;
- turbocharger compressor surging;
- increased (combustion) noise radiation;
- increased particulate and gaseous exhaust emissions; and
- engine stall, when an extreme load increase is abruptly applied.

The arguments introduced previously highlight, on the one hand, the importance of transient operation, and on the other hand, its complexity and its completely different development pattern compared with steady-state operation, requiring careful and systematic experimental and simulation analysis. Clearly, transient turbocharged operation cannot be considered a series of steady-state operating points, nor can the engine be assumed to behave in a quasi-steady manner during transients.

1.2 Typical Transient Operation Cases

There is a variety of operating conditions experienced by (diesel) engines that can be classified as ‘transient’; these may last from a few seconds up to several minutes. In this book, the term *transient* will be used to describe any of the following three *forced* changes in load and/or fueling:

1. load acceptance (change) at constant governor setting, mainly experienced in industrial applications, *e.g.,* electrical generation, but

also observed in various propulsion applications, *e.g.,* when a vehicle climbs a hill;
2. change in the fuel pump rack position or step change in pedal ('throttle') position resulting in speed changes (acceleration) of vehicular, locomotive or marine engines;
3. cold or hot starting.

These are the most fundamental transient cases,[3] from which the first two will be primarily addressed in this book. Combination of the above three transient schedules results in

4. simultaneous speed and load changes;
5. whole vehicle/ship propulsion schedule, *e.g.,* gear shift or full ahead/full astern pattern;
6. Transient Cycles, which consist of all of the above mentioned transients.

A number of typical transient cases will be described in the following sections, emphasizing the different development pattern of the engine properties as well as the requirements of the engine control system. At the same time, the mechanism of turbocharger lag and its implications for the engine response will be introduced before the more in-depth analysis of Chapter 2.

1.2.1 Load Increase (Acceptance) Transient Event

A detailed set of engine and turbocharger properties responses during a typical load increase transient event initiated from a step load change from 10 to 85% of full engine load, at constant governor setting, is illustrated in Figure 1.4; it concerns a four-stroke, six-cylinder, moderately turbocharged and aftercooled, medium-high speed, industrial diesel engine, with mechanical fuel pump-governor, rated at 236 kW at 1500 rpm. Such load increase transients are typical of industrial applications, *e.g.,* electrical generation (zero final speed droop is required here as will be shown later in the section), pump driving, *etc.,* but they are also experienced in marine and non-road mobile engines (*e.g.,* agricultural or excavator). On the other hand, although automotive engines do encounter load acceptance transients, *e.g.,* when a vehicle climbs a hill, or when engaging the clutch after a gear change, these are not so pronounced as the ones presented in this section. Eleven typical engine and turbocharger variables are depicted in Figure 1.4 with respect to the engine cycles or time, namely, engine speed, crankshaft angular acceleration, load torque, fuel pump rack position, peak cylinder pressure, air–fuel equivalence ratio, soot density and NO concentration, compressor boost pressure, turbocharger speed and turbine inlet temperature.

At the initial conditions, the engine and load (resistance) torques are equal and the air–fuel ratio is relatively high due to the low loading. As soon as the new

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[3] Engines or vehicles encounter several other operating conditions that can loosely be termed 'transient', such as cyclic variation, warm-up, particulate trap regeneration, *etc*. These are not considered in this book.

higher load is applied (this is accomplished in 1.3 s), there is a significant deficit in the net (engine minus load) torque, since the engine torque cannot instantly match its increased load counterpart, hence the engine speed drops. This is sensed by the governor sensing element, which, in turn, shifts the fuel pump rack towards a position of increased fueling. At the same time, the air–fuel ratio decreases since the air-supply cannot instantly match the higher fueling owing to the delayed response of the turbocharger in building-up the required delivery pressure. The instantaneous relative air–fuel ratio, therefore, may reach even lower than stoichiometric values leading to intolerable smoke emissions. At the same time, the increased gas temperatures owing to the low air–fuel ratios are also reflected into high NO_x emissions, with the amount of available oxygen playing here a significant role.

