

Thermodynamic Aspects of Transient Operation

The primary purpose of supercharging, nowadays universally employed in diesel engines, is the boost in power output without increase of speed. This is accomplished by increasing the engine air-supply (density) through compression in an external compressor, thereby facilitating increase in fueling. By far, the most popular and successful version of supercharging is the exhaust gas turbocharging. Here, the power required to drive the compressor is provided by a turbine that utilizes (part of) the available exhaust gas energy leaving the cylinders. When designing a turbocharged diesel engine, greatest care has to be given to the matching between engine and turbocharger and between fuel injection and air-supply/motion. These two interrelated optimizations are usually accomplished for steady-state operation but they are really put to the test during transients, where both the engine and the turbocharger are called upon to operate far from their design point conditions.

For a four-stroke engine, air-supply is primarily determined by rotational speed, delivery pressure and pressure difference between inlet and exhaust manifold during the valve overlap period; each of these three factors affects the transient breathing pattern of the engine too. Unfortunately, the flow characteristics of turbochargers are fundamentally different from those of reciprocating engines, a fact that leads to complex matching issues and problematic transient response. The performance of turbo-machines is very much dependent on the respective gas angles and it deteriorates when moving away from their design point. However, their use as superchargers is widely accepted due to their simplicity, small size and large flow capacity that enables significant increase in engine output, increase in the mechanical efficiency¹ but also due to their ability to operate under very high

¹ The fuel economy of a diesel engine is usually improved by turbocharging, since the mechanical losses do not increase in proportion to the increase in power output; this is not the case, however, for spark ignition engines, since turbocharging necessitates a reduction in compression ratio (hence indicated efficiency) in order to avoid knock.

mass flow-rates. As will be documented in the analysis of the next section, the key factors that primarily determine the transient torque and speed response of a turbocharged diesel engine is the engine air-supply and the EGR management (not fueling), being strongly influenced by the transient operation of the turbocharger; the latter affects unfavorably the whole system behavior and it is responsible for an overshoot in particulate, gaseous and noise emissions.

2.1 Turbocharger Lag and Transient Torque Pattern

The most notable feature of turbocharged diesel engine transient response that differentiates substantially the operation from steady-state conditions is the *turbocharger lag* or turbo-lag. Actually, there are various delays that affect the engine transient response, hence it may be more appropriate to speak of the whole system lag rather than only of the turbocharger; nonetheless, it is the latter's flow and dynamic inertia that contributes mostly to the total response delay. Turbocharger lag is caused because although the fuel pump responds rapidly to the increased fueling demand after a load or speed increase transient event, the compressor air-supply cannot match this higher fuel flow instantly but only after a number of engine cycles owing to the mechanical, thermal and fluid inertia of the whole (engine, manifolds and turbocharger) system. Consequently, without a properly matched transient fuel delivery on a *cycle-by-cycle basis*, the low trapped air-mass will initially lead to overshoot of the global fuel–air equivalence ratio. Turbocharger lag is pronounced with the continuous increase in engine rating; it affects primarily the engine torque pattern being realized as slow response after a load acceptance or acceleration as well as in the form of black smoke emissions coming out of the exhaust pipes 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.

After a ramp increase in engine load or fueling (see for example Figures 1.4 and 1.14), the governor forces an increase in the injected fuel quantity that is ultimately transformed into a boost of the exhaust gas energy. However, power production delays deteriorate this conversion; these are located in the fuel pump mechanism (Section 2.2) or in the in-cylinder processes (Section 2.3). For the former, mismatch in the (mechanical) fuel injection system is the key cause for the respective delay.² For the latter, the most notable delays emanate from the prolonged ignition delay period and the heat transfer to the cylinder and exhaust manifold walls; in fact, cylinder wall temperature adjusts to the new fueling conditions with a substantial delay, during which increased heat flux is observed from the gas to the walls. Although the detailed heat transfer mechanism during transients will be discussed in Section 2.3.1, it is important at this point to emphasize that this wall temperature adjustment lasts one order of magnitude

² A special behavior of the fuel injection system during transients, namely fuel limit is a consequence rather than a cause of the engine transient delay. In any case, fuel limit action further adds to the slow engine torque and speed build-up as will be discussed in Section 2.2.

longer than the respective transient event.³ For the same reason, higher frictional losses are observed, when comparing transient to quasi-steady cycles, since the oil temperature corresponds to its initial steady-state value during the first cycles of the transient owing to the lubricating oil's thermal inertia (low oil temperature increases oil viscosity, hence friction losses).

Other related power production delays concern the non-immediate change in bmep after the rack position has been altered, but only at the following injection for *each* cylinder, and the acceleration of the slider-crank mechanism rotating masses. The delay of the slider-crank mechanism kinematics observed after initiation of combustion is of the order of 90°CA (see Figure 2.1). Moreover, the increased gas enthalpy must 'wait' until the exhaust valve opens in order to move towards the exhaust manifold and turbine inlet. This further delays the energy effect on the turbine so that, in total, there is more than 200°CA delay between the first injection after a load or speed increase and the possible gains in turbine inlet pressure [1].

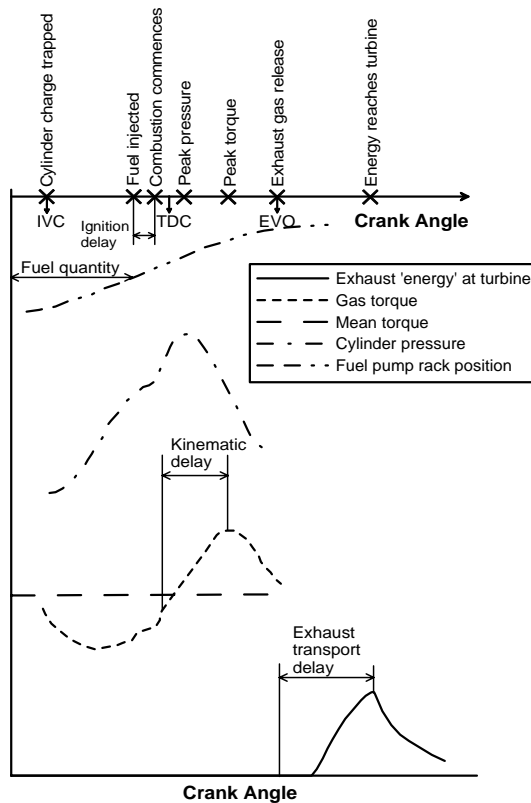


Figure 2.1. Delays associated with diesel engine power production (from Winterbone [1], reprinted by permission of Oxford University Press)

³ Interestingly, there is a positive effect of the slow cylinder wall temperature adjustment since low wall temperatures favor higher volumetric efficiency of the incoming air-charge.

Further to the rather limited power production system delays, the energy transfer to the turbocharger is delayed by the fact that the nature of this transfer is via a continuous flow of relatively low density gas, whose temperature is reduced due to heat transfer to the ‘colder’ exhaust manifold walls. Consequently, only a portion of the increased exhaust gas power produced in the cylinder ultimately reaches the turbine. The above-mentioned process is strongly influenced by the exhaust gas turbocharging scheme involved, *i.e.*, pulse or constant pressure.

With constant pressure turbocharging (generally preferred for high engine power output and at medium to high load operation, see also Figure 6.31), the exhaust ports from all cylinders are connected to a single exhaust manifold of sufficiently large volume to smooth out the unsteady flow exiting from each cylinder. Although turbine efficiency is usually higher and construction simpler with one turbocharger unit, even for engines with a large number of cylinders, transient delay is more pronounced when a constant pressure turbocharging configuration is used (*e.g.*, in large two-stroke engines, as was the case depicted in Figure 1.17). The primary mechanism behind this behavior is the considerably longer time required to boost the pressure of the exhaust manifold, since its large plenum increases substantially its flow inertia, while the energy transfer is accomplished by mass flow only without any pressure wave influence. In other words, the high kinetic energy of the exhaust gases leaving the cylinder is not utilized. This delayed response of the exhaust manifold (Figure 2.2) may even lead to engine stall when a large load increase is applied, as is, for example, the case when maneuvering from full ahead to full astern. Nonetheless, the slower build-up of exhaust manifold pressure noticed in constant pressure marine (or generator) diesel engines does not constitute that much of a disadvantage, as these engines experience much fewer transients, mainly when maneuvering in port. During their propulsion in the open sea, which comprises by far the greatest part of their operating life, steady-state operation at high engine load is practically established. In any case, and owing to the vessel’s large weight, engine response is always faster than the respective ship’s one.

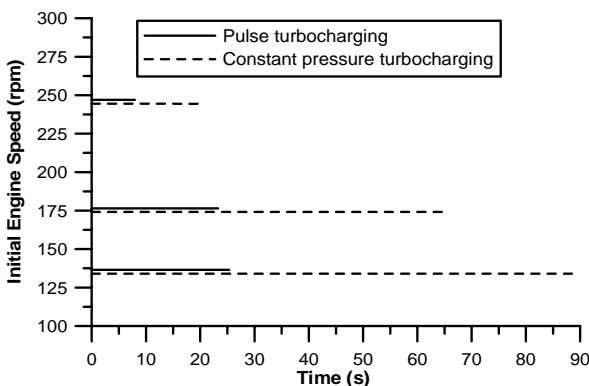


Figure 2.2. Comparison in the recovery period when accelerating from initial speed to full speed between pulse and constant pressure turbocharging (four-stroke, medium-speed diesel engine rated at 4,000 kW and 16 bar bmep at 400 rpm), (from Zinner [2], reprinted with kind permission from Springer Science+Business Media)

On the contrary, vehicular (diesel) engines make use of the pulse turbocharging scheme (patented by Alfred Büchi in 1925) although at the expense of lower turbine isentropic efficiency (recall that turbo-machines are better operated as steady flow rather than pulse wave devices). In general, manifold volume affects the amplitude of the pressure waves generated during the blow-down period of the exhaust process. The pressure waves travel at sonic velocity and their travel time in the exhaust manifold pipe is given by the following equation

$$\Delta\phi = \frac{12LN}{\alpha} \quad (2.1)$$

where α is the sonic velocity, L the equivalent pipe length and N the engine speed. The wave amplitude is governed by the difference between instantaneous gas flow-rates in (from cylinders) and out (through cylinders), and manifold volume. Thus, exhaust manifold volume influences turbine expansion ratio and, hence, turbine power. Pulse turbocharging aids faster response of the turbine since the exhaust manifold volume is minimized, being divided into pipes of small cross section area, and the gas flow in the form of waves reaches the turbine entry immediately after the exhaust blow-down commences. Hence, in pulse turbocharging schemes, the exhaust gas energy that reaches the turbine is higher than with constant pressure turbocharging but the unsteadiness of the process is responsible for its rather inefficient exploitation.

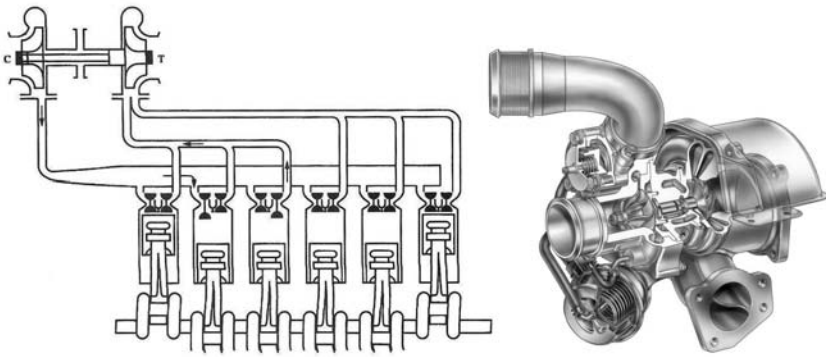


Figure 2.3. *Left:* Schematic diagram of a six-cylinder engine with pulse turbocharging configuration and twin-entry turbine. *Right:* Automotive turbocharger with twin-scroll turbine (courtesy of General Motors Corp)

In a pulse turbocharging configuration, use is made of turbines with multiple entries or scrolls, carefully chosen to serve specific cylinders in order to avoid interference between the traveling exhaust waves from the individual cylinders. Since the exhaust process duration in a four-stroke engine is of the order of 240°CA , it follows that not more than three cylinders should be connected to the same turbine entry. For example, cylinders No. 1, 2 and 3 on the one hand, and

cylinders No. 4, 5 and 6 on the other, are served by different turbine entries in a four-stroke, six-cylinder engine having an ignition order of 1-5-3-6-2-4 and equal intervals of the order of 120°CA between successive firing cylinders (Figure 2.3).

Despite its advantages, pulse turbocharging is not universally applied; one such case is when a large number of engine cylinders, hence, turbine entries and manifold sections is involved. When the engine has an odd number of cylinders, pulse turbocharging is not a very attractive solution since, in this case, uneven exhaust pulses are generated in each section. This subject has been treated in detail by Zinner [2] and Watson and Janota [3], who also provide useful examples regarding cylinder number and disposition effects, mainly for steady-state operation.

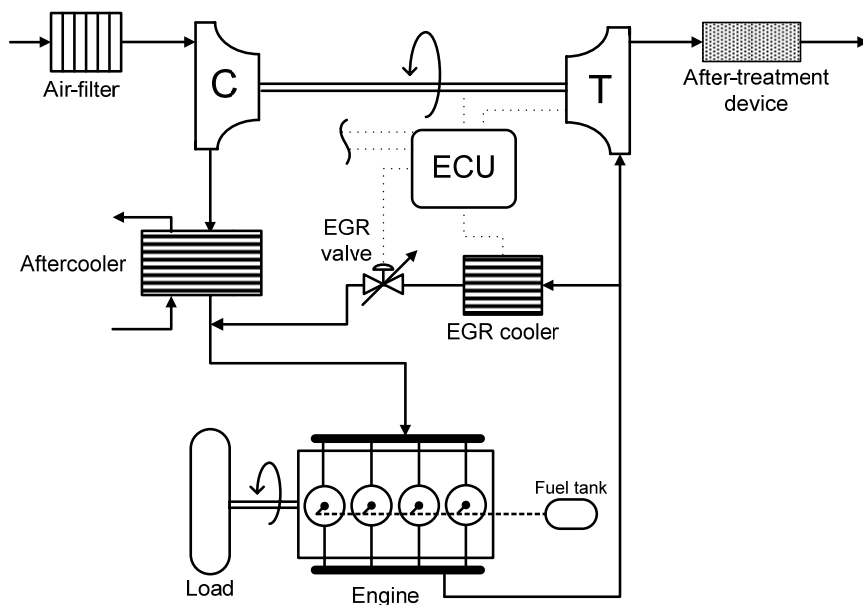


Figure 2.4. Schematic arrangement of current state of the art turbocharged diesel engine (C: compressor, T: turbine, ECU: engine control unit, EGR: exhaust gas recirculation; single line (—) indicates gas flow, double line (≡) indicates mechanical connection, light dotted line (...) indicates signal and heavy dotted line indicates fuel flow)

By the time the exhaust gas reaches the turbine rotor, there are further, considerable delays induced by the turbocharger operation. In order to understand these delays, the basic operating pattern of the turbocharger needs first to be addressed. The main characteristic of a turbocharged engine is the lack of mechanical connection between turbocharger and engine crankshaft (Figure 2.4). Consequently, turbocharger and engine speeds are not mechanically but only indirectly interrelated. Turbocharger compressor boost pressure and air-mass flow-rate are typically interconnected via maps such as the one illustrated in Figure 2.5, showing also contours of constant isentropic efficiency and rotational speed.

The map of a radial turbocharger compressor (Figure 2.5) is characterized by:

- constant speed lines of variable slope; for radial compressors (typical in internal combustion engines applications) these are practically horizontal close to the surge line, but drop off rapidly;
- flow chokes at the right of the map when it reaches the sound velocity, typically at the inducer or at the entry to a vaneless diffuser or at the throat of a vaned diffuser;
- the pressure ratio is strongly dependent on rotational speed;
- there is an unstable region to the left of the surge line, in which operation is not viable (see also Section 7.2); and
- at high boost pressures, the speed range is usually narrow (a fact, which as will be discussed in Section 6.4.3, is related to application of two-stage units) and the isentropic efficiency low; both arguments holds true for turbocharger turbines too.

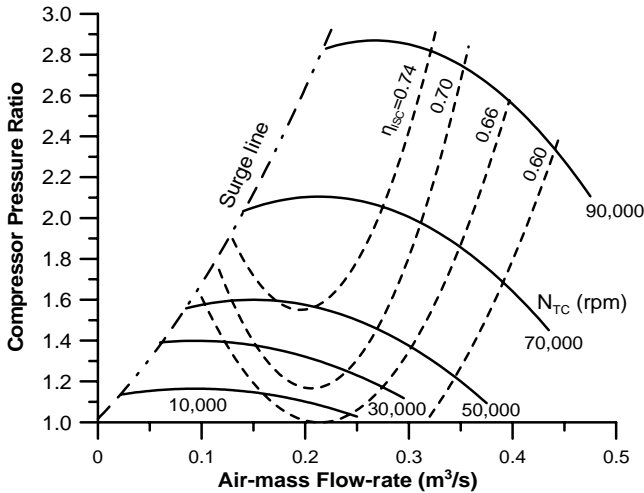


Figure 2.5. Typical map of aerodynamic type turbocharger compressor

It is made obvious then, that it is impossible for a radial compressor to achieve both high boost pressure and air-mass flow-rate when operating at low rotational speed. Instead, acceleration of the turbocharger to a higher speed is required, which, unfortunately, is delayed considerably by the turbocharger's mass moment of inertia. At steady-state conditions, the turbine power is equal to the sum of the compressor power and the losses in the turbocharger shaft bearings, so both compressor and turbine are in an equilibrium state. If the engine operating condition is changed after a ramp increase in load or fueling, the turbocharger needs to accelerate to a new operating point in order to provide the required higher boost. To achieve that, the turbine power must exceed the compressor power. For the turbocharger acceleration, the following differential equation describes the

transient angular momentum balance on its shaft, based on Newton's second law of motion for rotational systems

$$\eta_{mTC}\tau_T - |\tau_C| = G_{TC} \frac{d\omega_{TC}}{dt} \quad (2.2)$$

where τ_C and τ_T are the instantaneous values for the compressor and turbine indicated torque, respectively, and η_{mTC} is the turbocharger mechanical efficiency. Changes in turbocharger rotating inertia G_{TC} have been found to have the strongest influence on its transient response, as will be described in Section 6.4.1. In fact, the inertia of the turbocharger disk is closely related to the fifth power of diameter, $G_{TC} \propto d_T^5$. This suggests that reducing turbine (and compressor) rotor diameter d_T will powerfully influence turbocharger and, consequently, engine speed response [4]; the latter remark emphasizes also the favorable use of more than one small turbocharger instead of a single (larger) unit.