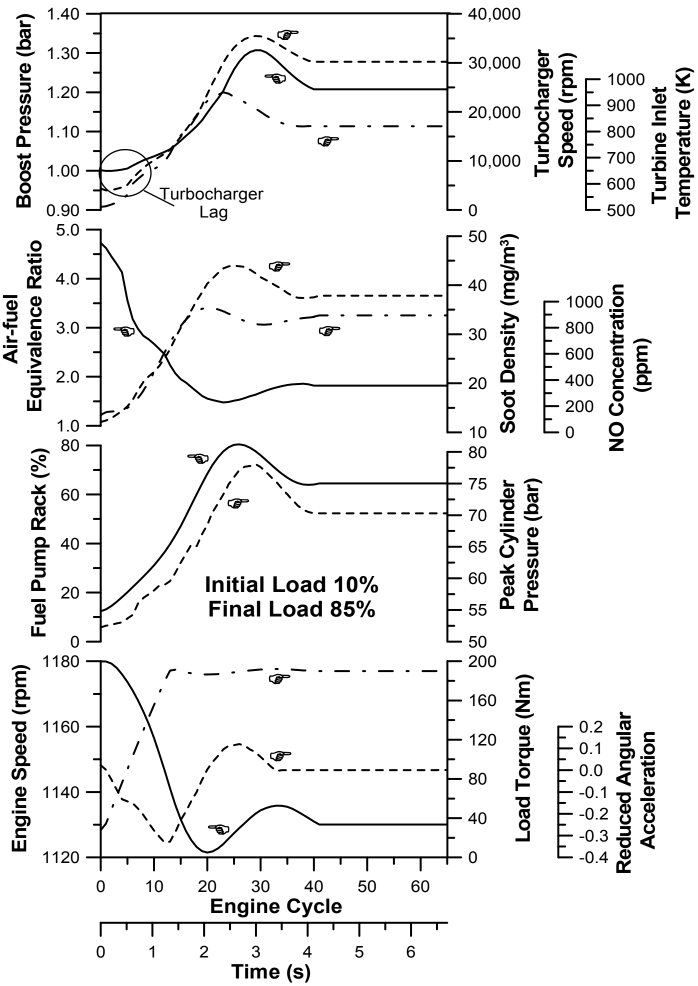

Figure 1.4. Development of engine and turbocharger properties during a load increase transient event of a four-stroke, moderately turbocharged and aftercooled diesel engine

Compressor boost pressure and turbocharger speed develop in a similar way, with the turbocharger lag being obvious during the first cycles of the transient event. Clearly, the increased exhaust gas power is not capable of instantly increasing the turbine power output, largely owing to the turbocharger inertia, so that the compressor operating point moves slowly towards the direction of increased boost pressure and air-mass flow-rate; during this period, the engine is practically running in naturally aspirated mode. During the early cycles (where the torque deficit is greatest), the highest value of crankshaft deceleration is experienced. The lowest engine speed or the maximum temporary speed droop (defined in Figure 1.5[4]), is observed for the present transient event, twenty cycles after the application of the load change. Because of the type of load involved (quadratic, as the particular engine is coupled to a hydraulic dynamometer), the drop in engine speed causes also a drop in resistance torque, resulting in quicker achievement of the final equilibrium between engine and load.

Final equilibrium, also termed recovery period, is defined in Figure 1.5 too; it concerns the time needed to reach the final (steady-state) engine speed. For the transient case demonstrated in Figure 1.4, this is achieved after almost 4 s (recall that the load application lasted for 1.3 s) or 40 cycles with a rather small final speed droop of the order of 47 rpm or 4% of the initial engine speed. For load changes of vehicular engines, much larger speed deviations are experienced since the speed governor has different, less tight characteristics (Section 3.2). The delays involved in the engine and turbocharger response build-up are best highlighted by the observed phase shift between the minima or maxima of the various parameters values during the transient test.

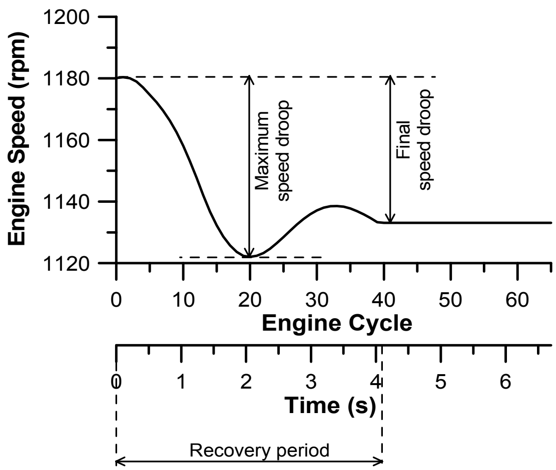

Figure 1.5. Definition of instantaneous maximum and final speed droop and recovery period for a load acceptance transient event

[4] In the literature, strict definition states that *droop* is the change in speed between no-load and full-load, while run-up is defined as the change in speed between full-load and no-load. Nonetheless, the majority of researchers in the field have used the term droop for intermediate load changes too and this approach will be followed in this book.

The engine of Figure 1.4 is fitted with a non-zero speed droop governor, a fact that led to positive final speed droop. What is more important is that it also possesses a high mass moment of inertia, which slowed down the development of the transient event and produced rather limited smoke emissions and crankshaft angular deceleration. The successful load acceptance was further supported by the speed-dependent resistance torque. The rather moderate rating of the engine (15 bar bmep) did not lead to complete deterioration of combustion during the early cycles as the turbocharger lag effects were rather limited. The particular engine makes use of the pulse turbocharging system with small exhaust pipes, and a twin entry turbine (each entry 'accepts' the exhaust gases from three cylinders, with minimum, if any, exhaust pressure waves overlap), a fact aiding fast response in contrast to constant pressure turbocharging. In the latter, usually met in large two-stroke (marine) engines, the large volume of the manifolds requires more time to be filled with the higher density gas, thus, delaying engine response even more (see also Figure 1.17 later in the chapter). Moreover, the imposed load change was not a very difficult task for the engine to cope with, as the final load was lower than the maximum acceptable and it was not applied instantly. For greater or more abrupt load changes engine stall might occur, especially if the engine is fitted with a fuel limiter.