At this point, it would be helpful to discuss briefly the merits of mechanical supercharging (Figure 2.6) with respect to transient delays. Contrary to the aerodynamic type compressors used in turbocharging systems, mechanical supercharging makes use of positive displacement compressors. The latter may be of piston type, usually installed in series with the exhaust gas turbocharger (this arrangement is met in large, two-stroke engines) or rotary type, *i.e.*, screw or Roots blower, common in smaller applications such as spark ignition engined passenger cars (lower sub-diagrams of Figure 2.6). Positive displacement compressors are characterized by relatively lower rotational speeds compared with exhaust gas turbochargers, whereas their flow characteristics resemble those of the reciprocating engine. In a mechanical supercharging configuration, a mechanical connection exists between the compressor and engine crankshaft that ultimately minimizes transient lag (at least during acceleration transients) ensuring much faster engine response. However, this is accomplished at the expense of higher system complexity and lower engine brake efficiency due to the mechanical losses associated with driving the compressor.

The key characteristics of positive displacement compressors can be summarized as follows, with reference also to Figure 2.7 [2]:

- steep speed lines, *i.e.*, small drop in flow-rate with increasing pressure ratio;
- the pressure ratio is largely independent of rotational speed, so that a large boost pressure can be obtained even with small flow-rates;
- there is no range of unstable operation, so that the compressor map area can be fully utilized; and
- the flow-rate is roughly proportional to speed, quite independent of pressure ratio.

There are two important parameters that favor engine transient response with such a supercharging scheme in contrast to exhaust gas turbocharging: the compressor speed is directly coupled to its engine counterpart via a fixed or

variable gear ratio, hence compressor speed follows engine acceleration; on the other hand, high boost pressure can be achieved even at low (engine) speeds and air-mass flow-rates.

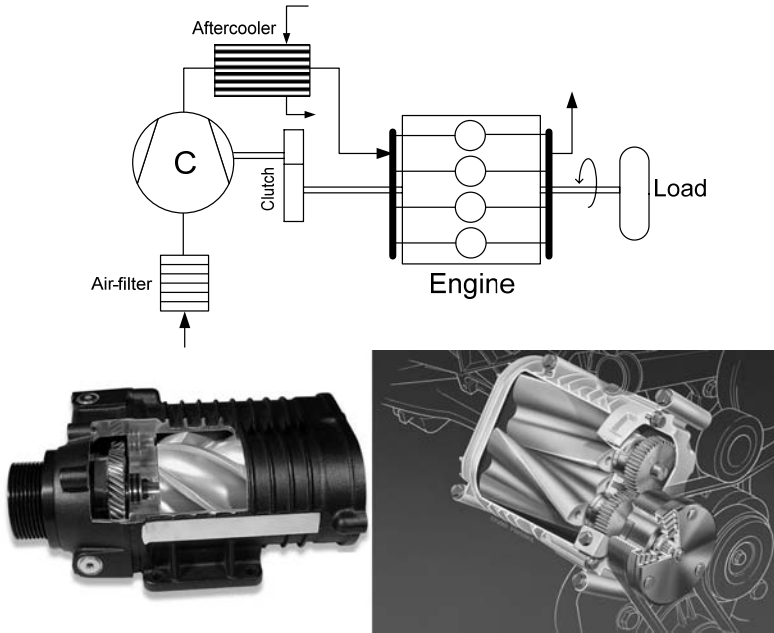


Figure 2.6. *Upper:* Schematic diagram of mechanically supercharged and aftercooled engine. *Lower left:* Mechanical, screw-type supercharger (courtesy of Lysholm® Technologies AB, Sweden). *Lower right:* Mechanical Roots blower installed on a Mercedes-Benz gasoline engine (source: Mercedes Benz)

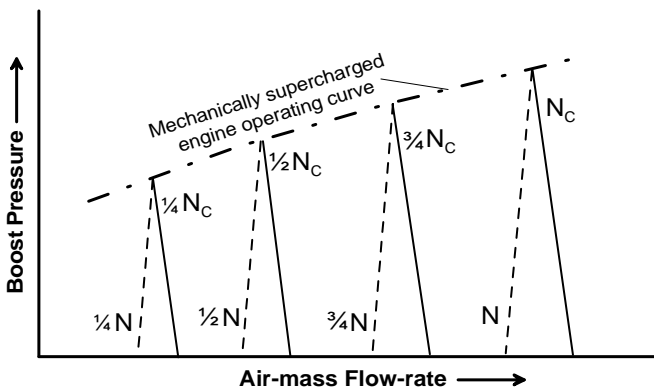


Figure 2.7. Flow map of a four-stroke supercharged engine with mechanically driven, positive displacement compressor (N corresponds to engine speed and N_C to compressor speed, solid line corresponds to positive displacement compressor curves and dashed line to four-stroke engine curves), (from Zinner [2], reprinted with kind permission from Springer Science+Business Media)

In addition to the delays discussed previously that concern the exhaust gas side, the inlet manifold contributes to the total system delay too via two mechanisms. The first is related to the EGR system and will be analyzed in more detail in Section 2.5. However, even at this point, it should be emphasized how the EGR management affects the system transient delay. In a diesel engine, recirculated exhaust gas replaces oxygen, hence during the turbocharger lag phase if the EGR rate is kept unaltered, an even lower amount of fresh air would hit the engine. Unsurprisingly, the main strategy in modern engines is shutting down the EGR valve during the first seconds of a transient event in order to aid faster build-up of combustion air-supply. The second mechanism originates from fluid mechanics grounds, namely the compressed air must first fill the inlet manifold volume before it affects the cylinder charge. Increased engine ratings generally result in higher inlet manifold volumes, hence slower response in the intake side. Similarly to its exhaust counterpart, the inlet manifold acts as a flow capacitor. This is documented in Figure 2.8, showing manifold pressure histories during the first second of a turbocharged diesel engine's load acceptance transient event.

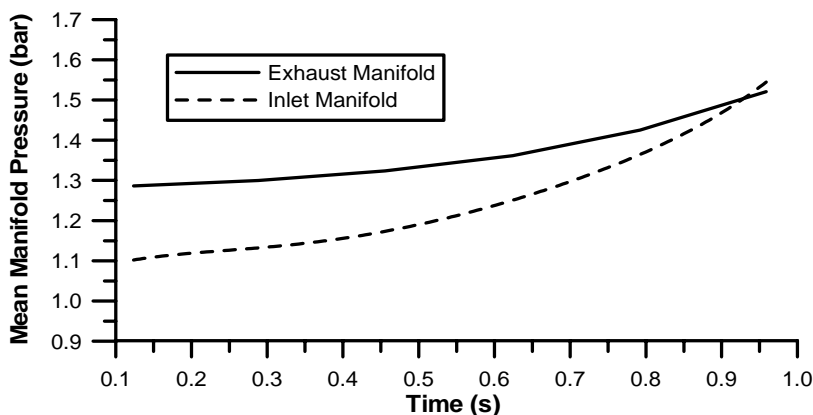


Figure 2.8. Manifold pressure histories during the first cycles of a 0–90% load increase transient event of a six-cylinder, DI, turbocharged diesel engine

Although the exhaust manifold pressure is increasing following the boost in fueling, the inertia of the turbocharger as well as the gas dynamics of filling the intake manifold cause a further lag on the intake side. With respect to the turbocharger compressor map curves (Figure 2.5), this lag concerns both boost pressure and air-mass flow-rate. Consequently, negative pressure difference is noticed for the first 0.95 s (roughly 15 cycles); on the other hand, during steady-state engine operation, a positive manifold pressure difference exists, *i.e.*, inlet manifold mean pressure is higher than its exhaust counterpart, a fact that ensures successful scavenging of the cylinder contents. It is, therefore, expected that during the turbocharger lag cycles and for the valve overlap period (which is particularly high in turbocharged engines to enhance their breathing and lower the gas

temperatures at the turbine entry) scavenging is difficult, with substantial reverse flow occurring.

The latter phenomenon induces an even more unfavorable operating aspect in two-stroke engines, due to the absence of separate induction and exhaust strokes. This scavenging dysfunction is reflected into an increase of the residual gas fraction, reducing further the trapped air–fuel ratio. In modern automotive or truck diesel engines equipped with variable geometry turbochargers, the above phenomenon is further enhanced, since the VGT vanes are rapidly closed following initiation of the transient event, thus increasing the exhaust back-pressure even faster. Reduction of valve overlap or, better still, variable valve timing with valve overlap adjustment during transients could be a promising consideration in order to cope with these scavenging issues.

Until the point where the air-supply inside the cylinder has matched the increased quantity of injected fuel, the overall fuel–air equivalence ratio assumes higher than stoichiometric values resulting in excessive smoke emissions. The problem may manifest itself in a more prominent way depending on the specific application. For example, in vehicles, the matching between turbocharger and engine is usually established for steady-state conditions and for low specific fuel consumption. Thus, the turbocharger size is determined for high torque output, which usually leads to high moment of inertia and consequently slow air-charge response. This slow air-flow response cannot match the required fast fueling commands during transients and so it causes poor response and black smoke emissions from the vehicles' exhaust pipes. Summarizing, Figure 2.9 demonstrates the major causes of turbo-lag after a load or speed increase transient event.

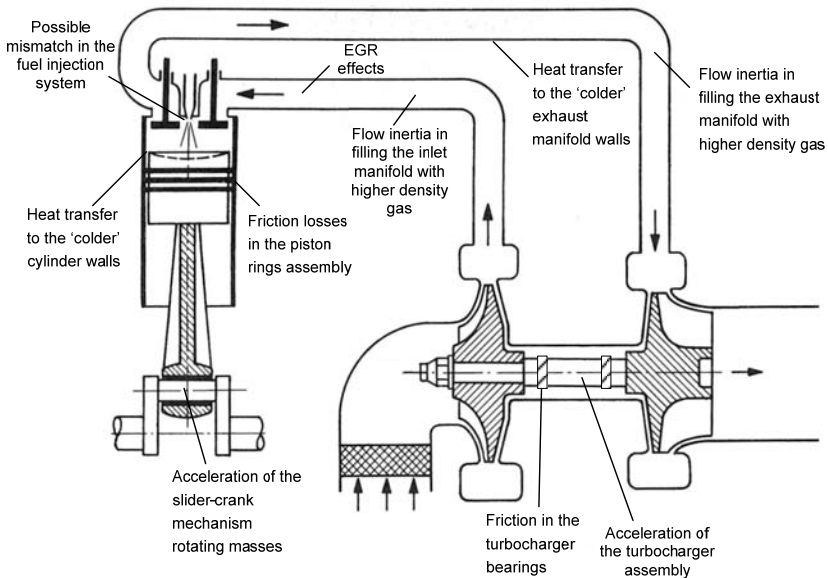


Figure 2.9. Schematic presentation of the major contributions to the system delay during transient response of a turbocharged diesel engine

The key parameter in the schematic arrangement of Figure 2.9 is the turbocharger mass moment of inertia, which is responsible for the biggest of the delays concerned. Unfortunately, turbocharger lag is more pronounced with increase in engine rating or the degree of turbocharging, which explains why it has become more prominent during recent years. Its most direct impact is on engine torque development and speed response.

An attempt to present the inherent difficulty of a turbocharged diesel engine to achieve fast response compared with a naturally aspirated engine, on a fundamental basis, was made by Watson [4]. He compared a naturally aspirated engine of 7 bar bmep with a turbocharged engine of the same size but twice this rating, *i.e.*, 14 bar bmep. Initially, the engines are running at no-load, no-speed conditions and they are rapidly accelerated to full-load, full-speed conditions. The governor responds to the demand and rapidly moves the fuel pump rack position to full-fuel delivery. Full-fueling for the turbocharged engine is twice that of the naturally aspirated, and it will need this fuel delivery to accept twice the load in the same timescale. Initially, however, the air-flow through the engines will be the same, since the turbocharger develops no boost at low-speed and low-load conditions. The need to accelerate the turbocharger rotor will then lead to slow boost pressure build-up. Thus, the fuel flow of the turbocharged engine is rapidly double that of the naturally aspirated, but the air-flow is pretty much the same. The maximum fuel delivery of the naturally aspirated engine will have been selected by the exhaust smoke-limited fuel-air ratio at steady-state full-load conditions. The fuel-air ratio developed instantly by the turbocharged engine will be twice its own fuel-air ratio when running steadily at full-load. In order to avoid excessive smoke emissions (well known dependence on fuel-air ratio), a fuel limiter (fueling determined according to boost pressure – see Section 2.2.2) would have to be installed, restricting further the engine torque developed during transients and leading to even poorer response. Figure 2.10 documents the above-mentioned remarks by highlighting

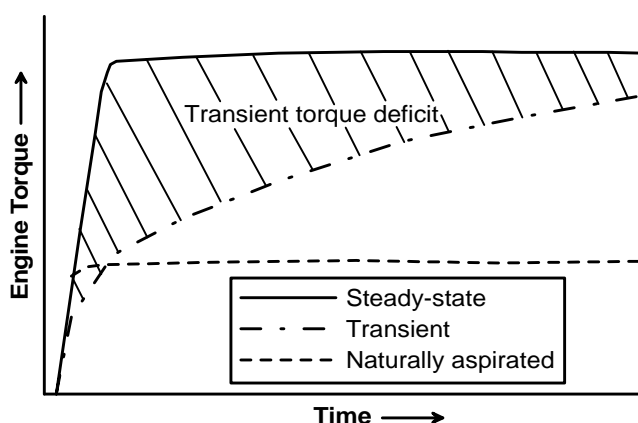


Figure 2.10. Comparison between transient and respective steady-state torque development during a load acceptance

- the *non-linearity* of turbocharged diesel engine transient torque build-up; and
- the substantial *torque deficit* during the first cycles of a load acceptance transient event when comparing the actual (transient) torque developed to the one under ‘similar’ steady-state operating conditions. Such torque deficit is observed during speed acceleration events too, as will be discussed later in the chapter.

Turbocharger lag and the subsequent fuel limit action are responsible for the turbocharged engine’s response pattern illustrated in Figure 2.10. As was mentioned earlier, a key parameter that largely influences the magnitude of the transient torque deficit is the engine rating. The lower the rating (bmep) of the turbocharged diesel engine, the smaller the above mentioned discrepancies and the smaller the transient torque deficit. This can be best explained by the following case study [5].

Table 2.1 presents data of several truck engines, all rated at the same maximum power output of 325 kW at 2300 rpm, but with different numbers of cylinders and kinds of aspiration. In all engines, a limit of 135 bar peak cylinder pressure, 1000 K maximum turbine inlet temperature and 20.5 minimum air–fuel ratio at 1000 rpm were imposed; the latter requirement resulted in fuel limiting control for the four-cylinder engine below 1300 rpm and for the six-cylinder engine below 1500 rpm under steady-state operation, as well as for the V-8 engine under transient conditions.

Table 2.1. Data of engines considered for the analysis of Figure 2.11 (reprinted with permission from SAE Paper No. 840134 [5], © 1984 SAE International)

Cylinder number and disposition	V-16	V-12	V-8	In-line 6	In-line 4
Total displacement (L)	25	18.75	12.5	9.375	6.25
Bmep (bar)	6.78	9.04	13.56	18.08	27.1
Compression ratio	16	15	13.5	12	10
Aspiration	Naturally aspirated	Single-stage, pulse t/c	Single-stage, pulse t/c	Single-stage, pulse t/c	Two-stage, pulse t/c
Intercooling	–	no	yes	yes	yes
Boost pressure (bar)	(1.0)	1.57	2.0	2.9	4.3

Figure 2.11 provides a very instructive comparison between the selected truck diesel engines’ acceleration, emphasizing the non-linearity of the turbocharged engines’ transient behavior. In the case of the naturally aspirated V-16 engine, acceleration is almost linear since full-load torque is available almost instantly, while the steady-state torque curve (not shown) is relatively flat. Acceleration of

the in-line, six- and four-cylinder engines is highly non-linear, largely owing to turbocharger lag. The fuel limiter allows here more fuel to be injected as boost pressure builds up (recall that there is a 20.5 minimum air-fuel ratio limit). Consequently, the response time of the most highly-rated engine (in-line, four-cylinder, 27.1 bar bmep, two-stage turbocharged and aftercooled) is nearly 2.5 times that of the naturally aspirated, while at the same time achieving only 74% of the distance covered by the V-16 engine in this first gear acceleration! Times at intermediate speeds are even worse for the turbocharged engines compared with the naturally aspirated and are due to the non-linear acceleration. However, it should be pointed out that continued acceleration to higher gear would be less affected by the degree of turbocharging as, in that case, the turbocharger speed would not fall to a no-load value during gear change.

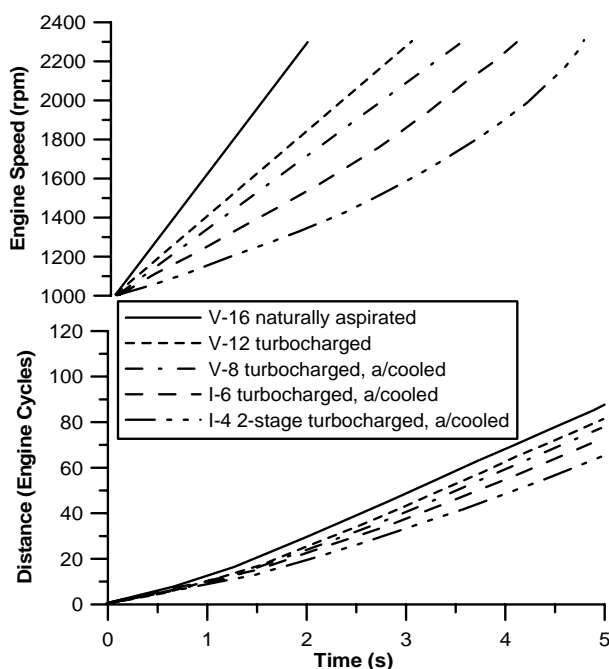


Figure 2.11. First gear acceleration from 1000 to 2300 rpm and distance covered (expressed in engine cycles) for the five truck engines of Table 2.1 possessing different kinds of aspiration and turbocharging options but rated at the same power output (simulation results reprinted with permission from SAE Paper No. 840134 [5], © 1984 SAE International)

The following interesting results were also revealed from the above analysis.

- The instantaneous engine bmep of the V-16 naturally aspirated engine equaled full-load, steady-state bmep after only 0.25 s (following governor response). This highlights the fast response a naturally aspirated engine can actually achieve. In fact, excluding starting, a naturally aspirated diesel engine can be assumed to behave in a quasi-steady manner during

transients. On the other hand, the relative fuel–air equivalence ratio never exceeded 0.7 (20.7 air–fuel ratio).

- The acceleration of the V-12 engine was so slow that the boost pressure did not exceed ambient pressure even by the time the engine had reached the desired speed.
- For the in-line, six-cylinder engine, although the boost pressure reached 1.6 bar at maximum engine speed, most of this increase occurred during the last second of the transient event. The non-linearity of the boost pressure build-up was determined by the injected fuel quantity via the fuel limiter, and influenced the whole acceleration of the engine.
- For the four-cylinder, two-stage turbocharged engine, the high-pressure turbocharger accelerated reasonably well but the low-pressure unit provided no boost for the first three seconds.
- For all turbocharged engines, the transient bmep was very much less than the steady-state capability of the engine (*cf.* Figure 2.10).

Watson concluded that although the turbocharger lag did not actually get any worse with the increase in rating, it is its influence on engine response that becomes more and more serious.

Turbocharger lag effects are noticed during load or speed *decrease* transients too. Again, the air-charge cannot instantly match the new, decreased fueling rate. However, in this case, the fuel–air ratio temporarily assumes very low values, hence smoke, NO_x and CO emissions are not a problem; as will be discussed in Section 5.3, this does not hold true for HC emissions. On the other hand, for naturally aspirated diesel engines, all the system's delays summarized previously except for the predominant turbocharger lag and the interrelated delays of filling the exhaust and inlet manifold, do affect the engine transient response; their relative importance is, however, much lower owing to the considerably smaller engine ratings.

Turbocharged gasoline engines suffer from turbocharger lag too during transients. In fact, the situation can prove quite challenging (particularly so for carbureted engines, where fuel hold-up on the fuel-wetted surfaces must be avoided) owing to the wider speed, hence air-mass flow range of spark ignition engines that leads to much harder transients, *e.g.*, when accelerating from closed to wide open throttle. In this case, however, the problem manifests itself as poor driveability or slow acceleration rather than black smoke emissions. Typical results are illustrated in Figure 2.12 for a 2.8 L turbocharged SI engine low-load acceleration. During first gear acceleration, the engine behaves in a naturally aspirated mode for at least 1 s (up to 2200 rpm); things are quite improved at fourth gear acceleration, owing to the much higher turbocharger initial operating point. In any case, torque development is substantially delayed compared with steady-state operation (*cf.* Figure 2.10) affecting negatively vehicle driveability.

Turbocharger lag and the associated system delays are the main reason for the problematic transient response of turbocharged (diesel) engines, affecting unfavorably the combustion process and exhaust emissions; it is not surprising then that the majority of the analyses and research have focused on methods for improving this delay/response. Variable geometry turbines, air-injection in the

compressor, retarded injection timing, electrically assisted turbocharging, variable valve timings are but a few measures that have been proposed as viable means for improving turbocharged diesel engine transient response; these and many other measures will be discussed in detail in Chapter 6.

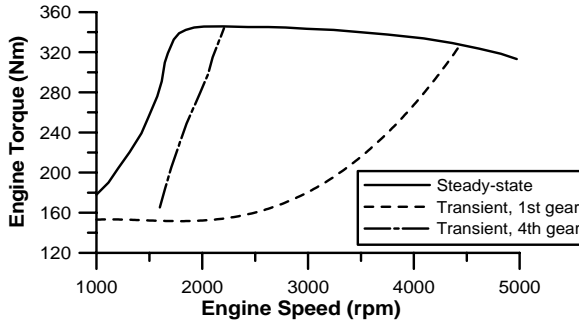


Figure 2.12. Comparison between transient and the respective steady-state torque development during acceleration in first and fourth gear for a 2.8 L displacement volume, turbocharged SI engine (from Heireth and Withalm [6], copyright VDI)

2.2 Fuel Injection

2.2.1 Mechanical Fuel Injection

One of the most important engine processes that influences combustion and thus heat release rate, exhaust emissions (mainly particulate matters) and transient response is the fuel injection system. As was discussed in the previous section, without a properly matched transient fuel delivery on a cycle-by-cycle basis the low in-cylinder trapped mass will initially result in overshoot of the global fuel–air equivalence ratio, leading eventually to intolerable smoke emissions. Hence, the dynamic characteristics of the injection system should be adjusted and matched to the current engine fueling requirements. The fuel injection mechanism operates in a highly transient manner even during steady-state engine operation (*e.g.*, at constant engine speed and control lever position).

In modern, high-pressure common rail injection systems governed by the engine ECU, the extremely high injection pressure poses one more significant challenge for the fuel injection dynamic response. Nonetheless, a relatively fast adaptation of the injection parameters to the incoming air-charge is usually achieved,⁴ provided that the measurement of air-mass flow-rate is made promptly and accurately; this is not the case, however, in conventional mechanical injection systems. During transients, the continuously changing engine/pump speed, control

⁴ At engine start-up, it usually takes 3 to 4 injections to establish line pressure in the high-pressure section of the injection system. Afterwards, stable, repeated injection is assured.

lever position and pump residual pressure result in the (mechanical) fuel pump experiencing a transient operation of its own. This is mainly the result of transient fluid mechanics of the fuel pump equipment and the dynamic response of the mechanical components of the fuel injection system (governor, needle lift and timing device). Consequently, various injection parameters such as line pressure, rate of injection, timing and duration of injection are influenced, with a subsequent differentiation in the amount and the profile of injected fuel per cylinder compared with the values under 'similar' steady-state conditions [7–10]. The latter behavior ultimately affects engine response.

Murayama *et al.* [7], by studying the accelerating behavior of a single-cylinder diesel engine fitted with mechanical fuel injection pump, found that

- owing to rapid and considerable changes in fueling, instantaneous torsional deformations in the driving system of the fuel pump take place, leading to incomplete combustion;
- a retardation of the dynamic injection timing takes place too, which was correlated to injected fuel-mass change via a linear relation (Figure 2.13);
- the shape of the fuel injection rate changes compared with the respective steady-state operation. A shortening of the throttling period at the early stage of fuel injection and the increase in fuel injection rate following throttling, are more pronounced under acceleration than under the corresponding steady-state conditions (Figure 2.14a). Later in the transient, the fuel injection rate curve assumes a more 'rectangular' shape. Moreover, during transients, injection starts later in the cycle compared with the respective steady-state operation; this originates in the lower values of the residual pressure of the fuel delivery system (Figure 2.14b).

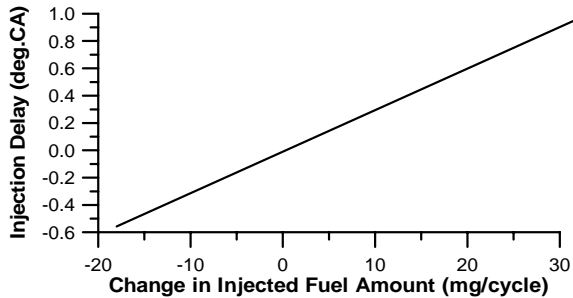


Figure 2.13. Change in the dynamic fuel injection timing due to the deformation of the injection pump driving system, resulting from changes in the amount of injected fuel (experimental results reprinted with permission from SAE Paper No. 800966 [7], © 1980 SAE International)

Although some of the above results were reached for naturally aspirated, indirect injection diesel engine operation, where the injection pressure are traditionally low, they have been confirmed for turbocharged DI engines too; in fact, even bigger discrepancies are expected for turbocharged diesel engines owing to the considerably higher difference between no-load and full-load fueling and to the much higher injection pressures involved.

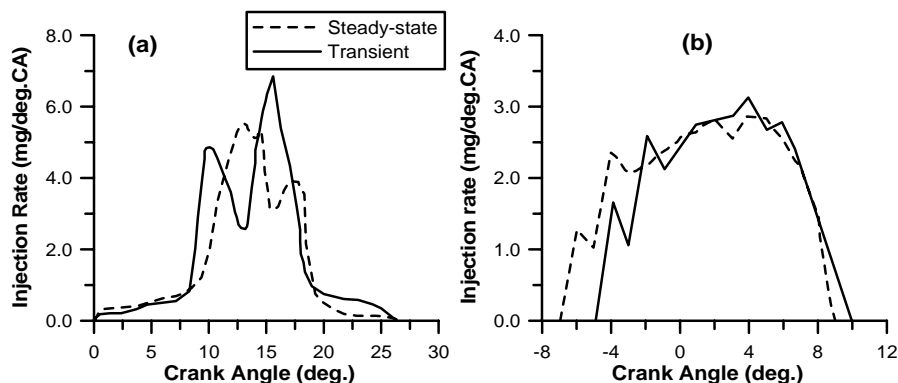


Figure 2.14. Change in fuel injection rate during an early cycle of an acceleration event compared with the respective steady-state operation. *a*: naturally aspirated engine (experimental results reprinted with permission from SAE Paper No. 800966 [7], © 1980 SAE International). *b*: turbocharged diesel engine (experimental results reprinted with permission from Harndorf and Kuhnt [11])

In order to focus on the dynamic injection characteristics, Arcoumanis and Baniasad [9] isolated the fuel injection system operation of a VE type distributor injection pump of an HSDI diesel engine and studied its transient response. Two dynamic operating conditions were examined mimicking ‘real’ engine transient operation, *i.e.*, 1. fuel pump setting (control lever) position varying from minimum to maximum in 13 ms at constant pump speed (termed ‘fueling increase’ transient), and 2. increasing pump speed at constant control lever position (termed ‘acceleration’ transient). From their analysis, the following were revealed.

During the fueling increase period at 2000 rpm engine speed (1000 rpm pump speed) 4 injection cycles were needed until the demanded fuel quantity is reached (Figure 2.15) (more cycles were needed for transients at higher engine/pump speeds). The transient injection rate signals (not shown in Figure 2.15) increased significantly in terms of the peak amplitude but only slightly in duration (*cf.* Figure 2.14), whereas the corresponding needle lift signal showed higher needle ascend velocities and longer signal durations due to the higher injection line pressures. Owing to the fluid and mechanical inertia of the system during these cycles, lower maximum needle lift, line pressure and injection rates were experienced compared with the demand conditions reached in cycle 4. The latter may result in poor fuel atomization and reduced spray penetration into the combustion chamber, as well as lower injected fuel quantities during the transient period in an operating engine. Actually, the injected fuel quantity was calculated to be 82, 86, 94 and 99%⁵ of the demanded for the 4 transient cycles depicted in Figure 2.15.

However, in view of the slow build-up of air-supply to the engine cylinders, this gradual adjustment of fueling is favorable, as it prohibits very large fuel–air

⁵ For a ‘transient fueling’ test at higher engine speed (2500 rpm), the respective percentages were 65, 73, 81, 86, 91, 97, 102 and 99% of the demanded fuel quantity. This leads to the conclusion that in terms of the injection cycle, the system has faster response at lower engine/pump speeds with the fuel quantity response deteriorating with increasing speed.

equivalence ratio overshoots during the first transient cycles. At the same time, residual pressure was, mainly, determined during the tail oscillation of the line pressure, and was dependent on the opening characteristics of the damping valve and the resultant reflected pressure.

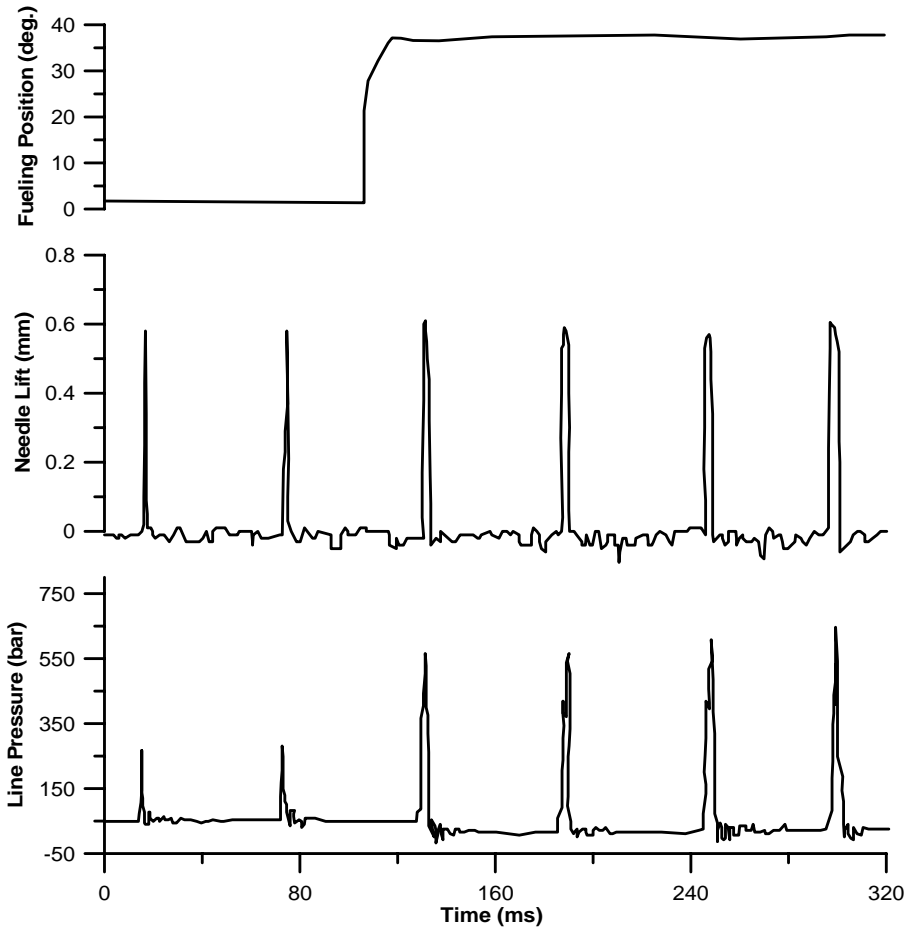


Figure 2.15. Response of speed control lever (fueling) position, needle lift and line pressure for a transient fueling test of a VE type distributor pump at constant speed (experimental results reprinted from Arcoumanis and Baniasad [9], with permission from Professional Engineering Publishing)

Catania *et al.* [10], studying a similar injection pump configuration, reached the conclusion too that the fuel injection system was capable of almost instantaneously adapting to the new stationary conditions in a few cycles, soon after the control lever was brought to its new steady-state position. They attributed the latter effect to the small sensitivity of the delivery pipe residual pressure to significant load variations and to the almost zero lasting cavitation effect on fluid properties at the

start of the pump compression stroke. With the increase in pump speed, however, a slightly more intense cavitation⁶ was observed, ultimately affecting the fuel fluid properties during the pump compression stroke.

During the increasing pump speed operation, residual pressures were found less than those at steady-state operation for lower pump speeds, but at higher pump speeds transient residual pressure exceeded its steady-state counterpart until final convergence towards the end of the transient event. At the start of the 'acceleration', transient injection advance timing suddenly dropped and recovered gradually with increasing pump speed (Figure 2.16 – upper sub-diagram). This trend in the transient injection timing, which was found independent of load, was attributed to: 1. advanced timing piston and spring mechanism response characteristics; and 2. fuel feeding pressure build-up in the pump housing cavity, being dependent on the pressure response of the fuel supply pump and the rate of pressure rise in the large pump cavity feeding chamber within the rather short transient time. The difference between transient and quasi-steady injection advance timing is one of the major contributors to the higher transient soot and NO_x emissions levels in diesel engines, to be discussed in Chapter 5.

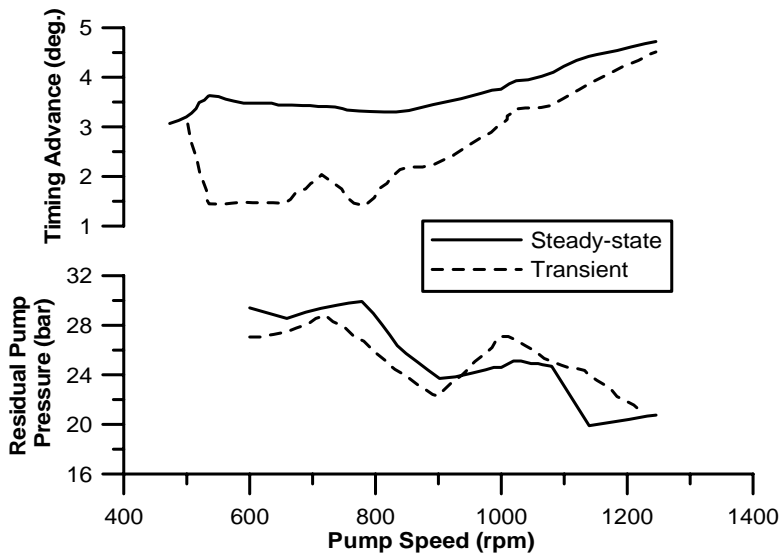


Figure 2.16. Comparison between transient and steady-state residual pressure and injection timing advance of a VE type distributor pump at constant control lever position (experimental results reprinted from Arcoumanis and Baniasad [9], with permission from Professional Engineering Publishing)

⁶ Cavitation in the pipe of a fuel injection system occurs when locally fuel pressure decreases below its vaporization value forming cavities; it is usually the more volatile components of diesel fuel that are prone to vaporization. Increased speeds and injection pressures as well as abrupt termination of injection are mainly responsible for creating vigorous wave actions inside the fuel pump mechanism leading to the above-mentioned local reduction of fuel pressure and thus formation of cavities. Vapor may subsequently condense inside the cavities leading to erosion and even fuel injection failure.