This is actually the case illustrated in Figure 1.6 for a similarly rated, industrial, turbocharged diesel engine; here the load change was 0–100%. Since the fuel quantity was not allowed to exceed temporarily its steady-state full-load value, the engine was unable to handle the severe load increase owing to turbocharger response delay, and ultimately stalled. Usually, a fueling overload of about 10% (above steady-state values) is allowed in order for a turbocharged diesel engine to be able to withstand step load changes of the order of 0–100%. In vehicular applications, on the other hand, it is highly unlikely that the engine could stall because of a load increase, since load increases are much softer compared with industial engines. Instead, the problematic transient response would rather manifest itself as increased smoke emissions coming out of the exhaust pipe, owing to the mismatch between air-flow and fueling during the early cycles of the transient event.

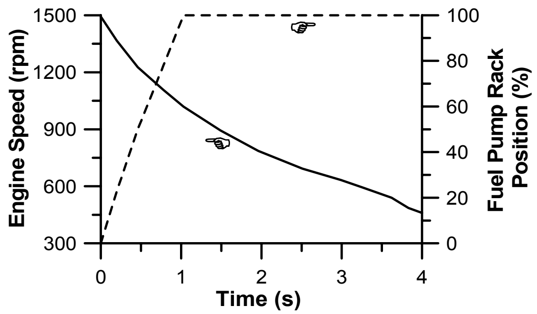

Figure 1.6. 0–100% step load increase transient event leading to engine stall

Summarizing the above remarks, the flow-chart of Figure 1.7 illustrates what happens in a turbocharged diesel engine after a new (higher) load has been applied.

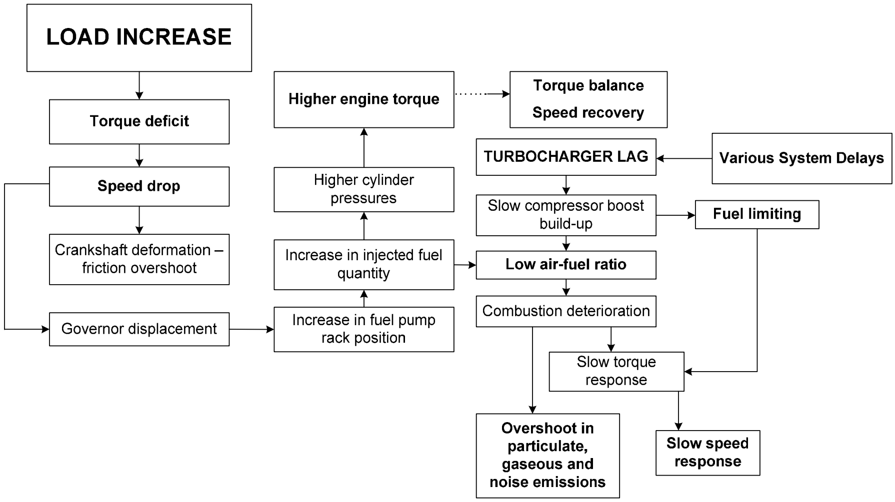

Figure 1.7. Series of events after a step load increase of a turbocharged diesel engine

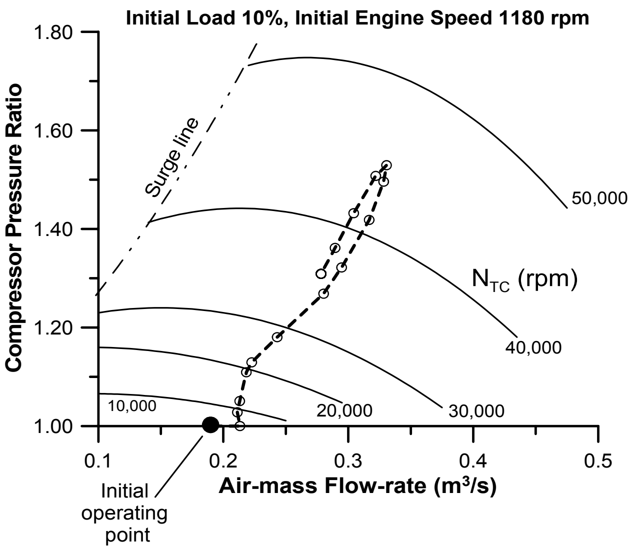

Figure 1.8. Development of engine transient response on the turbocharger compressor map

An alternative way of presenting a turbocharged diesel engine's transient response is by depicting it on the turbocharger compressor map, as is illustrated in Figure 1.8 for a 10–95% load increase transient event. This technique has the major disadvantage of ignoring the time scale involved; yet, it can prove quite useful when studying engine–compressor interdependence such as, for example, when the turbocharger compressor experiences surge or when the subject of concern is the matching between engine and turbocharger. Turbocharger lag is