2.2.2 Fuel Limiter

Figure 2.17 illustrates a typical fueling/bmep vs. engine speed operating envelope for two turbocharged diesel engines in comparison with a naturally aspirated one. Irrespective of output, the problematic operation for turbocharged engines is located in the low-speed, medium-high load operating range, where the turbocharger compressor delivery pressure, hence, the air–fuel ratio are still low since the exhaust gas enthalpy is not yet capable of providing the required power to boost the compressor. Increasing the degree of turbocharging results in more abrupt low-speed air–fuel ratio curves, leading to even higher engine torque non-linearity. In the mid-range, engine output is, mainly, limited by mechanical considerations (high cylinder pressures) and in the high-speed range by turbocharger and engine over-speeding.

In view of the above, at low-speed, steady-state engine operation in order to avoid intolerable smoke emissions, special adjustment of the fueling system to the corresponding air-supply is required. This adjustment is usually realized via a fuel limiting function, which determines maximum fuel delivery according to the current turbocharger compressor boost pressure. Ideally, EGR effects should be taken into account too since the amount of recirculated exhaust gas lowers accordingly the fresh charge air–fuel ratio. As regards mechanical fuel pump/governor systems, the fuel limiting function can be accomplished by a damper system momentarily limiting fuel pump rack travel, or, more commonly, via a boost pressure sensitive device. In modern engines, the fuel limiting module is incorporated in the engine ECU together with the speed governor (*cf.* Figure 3.11b). The usual inputs to the fuel limiter are the compressor boost pressure and/or the air-mass flow-rate.

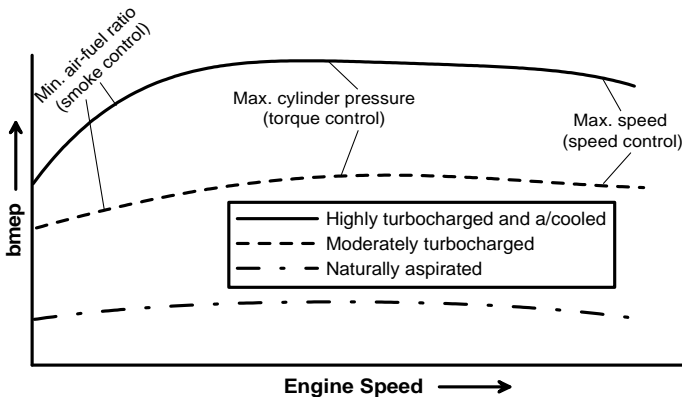


Figure 2.17. Envelope of maximum bmep vs. engine speed for steady-state operation of naturally aspirated and turbocharged diesel engines

Although a well designed fuel limiting device can actually prevent intolerable smoke emissions at steady-state engine operation, during turbocharger lag, much higher fuel–air equivalence ratios may be experienced. It is not surprising then that the steady-state matching often proves unsuccessful during transients. The most

usual side effect is the transient torque deficit (*cf.* Figure 2.10) and the poorer engine response/acceleration; a very instructive comparative case study can be found in [4]. To describe the mechanism behind the trade-off between fast response and low smoke emissions, it is necessary to consider in more detail the way the fuel limiter operates.

The operating principles of a typical fuel limiter will be discussed with reference to Figure 2.18, which illustrates fueling and air-supply build-up during an acceleration transient event. Initially (point 1), the engine operates at or near idling conditions with the fuel-air equivalence ratio Φ_{idle} corresponding to fueling F_1 and air-supply A_1 , both possessing very small values. Afterwards, the transient event commences, with the driver pushing the accelerator pedal to its maximum position. In case of a naturally aspirated engine, this would result to a direct increase in fueling (up to the corresponding final point). This is not the case, however, for the turbocharged diesel engine. Recall from the case study discussed in Section 2.1, that a turbocharged engine's maximum bmep is much higher than that of its naturally aspirated counterpart, hence maximum fueling corresponds to a considerably higher air-supply. However, owing to the various system delays analyzed in Section 2.1, air-supply build-up is, initially, very slow (1→3). In fact, for the first cycles after the acceleration commences, boost pressure and air-supply to the engine cylinders have not changed from their initial values.

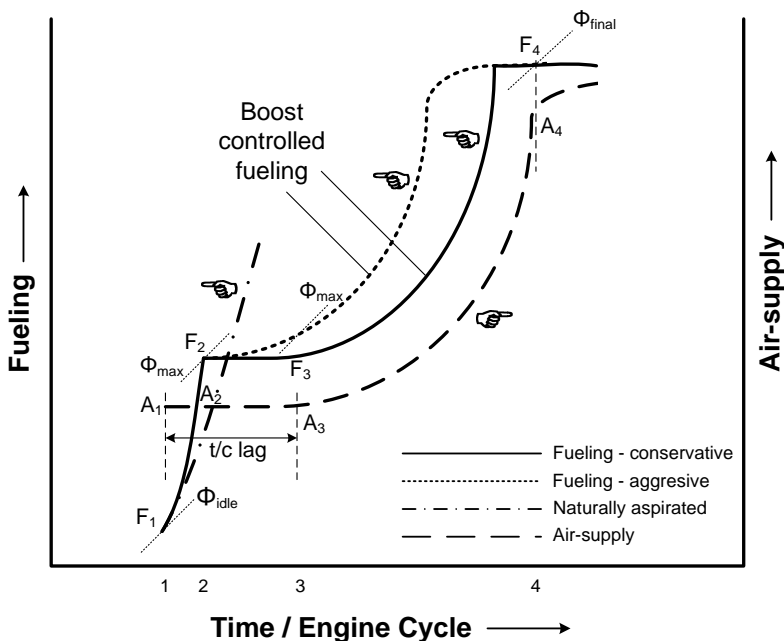


Figure 2.18. Typical fuel limit pattern of a turbocharged diesel engine acceleration event

During this period (turbocharger lag), the engine operates in a naturally aspirated mode. Because of this slow response, the fuel limiter allows, at first, only a modest increase in the injected fuel mass, namely from F_1 to F_2 , to prevent

excursion of the fuel–air equivalence ratio to very high values. At this point (2), maximum fuel–air ratio might be experienced. This fueling is then maintained, more or less, constant for the whole turbocharger lag period (up to point 3). As the acceleration event develops and the increased fueling produces exhaust gas of higher enthalpy, the turbocharger is gradually accelerated increasing the boost pressure and air-supply to the engine cylinders, curve A_3 – A_4 . During this period, fueling is steadily increasing too in proportion to air-supply, ultimately reaching point 4, which, usually, corresponds to the maximum fuel delivery for the current transient event. A critical parameter here, is the slope of curve F_3 – F_4 ; the steeper the boost controlled fueling curve (dotted line in Figure 2.18), the faster the fueling response, hence engine torque build-up and speed response, but also the higher the fuel–air equivalence ratio and the respective soot emissions (most probably at point 3). In the case of fixed geometry turbines, air-supply at the final point 4 would correspond to the maximum value for the particular transient event. For variable geometry turbocharged engines, however, the gradual increase in exhaust manifold temperature will lead to a further increase in the turbine back-pressure, hence a slight increase of turbocharger speed will be accomplished, possibly differentiating the final conditions after several seconds [12,13]. In conclusion, fueling during a typical acceleration event is accomplished in three distinct phases as follows

- During the first phase (1→2 in Figure 2.18), the mass of injected fuel is rapidly increased to a value limited by the available air-flow owing to boost pressure lack. In this phase, fueling has increased at a much sharper rate than air-flow.
- During the second phase 2→3, fueling is practically maintained constant owing to the respective delay in the build-up of boost pressure during the turbocharger lag period.
- During the third phase 3→4, fueling is controlled by the engine ECU and it basically follows air-supply pressure history until the final equilibrium. A steeper 3→4 fueling curve will result in faster fueling increase and speed response, coupled, however, with excursion to high fuel–air equivalence ratios and soot emissions.

The above-described fuel limiting action primarily aims at limiting intolerable smoke emissions, which would otherwise be experienced if fueling were determined irrespective of air-supply. Nonetheless, the slow movement of the fuel rack towards full-fueling comes at a cost, namely slow engine torque build-up and speed recovery. The latter can be best documented with reference to Figures 2.19 and 2.20. Figure 2.19 focuses on the torque development during vehicle acceleration in fourth gear and can be studied in conjunction with the remarks regarding Figure 2.18.

Initially (point A at 1250 rpm) and following the driver's demand, the transient engine torque increased sharply owing to the rapid movement of the governor to its maximum value (point B). However, the air-supply could not match the fuel flow and the engine practically operated as naturally aspirated. The slow build-up of boost pressure was sensed by the fuel pump mechanism, which in turn limited the amount of injected fuel according to the current compressor boost pressure. Consequently, transient torque developed at a much slower rate compared with the

steady-state curve (1300–2750 rpm), ultimately reaching point C_t , which corresponds to lower torque and at a higher engine speed than the respective steady-state point C_{st} . As a result, emission of excessive smoke has been prohibited but at the expense of poorer response. For naturally aspirated engines, the difference between points C_t and C_{st} would be much smaller, since the greatest of the above mentioned delays concerns the turbine acceleration [14].

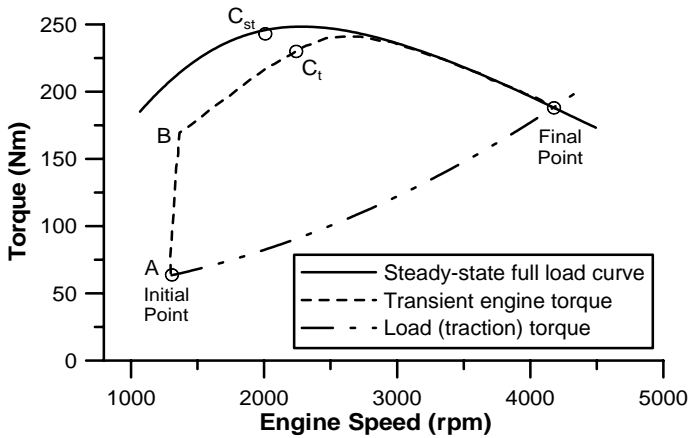


Figure 2.19. Curves of steady-state full-load torque, transient engine torque and load (traction) torque for vehicle acceleration in high gear, showing the effect of fuel limiting function (reprinted with permission from Woschni *et al.* [14])

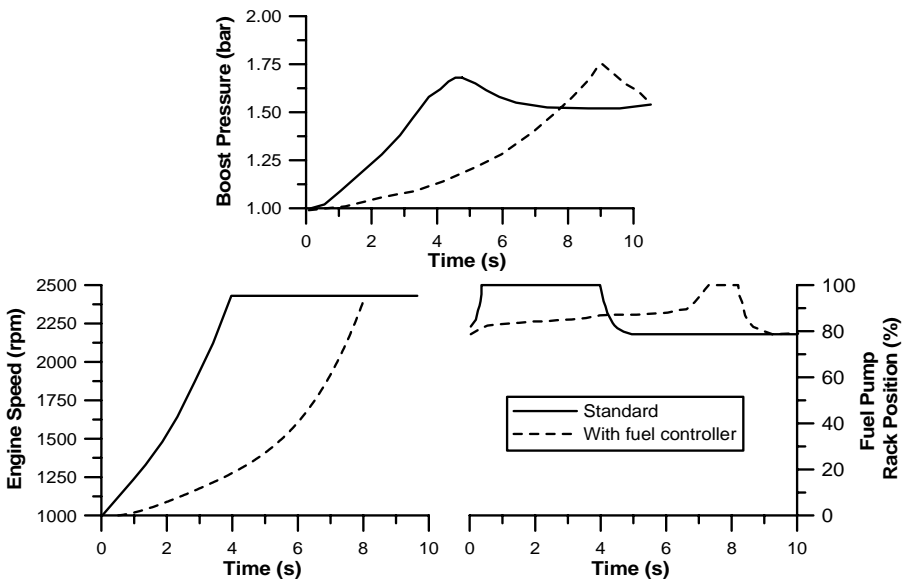


Figure 2.20. Acceleration of a turbocharged truck diesel engine against steady load with and without fuel limiting function (experimental data reprinted with permission from SAE Paper No. 810338 [4], © 1981 SAE International)

The effect of fuel limiting function on engine speed response is quantified in Figure 2.20, where the acceleration of a turbocharged truck engine of modest rating (10 bar maximum bmep) is compared with and without a system for controlling maximum fuel delivery as a function of boost pressure. Limiting maximum fuel delivery by 20% at zero boost is capable of reducing the total amount of emitted smoke owing to the fuel pump rack moving slowly towards full-fueling. However, the response time of the engine and turbocharger properties is doubled resulting in much poorer acceleration.

Further limitation of the fuel flow may prove critical for the engine's capability to recover high load changes leading even to engine stall, especially when the load application is very fast. This is, particularly, the case for industrial or marine applications, where the engine ratings as well as the load changes experienced are quite large. In view of the above, it seems reasonable to permit at least a 10% over-fueling margin for industrial engines that, furthermore, do not encounter transients that often, compared with vehicular applications. When the engine accelerates from an initial high load, smoke reduction from fuel limit action is generally negligible; on the other hand, the advantages of (slight) over-fueling with respect to successful engine speed recovery are considerable. Of course, the case is quite different when automotive engines are involved. The use of fuel limiters is here compulsory in order to conform with the stringent emission regulations. Nonetheless, other measures would have to be employed in order to deal with the root of the problem, *i.e.*, minimize turbocharger lag, hence simultaneously enhancing vehicle driveability and lowering emissions (Chapter 6).

2.3 In-cylinder Processes

2.3.1 Heat Transfer

At the onset of a transient event, cylinder wall temperature corresponds to the steady-state conditions, being mainly determined by engine loading/fueling. The following equation describes the respective heat flux from the gas to the cylinder wall

$$\frac{dQ_L}{dt} = A h_g (T_g - T_w) \quad (2.3)$$

where A is the respective surface area, h_g the heat transfer coefficient and T_g , T_w the instantaneous gas and wall temperatures, respectively. During the early cycles of the transient event, the gradual increase in fueling is transformed into higher gas temperatures T_g and heat transfer coefficients h_g . Consequently, higher heat transfer rate dQ_L/dt to the walls is noticed, since the cylinder wall temperature T_w cannot adjust instantly to the new fueling conditions but only after a time delay owing to the wall's thermal inertia. The latter behavior is one of the contributors to the engine response delay as was discussed in Section 2.1 and identified in

Figure 2.9. Later in the section, it will be shown that this time delay can be quite extensive, typically one order of magnitude greater than the actual duration of the load or speed increase transient event or the time needed for the engine speed to reach its new final value. Among other things, this behavior affects also the evolution of (transient) combustion noise.

In order to evaluate the wall's thermal inertia, which determines the wall's temperature response, a heat transfer scheme, based on electrical circuit analogy, can be applied. This models the temperature distribution from the gas to the cylinder wall up to the coolant. By so doing, the cylinder wall thickness, thermal conductivity and thermal diffusivity are taken into account. For the heat transfer inside the cylinder wall, the one-dimensional unsteady heat conduction equation reads

$$\frac{\partial T}{\partial t} = \alpha_w \frac{\partial^2 T}{\partial x^2} \quad (2.4)$$

with $\alpha_w = k_w / (\rho_w c_w)$ the wall thermal diffusivity, ρ_w the wall density and c_w its specific heat capacity. Applying the boundary conditions to all wall sides (gas side and coolant side) of a four-stroke diesel engine, the following equation is valid with reference to Figure 2.21:

$$\frac{1}{4\pi} \int_0^{4\pi} \frac{dQ_L}{d\phi} d\phi = A \frac{k_w}{L_w} (\bar{T}_{w,g} - \bar{T}_{w,c}) = A h_c (\bar{T}_{w,c} - T_c) \quad (2.5)$$

where L_w is the cylinder wall thickness with k_w its thermal conductivity h_c the heat transfer coefficient from the external wall side (respective temperature $\bar{T}_{w,c}$) to the coolant; the overbar denotes mean temperatures over an engine cycle.

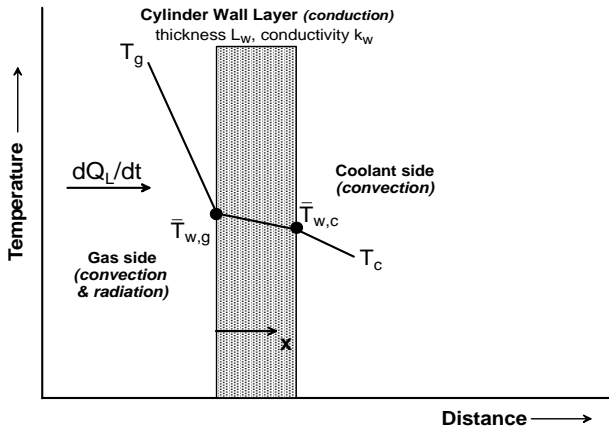


Figure 2.21. Schematic arrangement of cylinder wall – coolant heat transfer scheme

Representative results quantifying the influence of the above mentioned wall thermal balance are reproduced in Figures 2.22 and 2.23, showing cylinder wall

and cylinder liner thermal equilibrium procedure after a load increase and a speed increase transient event, respectively. Measurements were accomplished using special 'J'-type thermocouples, which comprised of two thin insulated wires each having a 0.003" diameter. The two wires were fixed inside a hollow quartz tube of 1.75 mm outside diameter with special high temperature cement and fastened inside their position on the cylinder head or liner with the aid of the same material. Point TC1 in Figures 2.22 and 2.23 refers to the thermocouple installed on the cylinder head between two fins, in contact with the outside metal surface, TC2 refers to the thermocouple installed on the cylinder head interior at a distance of 0.6 mm from the fire deck, while TC3 thermocouple was installed at the valve bridge interior at a distance of 10 mm from the fire deck. In a similar way, the cylinder liner thermocouples were installed at a distance of 5 mm from the gas side surface, at two opposite positions across the cylinder bore on the valve plain, one on the exhaust valve side (TC4) and the other on the inlet valve side (TC5).

As is depicted in Figure 2.22, the steepest slope is observed for the valve bridge thermocouple (TC3), while the most moderate response is at the outside surface of the cylinder head point (TC1). As expected, the valve bridge being one of the most sensitive areas on the cylinder head, suffers from thermal distortion caused by sharp temperature gradients during a transient event (thermal shock). The important finding from Figure 2.22 is that even 250 s after initiation of the load increase, the temperatures have still not reached their final steady-state values. Hence, during the few seconds of the transient event (up to the point of engine speed recovery) transient heat loss to the cylinder walls is higher than under the respective steady-state conditions. For turbocharged engines, this behavior would lower the available exhaust gas energy to the turbine and further contribute to the turbocharger lag delay. For the cylinder liner alone, and due to the air-cooling of the engine, the temperature response (TC4 and TC5) is rather unstable although the main finding of slow thermal equilibrium is confirmed.

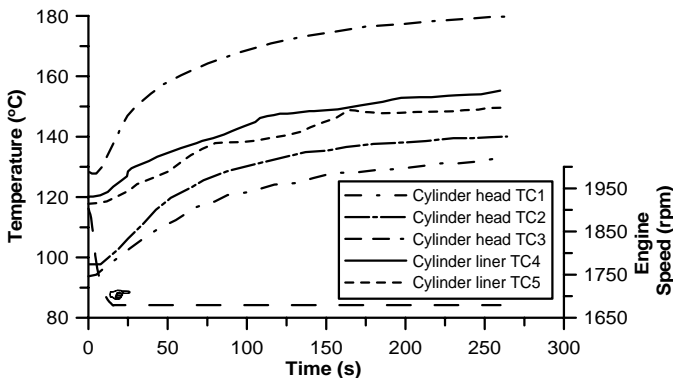


Figure 2.22. Experimental temperature profiles vs. time during a 12–70% load increase transient event of a single-cylinder, air-cooled, naturally aspirated diesel engine [15]

Similar observations are valid for the case of speed increase transients, as is illustrated in Figure 2.23. Temperature levels, for all positions studied, present smaller differences when comparing initial and the final operating conditions.

Although fueling reached 100% for a few seconds during the initial phase of the acceleration, the high thermal inertia of the cylinder structure delayed significantly the wall temperature response, hence the highest temperatures observed actually correspond to the final fueling of 30% rather than to the instantaneously maximum 100%. Again, the steepest thermal gradient is noticed in the valve bridge area. Contrary to the (usually much more demanding) load acceptance case, thermal equilibrium is achieved now after 120 s for all positions examined on the cylinder head and liner.⁷

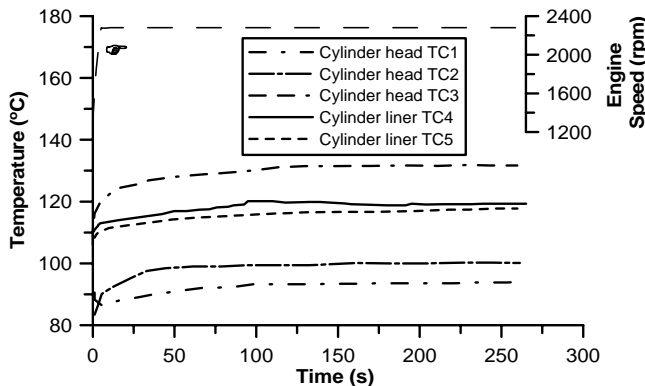


Figure 2.23. Experimental temperature profiles vs. time during a low–medium load, 1300–2300 rpm acceleration event of a single-cylinder, air-cooled, naturally aspirated diesel engine [15]

A similarly slow, but not that long in duration, response is observed for the heat transfer rate to the exhaust manifold walls (Figure 2.24). Evaluation of the exhaust gas temperature depends strongly on the location of the thermocouple, which for the analysis of Figure 2.24 was located at the exhaust manifold at an approximate distance of 0.2 m from the corresponding valve.

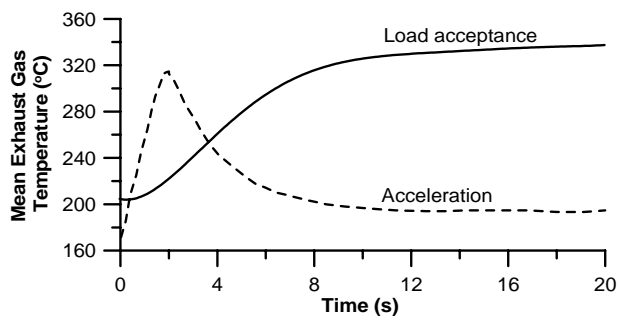


Figure 2.24. Experimental mean exhaust gas temperature vs. time for the load acceptance and acceleration transient events of Figures 2.22 and 2.23 [15]

⁷ The initial drop in TC1 temperature is caused by the increase in engine speed at the first five seconds of the acceleration, which increases the air velocity through the fins and the respective heat transfer coefficient with a simultaneous decrease in air temperature.

Further aspects of transient heat transfer phenomena will be discussed in Section 5.7 concerning evolution of combustion noise, and in Section 7.3 concerning low-heat rejection engine operation.

2.3.2 Combustion

Diesel engine combustion comprises a variety of physical and chemical processes being typically characterized by a high degree of non-uniformity. In a diesel engine, the overall air–fuel ratio is always weak of stoichiometric, in order to achieve complete combustion; this is a consequence of the rather limited time available for mixture preparation. The fuel is injected in liquid form as nearly conical jets (sprays) into the engine cylinder, under a very high-pressure difference between injection line and cylinder gas. However, before it can be burned, atomization and spray penetration into the compressed air must be established that will enable heating, evaporation and mixing of the fuel with a sufficient quantity of air entrained from the surrounding air via a diffusion process. At the same time, pre-flame oxidation and localized ignition occur that prepare chemically the fuel–air mixture for burning by decomposing the heavier hydrocarbons into lighter components, forming combustion radicals and facilitating pre-ignition chemical reactions. The above physical and chemical processes are referred to collectively as ‘preparation’, with the temperature and pressure of the compressed air prior to fuel injection playing here an important role and air-swirl promoting the fuel–air mixing rate. It is the droplets at the outer edge that evaporate first creating a fuel–air vapor film around the still liquid cone jet. The fuel–air equivalence ratio is highest on the centerline of the jet, decreasing to zero at the boundaries (Figure 2.25). The sprays eventually hit the cylinder wall or piston and mix together.

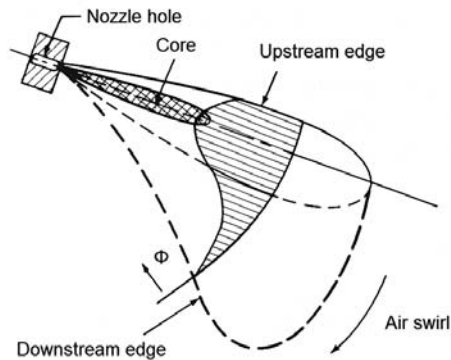


Figure 2.25. Schematic of fuel spray injected radially outward from the chamber axis into swirling air-flow; shape of fuel–air equivalence ratio distribution within jet is indicated (from Heywood [16], © 1988, reproduced with permission of the McGraw-Hill Companies)

The prepared fuel–air mixture is then under the control of a chemical process and may be burned (reacted) at a rate that can be calculated by a chemical kinetics

equation. Diesel combustion in conventional type DI or IDI engines is primarily influenced by the inducted air properties (density, swirl, temperature) as well as by the injection system characteristics (number and pattern of injections, injection pressure and rate, fuel cetane number, number and diameter of nozzle holes). A variety of other parameters such as compression ratio, cylinder wall temperature, intake system design, valve configuration, combustion chamber geometry, *etc.*, affect strongly the whole process development. Summarizing, for cylinders with single injection strategy, diesel engine combustion comprises the following stages:

- ignition delay period, where the injected fuel is physically and chemically prepared and mixed with air;
- a rapid premixed burning phase of the already prepared mixture during the previous ignition delay period; this stage is usually characterized by a high rate of gas pressure increase and is, mainly, responsible for radiation of combustion noise; and
- a slower, mixing- or diffusion-controlled burning phase, where the burning rate is not governed by chemical kinetics but, rather, by the fuel injection and subsequent turbulent mixing rates.

Application of multi-hole injectors, pre- and pilot injections, exhaust gas recirculation, swirl, and the use of bowl-in pistons complicate the development and modeling of combustion even more. As a result, advanced multi-dimensional simulation models as well as optical engine cylinders are required for complete understanding of the heterogeneous in nature in-cylinder processes in a compression ignition engine; for transient operation studies, the use of advanced multi-dimensional simulation models is, at the moment, prohibited due to the need for simulating a large number of engine cycles up to the final equilibrium conditions. A thorough review of diesel engine combustion principles during steady-state operation can be found, for example, in Heywood [16] and Benson and Whitehouse [17]. In this book, only the differentiations that are experienced during transients compared with the respective steady-state operation will be discussed.⁸ These are primarily the result of 1. the dynamic response characteristics of the fuel injection system, 2. the different quantity and pressure of air-supply owing to the previously discussed turbocharger lag (Section 2.1), which affects directly fuel–air ratio, heat release rate and exhaust emissions, and 3. the lower cylinder wall temperature compared with the respective steady-state operation as discussed in the previous section.

Figure 2.26 shows data for a turbocharged DI diesel engine's response to a step increase in engine load. Cycles No. 6–11 of the transient event have been isolated and the development of engine and exhaust manifold pressures, turbine inlet temperature, engine speed, fueling, mean boost pressure, and relative air–fuel ratio are illustrated with respect to crank angle or time.

⁸ By the term *respective* steady-state operation, it is meant operation at the same engine speed and fuel pump rack position as the corresponding transient cycles; however, as was discussed in Section 2.1, differentiation in air-mass flow-rate, hence, fuel–air equivalence ratio exists that affects accordingly the engine processes.

It is obvious how the increased fueling and boost pressure are gradually transformed into much higher cylinder gas pressures, hence exhaust manifold pressures and temperatures (not shown on Figure 2.26); maximum cylinder pressure increases from 57.5 bar (cycle No. 6) to 83.8 bar (cycle No.11). This is a 45% increase in peak cylinder pressure resulting in an equally high rate of bearings loading (Section 3.1) and engine construction stress. Relative air–fuel ratio assumes very low values during the depicted cycles, which is expected to increase smoke emissions.

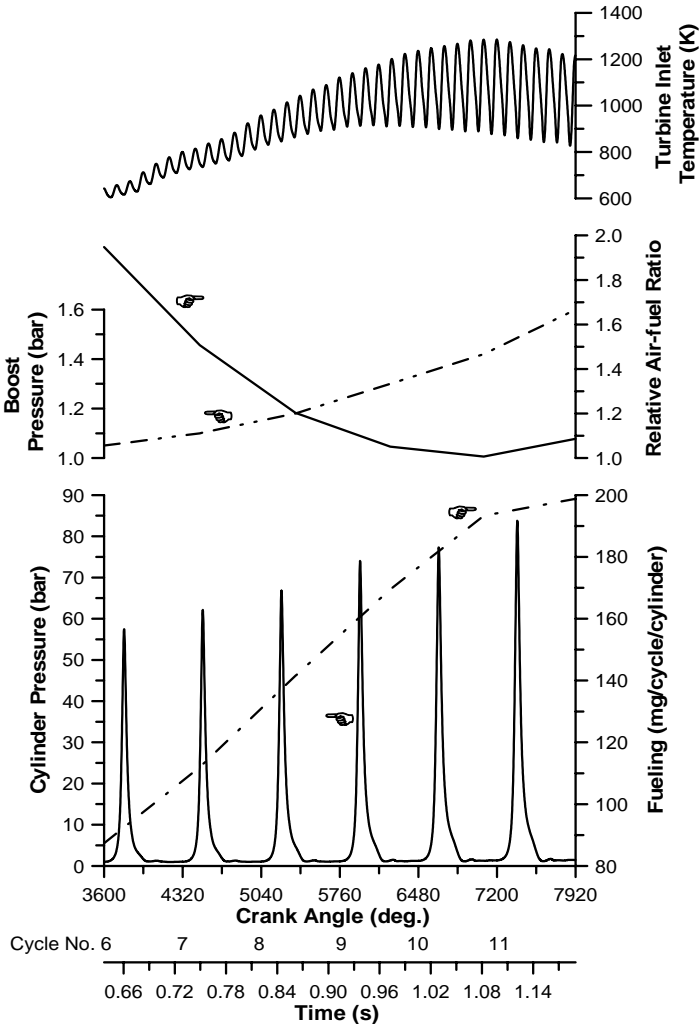


Figure 2.26. Development of various engine properties during cycles No. 6–11 after a 10–90% step increase in engine load (experimental and simulation results for a DI, turbocharged and aftercooled diesel engine with pulse turbocharging configuration and mechanical fuel injection pump without fuel limiter)

During the early transient cycles after a load or speed increase, the higher pressure, larger volume fuel jets are injected into an environment that is practically unchanged from the condition before the transient commenced, thus the higher-momentum fuel jet is not accompanied by enhanced gas motion. The rate of mixture preparation is, therefore, reduced and the heterogeneity of the mixture is increased. Figure 2.27 illustrates in the most explicit way the difference between steady-state and transient spray formation for a typical acceleration transient event compared with the respective steady-state conditions. Clearly, the rate of fuel vaporization during transients is much lower, being responsible for worse mixture preparation and higher rate of wall impingement [11,13].

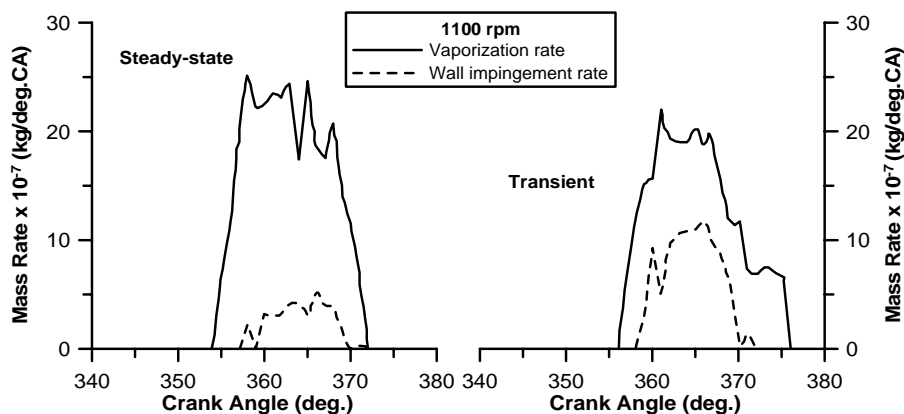


Figure 2.27. Comparison of the behavior of a spray jet during steady-state and transient conditions at the 1100 rpm operating point (experimental results reprinted with permission from Harndorf and Kuhnt [11])

In general, the following off-design phenomena are experienced during combustion of the early transient cycles

1. an influence in mixture formation due to air-deficiency (for the increased fueling) caused by turbocharger lag;
2. a change (increase) in the average fuel droplet diameter caused by the lower density and swirl compared with the respective quasi-steady conditions, which leads to increased jet penetration; over-penetration results in impingement of liquid fuel on the cooler cylinder surfaces, lowering mixing rate;
3. the lower end gas and wall temperatures in combination with a higher amount of end gas result in an increased ignition delay and hard combustion course, during the early cycles, where the turbocharger lag is more pronounced.

Winterbone and Tennant [18] worked on a six-cylinder, turbocharged diesel engine and analyzed pressure–crank angle data after load increase transients, using the Whitehouse–Way combustion model [19] (see also Section 9.3.2.3). This model

concerns a two-part expression; specifically for the preparation rate P (kg/°CA), it holds

$$P = K \cdot m_{fi}^{1-x} \cdot m_u^x \cdot p_{O_2}^y \quad (2.6)$$

with p_{O_2} the partial pressure of oxygen in the mixture, $m_{fi} = \int (dm_{fi}/d\phi)d\phi$ the mass of injected fuel up to the crank angle ϕ considered, with injection rate $dm_{fi}/d\phi$ given in Section 9.8, $m_u = m_{fi} - \int Pd\phi$ the mass of unprepared fuel, and K , x and y are constants derived after calibration against experimental data. This relation was initially derived on the assumption that the rate of preparation is proportional to the surface area of all the droplets having uniform diameter, while proportionality with respect to the droplets diameter has also been proposed. A variable value of the exponent x makes allowance for this fact and for the usually existing non-uniform droplets' diameter distribution. The last term on the right hand side of Equation 2.6 allows for the effect of oxygen availability on mixing. Winterbone and Tennant concluded from their experimental analysis of transient pressure diagrams that the combustion process was somehow deteriorated after a load increase. Further to the discrepancies discussed in the previous page:

4. Reduction of the injected fuel preparation rate constant 'K' of the Whitehouse–Way model (Equation 2.6) was observed, indicating that, other things being equal, the preparation proceeds more slowly. This can be attributed to the different evolution profile of injection rate (Figure 2.14), particularly to the shortening of the throttling period at the early stage of fuel injection that affects the premixed combustion phase.
5. A reduction in the exponent 'x' was also detected, which was attributed to poorer mixing owing, probably, to over-penetration resulting in deposition of the fuel on the piston (this can be attributed to the dynamic response of the fuel injection system as was discussed in Section 2.2.1). The reduction of exponent 'x' is then responsible for a reduction of the mass transfer rate between fuel and air and, hence, slowing down of the whole combustion process.

Figure 2.28 expands on the in-cylinder properties evolution of Figure 2.26, going one step further and focusing on the rate of heat release (RoHR) development during the same cycles of the 10–90% load increase. What is obvious during the early cycles of the transient event is a quite sharp rate of premixed heat release (for cycles No. 6 and 7); even in modern common rail systems with pilot injection, the above phenomenon cannot be completely eliminated.

As the boost pressure and injected fuel quantity increase, the peak cylinder pressures increase as well (Figure 2.26), while injection is advanced in order to allow optimal combustion of the larger fuel quantity. Hence, for the rest of the cycles depicted in Figure 2.28, a lower initial HRR peak and a higher duration of diffusion combustion are noticed, mainly, attributed to the slight increase in cylinder wall temperature as the transient event develops, and to the increased quantity of injected fuel.

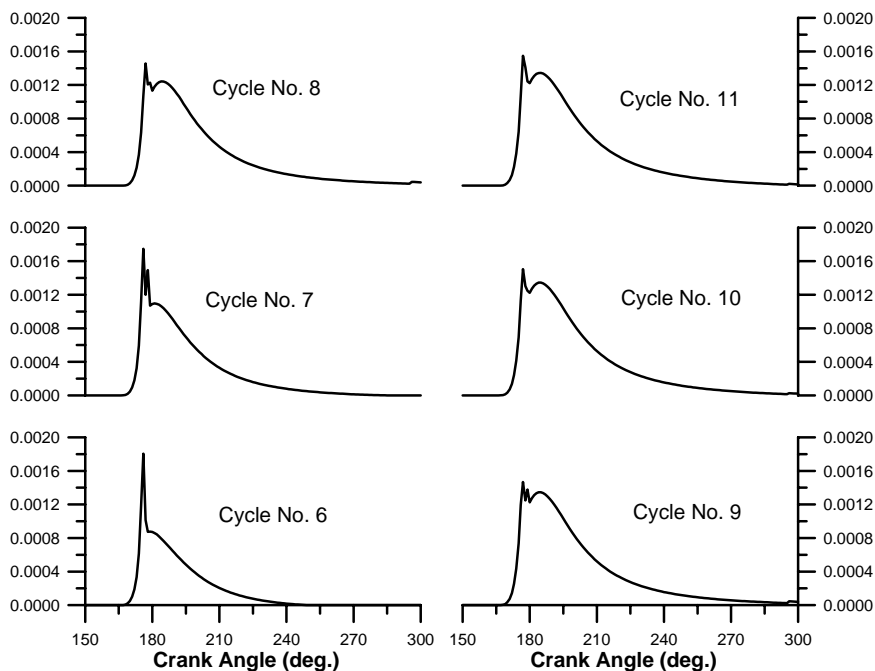


Figure 2.28. Rate of heat release (kJ/deg. CA) diagrams for cycles No. 6–11 of the 10–90% load increase transient event of Figure 2.26

The influential parameter that is responsible for the heat release behavior in Figure 2.28 is the ignition lag during the transient response. The ignition delay period is composed of a physical delay, incorporating liquid fuel atomization, vaporization and mixing, coupled with a simultaneous chemical delay process; the latter is the result of pre-combustion (decomposition and oxidation) reactions in the fuel–air mixture. It is the mixture preparation that is mainly responsible for the physical delay prior to ignition, with the air-charge pressure and temperature playing a primary role. Ignition delay has a direct impact on the intensity of heat release immediately after mixture auto-ignition. A relative reduction in the in-cylinder gas density due to turbocharger lag may cause less vigorous spray break-up and fuel–air mixing, thus resulting in increased ignition delay period and particulate matter emissions; measurements have documented this behavior, as is shown in Figure 2.28 and further quantified in Figure 2.29.

Consequently, the synergistic effect of higher fueling with still low wall temperatures and charge air pressure during turbocharger lag, ultimately, results in

- longer ignition delay period during the early cycles of the transient event; in fact, as is illustrated in Figure 2.29, the transient ignition lag is still higher than the respective steady-state conditions would dictate, even after 3 s through the specific 850–2000 rpm acceleration transient event, although a strong decreasing trend is established after the first cycles; and

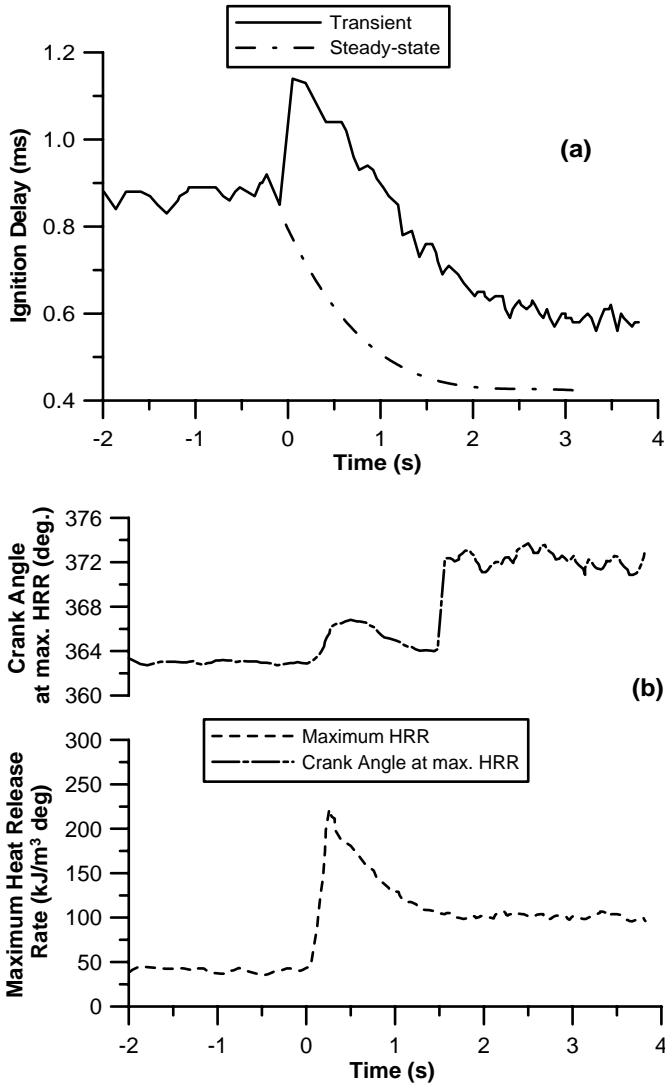


Figure 2.29. *a*: Comparison between steady-state and transient ignition delay; *b*: maximum heat release rate and crank angle at maximum HRR during an acceleration transient event (experimental results reprinted with permission from Harndorf and Kuhnt [11])

- significant and abrupt initial peak in the amount of premixed heat release.⁹ In general, the peak magnitude of the heat release rate during the premixed combustion phase directly correlates with the magnitude of the maximum

⁹ The duration of premixed combustion can be defined as the crank angle duration between the start of combustion and the minimum heat release before the start of diffusion combustion. Likewise, the duration of diffused combustion can be defined as the period from the end of the premixed phase up to the time where heat release crosses the zero line.

pressure rise. Therefore, the fact that during transients, turbocharger lag promotes longer ignition delay and a higher premixed combustion rate is reflected into an increase in the radiated (combustion) noise and smoke emissions as will be discussed in detail in Chapter 5.

Later in the transient, the distribution profile changes with the premixed peak being much smaller and the diffusion-burning phase becoming predominant; actually, diffusion burn fraction has been calculated to be at least 90% of the total during the late cycles of a transient event. The latter behavior is responsible for the gradual shift of the point of maximum HRR later in the cycle (upper sub-diagram of Figure 2.29b).

Ignition delay is typically defined experimentally as the time between start of (dynamic) injection and start of combustion; the former is determined as the start of injector needle movement, whereas the latter can be defined as the start of cylinder gas pressure rise (first or second derivative of gas pressure signal). Traditionally, constant volume vessels or bombs have been used for ignition delay measurements, ignoring the effect of piston movement. Steady-state ignition delay data are usually correlated by Arrhenius type equations of the form

$$\tau_{id} = C_1 p_g^{-C_2} \exp\left(\frac{E_a}{R_{mol} T_g}\right) \quad (2.7)$$

where τ_{id} is the ignition delay time in ms, p_g and T_g are the integrated mean gas pressure and temperature during the ignition delay, E_a is an apparent activation energy for the fuel auto-ignition process, R_{mol} is the universal gas constant, and C_1 , C_2 are constants depending on the fuel and the injection characteristics. Based on the fuel–air equivalence ratio Φ discrepancies during transients compared with steady-state operation, Assanis *et al.* [20] proposed an updated version of Equation 2.7 valid under transient conditions

$$\tau_{id} = C_1 p_g^{-C_2} \Phi^{-C_3} \exp\left(\frac{E_a}{R_{mol} T_g}\right) \quad (2.8)$$

By including the global equivalence ratio dependence, it was postulated that Φ is actually a measure of the probability of finding local pockets of fuel–air mixture within flammability limits for auto-ignition sites to be promoted. The adjustable constants in Equation 2.8 were fitted in order to minimize the least-square error between measured and correlated ignition delay, as well as to ensure that the latter would tend to zero at extremely high load. The ignition delay data were correlated as a function of the overall equivalence ratio and the mean pressure and temperature over the ignition delay interval. This ignition delay formula gave satisfactory results, mainly for medium to high engine speeds.

Summarizing the previous remarks, Figure 2.30 illustrates the various discrepancies/anomalies encountered during the early seconds of a turbocharged diesel engine load increase or acceleration transient event.

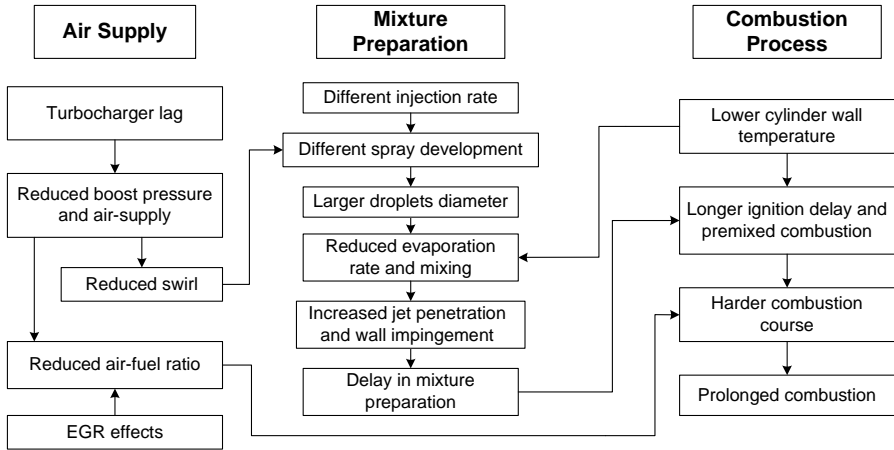


Figure 2.30. In-cylinder discrepancies during the early cycles of a turbocharged diesel engine transient event (adapted from Harndorf and Kuhnt [11])

With the gradual acceleration of the turbocharger, hence build-up of air-supply and delivery pressure (typically, after 15–30 cycles for an HSDI engine), combustion improves and the previously discussed anomalies diminish. At this point, (slight) retardation of the injection timing might be imposed by the engine ECU in order to control the already high gas pressures.

2.4 Variable Geometry Turbine

One of the main causes for the unsatisfactory transient response of turbocharged diesel engines lies in the fact that the matching between engine and turbocharger cannot be optimum throughout the whole engine operating range. Whereas internal combustion engines are designed to operate at wide air-mass flow range, the performance of turbo-machines is very much dependent on the respective gas angles and it deteriorates when moving away from their design point. Automotive diesel engines employ the pulse turbocharging configuration. Consequently, the turbocharger turbine is called upon to operate under non-steady flow conditions, with a two or even three scroll arrangement creating different instantaneous flow regimes in each sector. During an engine cycle, the variation in air-mass flow-rate is quite large, sometimes ranging from zero flow to choke; this variation depends on the number of engine cylinders and their disposition, degree of turbocharging, exhaust manifold geometry, *etc.* Since the instantaneous mass flow reaching the turbine blades varies substantially, the resultant gas angles will definitely depart from their design values, creating various losses, namely incidence and boundary layer separation. Moreover, the turbine practically operates in a highly transient manner even at steady-state engine operation, since the instantaneously high gas flow tends to accelerate it, whereas reduction in the air-flow produces the opposite

effect (Figure 2.31). From the above remarks, it is made clear that since the flow and operating characteristics of a turbocharger (turbine) are fundamentally different from those of the reciprocating engine, complex matching issues arise leading also to problematic transient response.

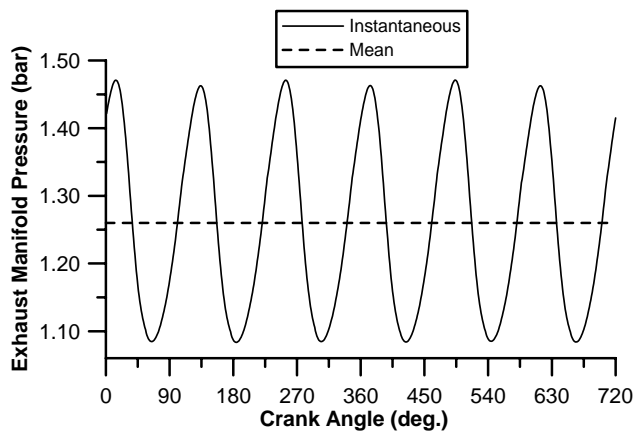


Figure 2.31. Instantaneous and mean exhaust manifold pressure during steady-state engine operation of a six-cylinder, pulse turbocharged diesel engine (simulation results)

Usually, engine operation is optimized at or around its design point/conditions. For example, an engine driving an electrical generator for base load operation will be optimized at full-load conditions and the turbine nozzle area will be typically selected for optimum performance at this operating point. When moving away from the design point however, *i.e.*, at part load operation, the turbine nozzle area will be too large for the reduced gas flow and temperature, reducing accordingly the turbine available energy. As a result of this behavior, shortage of air-supply to the engine cylinders will be established (particularly so if the EGR rate is kept at a high level for successful NO_x emissions control), as the power produced by the turbine will be reduced at a much higher rate.

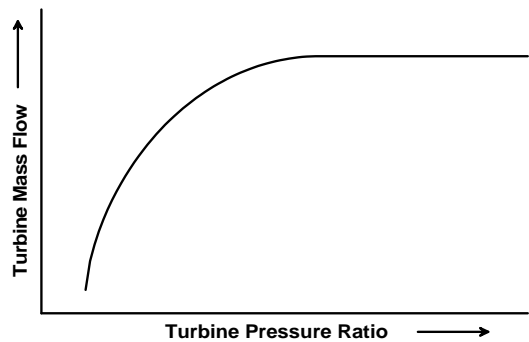


Figure 2.32. Typical curve of axial turbine expansion ratio vs. mass flow

This will deteriorate engine performance at steady-state conditions by lowering the allowable amount of injected fuel for smoke-free operation, thereby reducing bmep. Consequently, transient response will suffer too. The previously mentioned disproportionality emanates from the typical nozzle-type relation between turbine expansion ratio and mass flow that is depicted in Figure 2.32. Actually, it has been argued [5] that a linear increase in turbine expansion ratio with volumetric flow-rate would be ideal, but this cannot be realized with a conventional exhaust gas turbocharger of fixed geometry. Obviously, in order to accommodate the different operating requirements induced at part load operation, a reduction of turbine nozzle area would be needed.

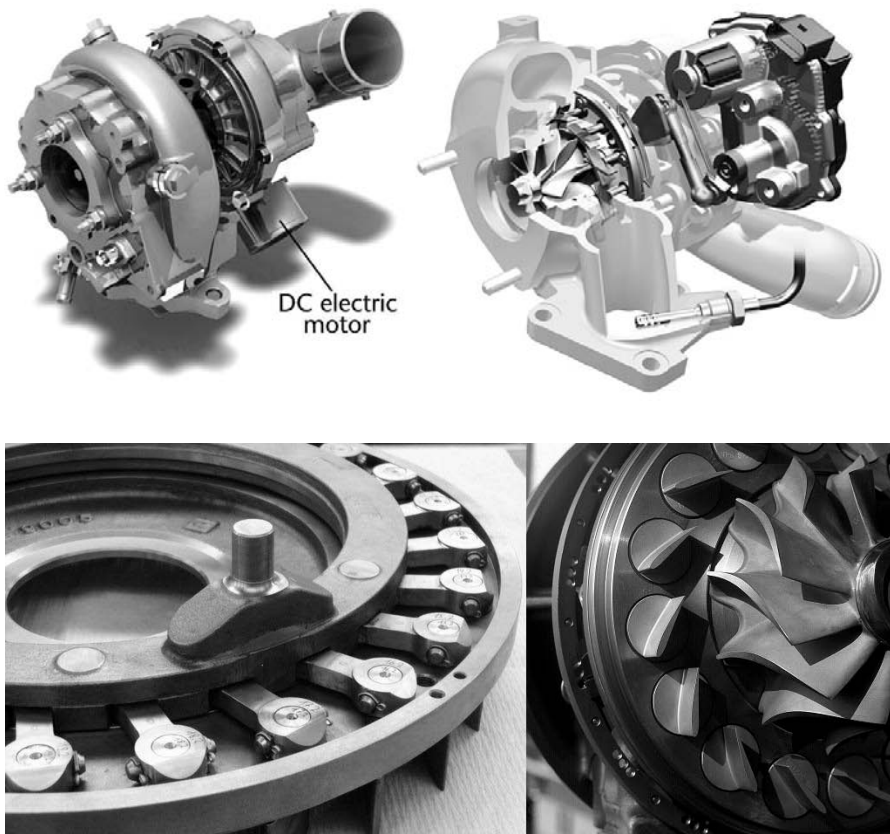


Figure 2.33. *Upper left:* Variable geometry turbocharger with electric vane actuation for automotive use (courtesy of Toyota Motor Europe). *Upper right:* Cutaway of automotive turbocharger showing the swing blade angle mechanism (courtesy of Audi AG). *Lower left:* VGT arrangement for radial turbine, featuring nozzle rings with adjustable vanes (courtesy of MAN Diesel); *Lower right:* Adjustable turbine nozzle blades/variable turbine geometry of an ABB TPS57 turbocharger; the clearances for the movable nozzle blades are reduced by springs that push the blades against the opposing casing wall (courtesy of ABB Turbo Systems Ltd)

On the other hand, optimizing engine performance for low-load operation (*i.e.*, by selecting a smaller turbocharger frame) would result in poorer efficiency overall as well as small speed range, over-boosting and turbine flow choking at high engine speeds and loads. It follows then that, ideally, a continuously variable turbine nozzle area would be needed to adjust the turbocharger characteristics to the specific engine operating point (preferably approaching the linear expansion ratio vs. mass flow-rate relation), hence maximize delivery pressure and air-supply to the engine cylinders. This holds particularly true for automotive applications, which are characterized by wide speed and load operating range as well as by frequent transient events; another potential application is high bmepp engines, where the increased rating enhances the previously discussed matching discrepancies between no-load and full-load conditions. The adjustment of the turbine nozzle area can be realized through a variable geometry turbine (VGT, Figure 2.33) that alters the whole turbine map by pivoting the swing blade angles or by keeping the nozzle blades fixed and changing the inlet area by moving the nozzle sidewalls. A variable geometry turbine can be considered, therefore, as a series of different turbine frames incorporated in a single unit, with variable flow characteristics but constant moment of inertia.

Variable geometry turbines are quite flexible in their operation and can prove beneficial in terms of engine performance at both steady-state and transient conditions; on the contrary, during engine cold starting, the gains obtained are much smaller. During steady-state operation, a small turbine nozzle area will increase the overall boost pressure level (this is primarily desirable at low-load conditions), establishing a less abrupt low-speed torque curve (Figure 2.34). By so doing, the low-speed, smoke-limited operating range of a turbocharged diesel engine can be significantly reduced. The overall higher air-mass flow-rate and air-fuel ratio will reduce smoke emissions, and the high exhaust gas pressure will ensure that a positive pressure difference exists between exhaust and inlet manifold supporting exhaust gas recirculation. When the engine speed and load are high enough, the blades are opened (nozzle area is increased) reducing gas velocities and preventing the excessive increase in boost pressure, which may harm the engine and lead to turbocharger over-speeding.

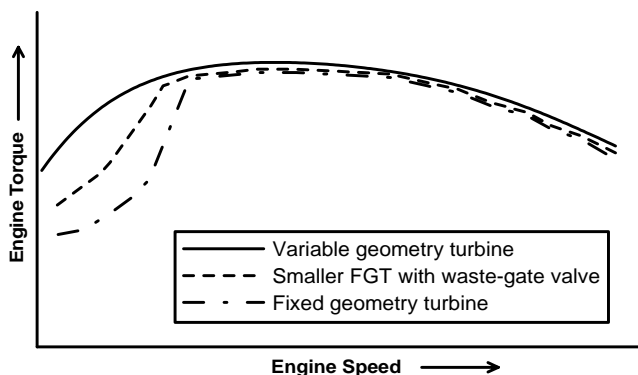


Figure 2.34. Improvement in low-speed torque from VGT operation compared with conventional (fixed geometry – (FGT)) or waste-gated turbocharging

A serious drawback that accompanies variable geometry turbine operation is the fact that the increased exhaust manifold pressure compared with its inlet manifold counterpart leads to back-flow of exhaust gas into the engine cylinders during the valve overlap period requiring a differentiated valve timing schedule. Moreover, pumping work is increased, worsening engine efficiency. An additional drawback is the, usually, lower turbine efficiency at the small nozzle areas established during low-load/speed operation.

An important aspect of VGT configuration is the need for a much more comprehensive and lengthy matching procedure compared with fixed geometry units; this has to take into account both steady-state and transient requirements for the various nozzle areas over the complete operating range, accounting also for EGR effects. Furthermore, a sophisticated closed-loop control system (see Appendix B for some fundamental aspects of control theory) that determines nozzle area according to the operating conditions (Figure 2.35) is needed [21–26]. This typically uses boost-pressure as the measured feedback output, as is demonstrated in Figure 2.36. At any operating point, the engine ECU uses the rotational speed and fueling values to calculate the desired boost pressure based on steady-state optimized maps. The aim of the controller is then to keep this pressure as close as possible to the mapped values applying, for example, P-I-D control, by relaying to the appropriate VGT vane position. A very instructive analysis in terms of VGT P-I-D control can be found in [27].

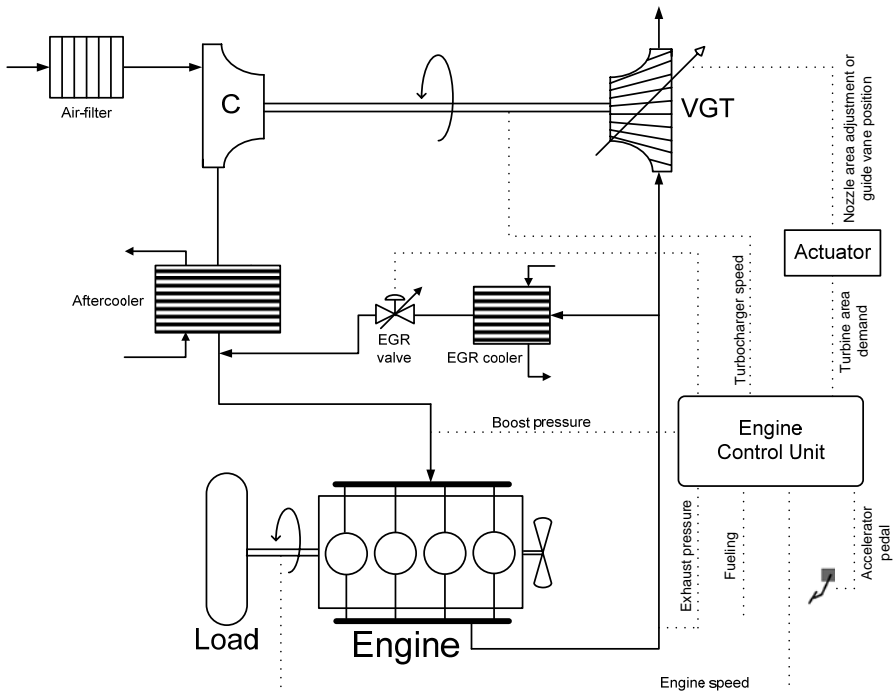


Figure 2.35. Schematic arrangement and simplified control diagram of variable geometry turbocharged diesel engine with exhaust gas recirculation

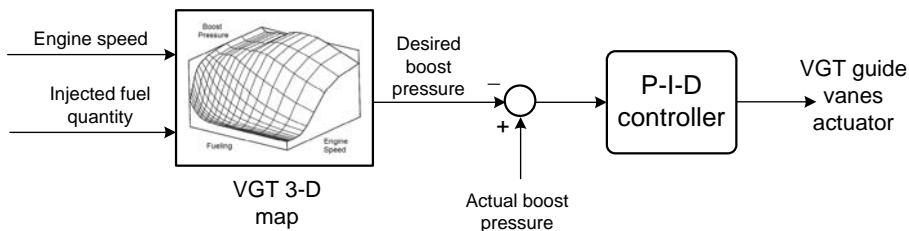


Figure 2.36. Simplified diagram of boost-pressure P-I-D, closed-loop controlled VGT system

Obviously, the response of the engine is strongly influenced by the specific VGT control system gain. Figure 2.37 illustrates how a relatively low gain, in a closed-loop strategy, results in smooth but rather long recovery period, in contrast to a high gain setup that responds almost instantly but leads to ‘hunting’ due to over-compensation. The latter behavior is typical when conventional P-I control algorithms are applied based on steady-state maps that often prove unable to handle the system constraints and delays. An optimization procedure is required taking into account both driveability and emission effects.

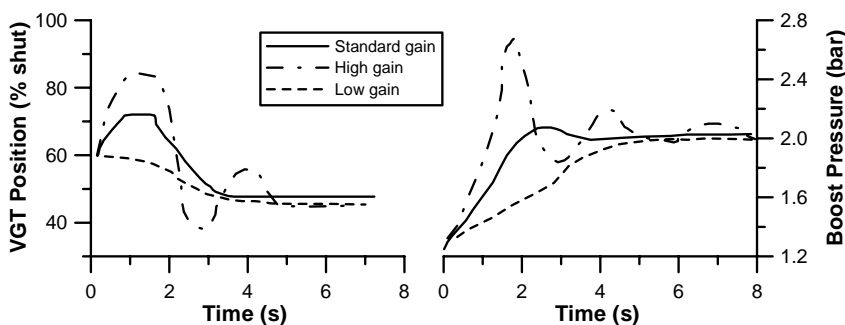


Figure 2.37. Effect of VGT gain on engine transient response at constant engine speed (simulation results adapted from Dekker and Sturm [28])

Historically, VGT control systems in low-bmep engines aimed to keep the VGT at its minimum area, increasing back-pressure and therefore turbine power at all times until the target boost pressure is reached. The VGT then opens to hold the boost at that value. Alternatively, speed dependent control strategies have been employed with smaller nozzle areas assigned the lower the engine speed. In recent years, increased global attention regarding exhaust emissions and the evolution of electronics have led to alternative VGT control strategies that are more suited to high-bmep engines; these usually focus on minimization of fuel consumption or exhaust emissions (among other things through accurate EGR control) or a combination of the above, depending on the specific engine operating conditions.

A possibly differentiated strategy is applied during transients, primarily aiming at turbocharger lag minimization. During transient operation (either load or speed

increase) the main strategy of the VGT is the reduction of the nozzle area by closing down the vanes so as to increase rapidly the back-pressure and enthalpy drop across the turbine, thereby boosting the compressor operating point (see also the conceptual diagram of Figure 2.38). By so doing, fast increase of engine air-supply is established, minimizing fuel limiting function and improving driveability and emissions. As soon as the air-supply to the engine has been built-up, the VGT vanes are gradually opened in order to prevent over-boosting. The positive effects of VGT during transients are, therefore, the result of improvement in flow characteristics and air-supply, whereas the dynamics (inertia) of the turbocharger are not altered.

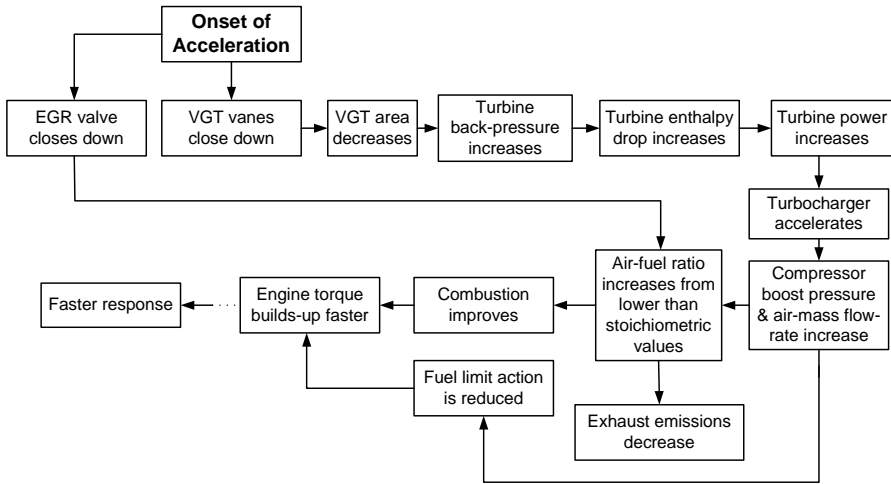


Figure 2.38. Series of events after onset of acceleration for a VGT diesel engine

Figure 2.39 describes in an explicit way the previously discussed operating mode of a variable geometry turbocharged engine during a transient event at constant engine speed. As soon as the driver applies the new fueling (corresponding to almost 27 mg/stroke), the air-fuel ratio decreases since the air-supply cannot instantly match the higher fueling due to the various system delays discussed in Section 2.1. The abrupt increase in fueling causes a step increase in desired boost pressure, hence a large error exists between demand and actual boost pressure. The VGT controller, responds to this error by closing down the VGT vanes. As a result, exhaust manifold pressure increases very fast, increasing accordingly the turbine expansion ratio and power. The increased power delivered to the compressor boosts its operating point and it is subsequently transformed into higher air-supply. Consequently, air-fuel ratio recovers much faster than with fixed geometry units and smoke emissions (not shown in Figure 2.39) are limited. Nonetheless, a brief peak in soot emissions cannot be utterly prohibited as the minimum value of AFR at approximately $t = 0.2$ s suggests. At about $t = 1.5$ s, VGT vanes are beginning to open in order to avoid over-boosting of the engine. What is missing from this figure is an indication of the engine efficiency during the

transient event. This has surely decreased due to the elevated pumping losses following the increased exhaust back-pressure. We should keep in mind though that it is very difficult to achieve optimization in both fast response and low fuel consumption for each turbine nozzle area. In any case, this disadvantage of VGT operation is more than offset during transients from the faster turbocharger and engine speed response (see also Section 6.4.4 for comparative VGT vs. fixed geometry turbine results).

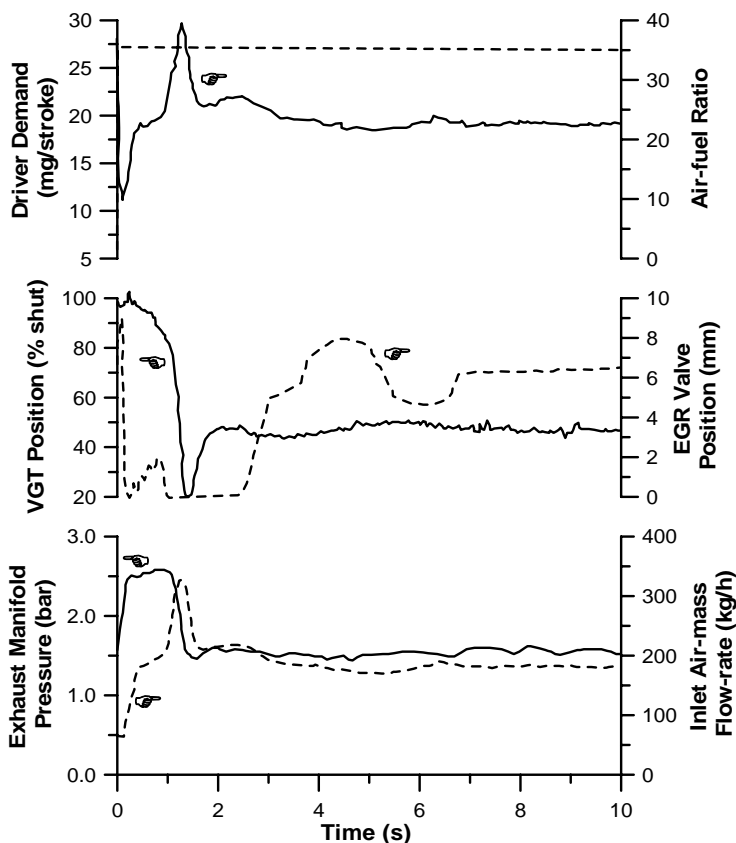


Figure 2.39. Engine and turbocharger properties development after a 10–170 Nm step load increase at 2500 rpm of a VGT engine (four-stroke, four-cylinder, 2 L displacement volume, HSDI, automotive diesel engine – experimental results reprinted with permission from SAE Paper No. 1999-01-0829 [23], © 1999 SAE International)

A case in point is that of pneumatically actuated VGT configurations. As the lower sub-diagram of Figure 2.39 suggests, a dynamic problem may arise, namely large overshoots in boost pressure and air-flow are experienced owing to hysteresis in the VGT vane mechanism. The vacuum at the vane actuator reduces steadily but the response of the vanes themselves is non-linear, which leads to a period of

stagnation followed by a sudden flip open. The stagnation causes an overshoot in pressure and air-flow, then the rapid opening leads to undershoot, settling only after several seconds.

2.5 Exhaust Gas Recirculation

The increasingly stringent emission regulations have rendered exhaust gas recirculation (EGR) quite popular in recent years as a successful means for reducing NO_x emissions from both spark and compression ignition engines. Introduction of cooled (exhaust) gas into the combustion chamber (see also the conceptual diagram of Figure 2.40), results in dilution of the air-charge by replacing O_2 with the non-reacting CO_2 . Consequently, both the specific heat capacity of the in-cylinder gas mixture and the peak flame temperatures of the cycle are reduced. As a result, NO_x emissions are reduced too, aided by the lower oxygen availability. However, smoke emissions increase since the soot oxidation process is diminished; the same holds true for HC and CO emissions.

The EGR system may be high pressure or low pressure. In the former case (Figure 2.40), the exhaust gas is drawn from upstream of the turbocharger turbine and is diverted back to the inlet side (typically downstream of the aftercooler); the recirculated exhaust gas flow is established from the pressure difference between the higher-pressure exhausted gas from the cylinders and the lower-pressure inducted air-supply. Since in this case, both EGR and turbine flow are driven by the exhaust gas, and, further, EGR flow reduces the amount of exhaust gas to be expanded in the turbine, there is clearly a strong relationship between the two flows that needs to be taken into account at the design stage in terms of optimum matching between engine and turbocharger (speed limits, peak cylinder pressure and surge avoidance). Nowadays, exhaust gas recirculation in automotive diesel engines is usually combined with a variable geometry turbine arrangement. In this case, the pressure difference between exhaust and inlet side for safe EGR flow can be determined by manipulating the VGT guide vanes; such a coupled EGR-VGT control system is discussed by Wijetunge *et al.* [29]. In general, interactions between EGR and other sub-systems, notably VGT and fuel injection, have been documented to have significant effects on the engine dynamic behavior, as will be discussed later in the section.

A low-pressure EGR unit, on the other hand, draws exhaust gas from downstream of the turbine or the DPF and diverts it to a point upstream of the turbocharger compressor; its main advantages are the somewhat cooled exhaust gas as well as the prevention of fouling of the EGR components, typical for the HP units exposed to the un-processed exhaust gases.

Besides, an EGR system can be internal or external; in the former case, the exhaust gas is re-routed back into the inlet manifold at the end of the exhaust stroke, for example through variable valve phasing. Most commonly, however, external EGR systems are employed, which have the added advantage of cooling down the exhaust, thus preventing decrease of the engine volumetric efficiency.

A typical, high-pressure, external EGR system, represented schematically in Figure 2.40, comprises the following components:

- EGR valve; this can be a continuously variable unit incorporating electric or pneumatic (or both) actuation;
- EGR cooler for cooling down the exhaust stream;
- EGR venturi mixer, where the exhaust gas is mixed with the incoming air-charge; such devices have been also applied for promoting gas flow from the exhaust to the inlet side;
- temperature and pressure sensors in the exhaust and inlet streams, and/or air-mass flow measurement, typically via hot wire anemometer; and
- comprehensive open- or closed-loop control system; this gathers the signals from the various sensors including engine and turbocharger, processes them based on look-up tables (steady-state maps) or model-based control theory, most probably applying correction factors to account for transient effects, and eventually determines the amount of exhaust gas to be recirculated through the EGR valve, *i.e.*, it determines the EGR valve position.

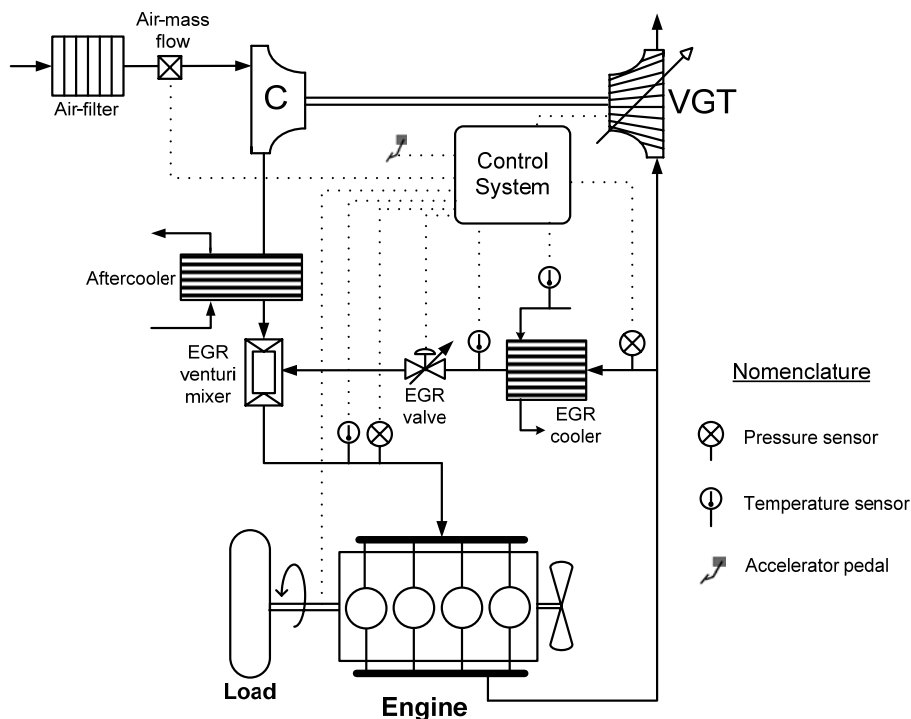


Figure 2.40. Schematic arrangement of an external, high-pressure EGR scheme with its simplified, closed-loop control system

In an open-loop EGR control system, the air–fuel ratio with the best NO_x reduction under the boundary condition of no smoke increase is mapped over injected fuel quantity and engine rotational speed; compensation is applied for high altitude operation, cold starting conditions (through the cooling water temperature) and transient conditions through a correction coefficient. However, during transients, feedback (closed-loop) systems are better suited to prevent excursion of the fuel–air equivalence ratio to high values and overshoot in smoke emissions. Guzzella and Amstutz [30] describe such a closed-loop EGR control system (Figure 2.41), where the measured air-mass flow-rate is used as the feedback property. Unlike in SI engines, recirculated exhaust gas in a diesel engine replaces oxygen, hence promotes enrichment of the mixture. During normal EGR mode, the intake manifold may be filled with 20 or 30% recirculated gas. If a rapid acceleration occurs, further deterioration of the air–fuel ratio and smoke emissions would be established, since the slow turbocharger response would delay increase of air-supply, whereas the smoke maps based on measured boost will fail because they assume that the mass-flow to the engine is all air. The closed-loop, feed-forward control system of Figure 2.41 describes how the net air-supply to the engine cylinders is estimated based on both EGR rate and total mass-flow to the engine (Equation 9.9) by using the measurement of the air-mass flow through the compressor. Based on the estimated effective air-mass flow to the engine, the maximum smoke-free, fuel injected quantity is then determined.

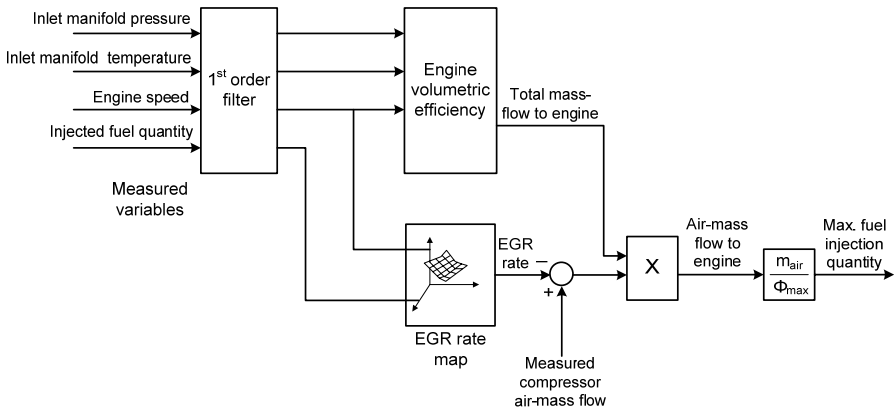


Figure 2.41. EGR calculation during transient operation (reprinted from Guzzella and Amstutz [30], © 1998 IEEE)

However, EGR flow is still largely controlled based on steady-state philosophy, *i.e.*, the EGR valve is kept open throughout the transient event. This eventually results in an extensive fuel limiting action (to avoid smoke emissions), delaying torque response and directly affecting driveability, up to the point where the turbocharger compressor boost pressure and the air-supply have been built-up satisfactorily. Since the slow turbocharger response induces quite different

operating requirements during transients compared with steady-state operation, it is doubtful that the above EGR control strategy can prove successful overall. This will be shown by the following comparative case study.

Figure 2.42 illustrates the response of a turbocharged diesel engine during a load acceptance transient event at constant engine speed by comparing two different EGR strategies; in the first case, the EGR valve remains open during the transient event (most probably incorporating a corrective function through the measured compressor air-supply, such as the one described in Figure 2.41, to avoid excessive smoke emissions), whereas in the second case, the engine ECU closes down the EGR valve during the first seconds of the transient event.

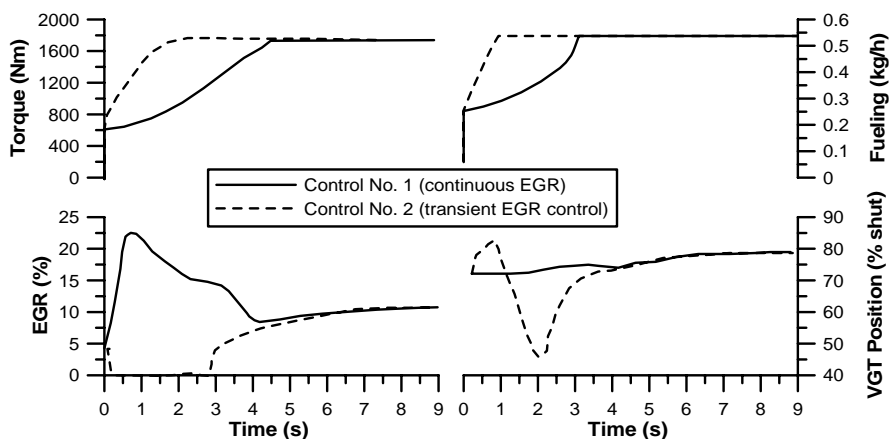


Figure 2.42. Engine response with continuous vs. transient controlled EGR valve during a load acceptance transient event at 1500 rpm (12 L displacement volume, VGT diesel engine, rated at 315 kW at 2000 rpm – simulation results adapted from Dekker and Sturm [31])

During the short period of time (3 s) where the EGR valve is closed down, the VGT guide vanes, being boost pressure loop-controlled, are directed to a more closed position in order to help build-up of turbine enthalpy drop. This ultimately results in much faster rise of turbocharger speed, compressor boost pressure (not shown in Figure 2.42), fueling and engine torque response (upper sub-diagram of Figure 2.42), hence driveability. The main mechanism behind the need for a drastic differentiation of the EGR strategy during transients is the deficiency of combustion air through the turbocharger lag phase. Even in an engine without EGR, the slow increase of boost pressure and air-supply during the early transient cycles results in a fuel limiting action to prevent excessive smoke emissions. Recall that recirculated exhaust gas in a diesel engine replaces oxygen promoting enrichment of the mixture. In an engine with EGR, if the strategy of Figure 2.41 (*i.e.*, continuous EGR) were followed, overshoot in smoke emissions would be prevented, but since the EGR valve is kept open, this would be accomplished only with an even more extensive fuel limitation, and driveability would suffer. Instead, the main engine control strategy in modern turbocharged engines aims at shutting

down the EGR valve as soon as a load or speed transient is detected (detection can be established through the accelerator pedal signal in Figure 2.40). By so doing, the primary gain is that the EGR-caused fueling restriction is diminished and engine torque can be built-up faster (still slower though than a naturally aspirated engine)¹⁰ with a penalty in nitrogen oxide emissions paid during this period, as will be discussed in Section 5.2. At the same time, closing of the variable geometry turbine vanes is initiated for better boost pressure build-up.

Returning to Figure 2.39 (in Section 2.4), we can now expand on the previously discussed remarks by highlighting specific aspects of the ECU control behavior in terms of coupled EGR and VGT operation during a load acceptance transient event. Initially, the EGR valve in Figure 2.39 is open, and then at the start of the transient it shuts rapidly in an attempt to maintain an adequate air–fuel ratio. Immediately after the valve closure, however, there will still be exhaust gas residuals present in the inlet manifold¹¹ – owing to the relatively large residual volume between the EGR valve and the combustion chamber – particularly so if the exhaust manifold pressure rises sharply, while the valve is closing due to the flow restriction of the closed VGT. The peak in exhaust manifold pressure following the closing of the VGT vanes causes a brief high EGR flow into the inlet manifold (the VGT control responds faster than the EGR valve closing), which in turn results in very low transient air–fuel ratios. Of particular note, here, is the brief opening of the EGR valve 3 s after initiation of the transient event; this is the result of the specific control strategy being derived directly from steady-state ECU maps. At the start of the transient event, the boost error exceeds acceptable limits and this in itself disables exhaust gas recirculation; however, the ECU still looks up the new EGR valve position demand based on the current fueling and speed conditions. As soon as the boost pressure error falls to within acceptable limits, control is handed back to the EGR map-based system, which in turn opens the valve because in the steady-state condition EGR is scheduled for the current engine speed and fueling [23].

An important aspect of the exhaust gas recirculation scheme applied is the magnitude of pressure difference between exhaust and inlet manifold. Obviously, the higher this difference (*e.g.*, by closing down the vanes in variable geometry turbine engines) the higher the exhaust gas flow through the EGR cooler, albeit, at the expense of higher pumping losses. An interesting comparison during both steady-state and transient conditions was conducted by Weber *et al.* [32] and a representative result from this work is reproduced in Figure 2.43. The low-pressure EGR unit (*i.e.*, exhaust gas drawn downstream of the turbine) managed to achieve the target brake mean effective pressure almost 0.9 s (or 40%) earlier than its high-pressure counterpart. It has been argued, however, that at higher power output, the high-pressure EGR loop can prove more efficient, mainly, in terms of fuel consumption.

¹⁰ Ironically, owing to the usually slow response of the EGR valve, the valve area restriction causes during the early cycles of a transient event an increase in the pressure difference, hence increase in the amount of recirculated exhaust gas.

¹¹ This exhaust gas residual may distribute unevenly to the engine cylinders resulting in cylinder-to-cylinder variation in terms of NO_x emissions.

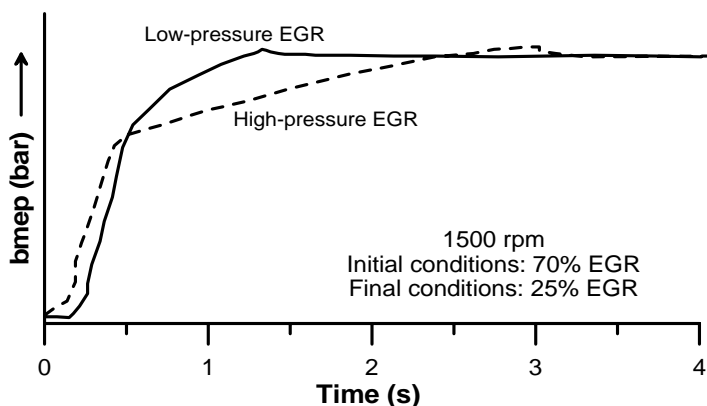


Figure 2.43. Effect of EGR strategy on transient performance of a VGT engine at 1500 rpm (simulation results adapted from Weber *et al.* [32])

Further aspects of exhaust gas recirculation and variable geometry turbine effects on transient operation will be discussed in Section 5.2, concerning NO_x emissions at the expense of smoke, and in Section 6.4.4, regarding engine and vehicle response throughout a Transient Cycle.

References

- [1] Winterbone DE. Transient Performance. In: Horlock JH, Winterbone DE (eds). The thermodynamics and gas dynamics of internal combustion engines, Vol. II. Oxford: Clarendon Press, 1986; pp. 1148–212.
- [2] Zinner K. Supercharging of internal combustion engines. Berlin Heidelberg New York: Springer-Verlag, 1978.
- [3] Watson N, Janota MS. Turbocharging the internal combustion engine. London: MacMillan, 1982.
- [4] Watson N. Transient performance simulation and analysis of turbocharged diesel engines. SAE Paper No. 810338, 1981.
- [5] Watson N. Eliminating rating effects on turbocharged diesel response. SAE Paper No. 840134, 1984.
- [6] Heireth H, Withalm G. The transient behavior of turbocharged SI engine. 18th FISITA Congress, Hamburg 1980 (in German).
- [7] Murayama T, Miyamoto N, Tsuda T, Suzuki M, Hasegawa S. Combustion behaviors under accelerating operation of an IDI diesel engine. SAE Paper No. 800966, 1980.
- [8] Takaishi T, Tateishi M, Kunimoto E. Dynamic response of fuel injection system of diesel engines. Institution of Mechanical Engineers, Seminar on 'Engine Transient Performance', Nov. 1990, pp. 5–12.
- [9] Arcoumanis C, Baniasad MS. Analysis of transient response of diesel fuel injection systems. Institution of Mechanical Engineers, Seminar on 'Diesel Fuel Injection Systems', 28–29 Sept 1995, pp. 99–113.

- [10] Catania AE, Dongiovanni C, Mittica A, Negri C, Spessa E. Study of automotive diesel injection-system dynamics under control. SAE Paper No. 962020, 1996.
- [11] Harndorf H, Kuhnt H-W. Improvement of transient behavior of turbocharged diesel engines through additional air injection in the turbocharger. MTZ 1995;56:20–8 (in German).
- [12] Benajes J, Luján JM, Bermúdez V, Serrano JR. Modelling of turbocharged diesel engines in transient operation. Part 1: insight into the relevant physical phenomena. Proc Inst Mech Eng, Part D, J Automobile Eng 2002;216:431–41.
- [13] Assanis DN, Filipi ZS, Fiveland SB, Syrimis M. A methodology for cycle-by-cycle transient heat release analysis in a turbocharged direct injection diesel engine. SAE Paper No. 2000-01-1185, 2000.
- [14] Woschni G, Doll M, Spindler W. Simulation of the stationary and transient performance of small high speed car diesel engines. MTZ 1991;52:468–77 (in German).
- [15] Rakopoulos CD, Hountalas DT, Mavropoulos GC, Giakoumis EG. An integrated transient analysis simulation model applied in thermal loading calculations of an air-cooled diesel engine under variable speed and load conditions. SAE Paper No. 970634, SAE Trans, J Engines 1997;106:923–39.
- [16] Heywood JB. Internal combustion engine fundamentals. New York: McGraw-Hill, 1988.
- [17] Benson RS, Whitehouse ND. Internal combustion engines. Oxford: Pergamon Press, 1979.
- [18] Winterbone DE, Tennant DWH. The variation of friction and combustion rates during diesel engine transients. SAE Paper No. 810339, 1981.
- [19] Whitehouse ND, Way RGB. Rate of heat release in diesel engines and its correlation with fuel injection data. Proc Inst Mech Eng, Part 3J 1969-70;184:17–27.
- [20] Assanis DN, Filipi ZS, Fiveland SB, Syrimis M. A predictive ignition delay correlation under steady-state and transient operation of a direct injection diesel engine. ASME Trans, J Eng Gas Turbines Power 2003;125:450–7.
- [21] Watson N, Banisoleiman K. A variable-geometry turbocharger control system for high output diesel engines. SAE Paper No. 880118, 1988.
- [22] Pilley AD, Noble AD, Beaumont AJ, Needham JR, Porter BC. Optimization of heavy-duty diesel engine transient emissions by advanced control of a variable geometry turbocharger. SAE Paper No. 890395, 1989.
- [23] Wijetunge RS, Brace CJ, Hawley JG, Vaughan ND, Horrocks RW, Bird GL. Dynamic behaviour of a high speed direct injection diesel engine. SAE Paper No. 1999-01-0829, 1999.
- [24] Filipi Z, Wang Y, Assanis D. Effect of variable geometry turbine (VGT) on diesel engine and vehicle system transient response. SAE Paper No. 2001-01-1247, 2001.
- [25] Bartsch P, Prenninger P, Allmer I. Transient performance optimization of turbocharged engines by means of gas exchange simulations. Institution of Mechanical Engineers, International Conference on 'Turbocharging and Air Management Systems', Paper C554/016, London, 1998, pp. 237–251.
- [26] Winkler N, Ångström H-K. Simulations and measurements of a two-stage turbocharged heavy-duty diesel engine including EGR in transient operation. SAE Paper No. 2008-01-0539, 2008.
- [27] Brace CJ, Cox A, Hawley JG, Vaughan ND, Wallace FW, Horrocks RW, Bird GL. Transient investigation of two variable geometry turbochargers for passenger vehicle diesel engines. SAE Paper No. 1999-01-1241, 1999.
- [28] Dekker JH, Sturm WL. Model based development of engine control algorithms. Institution of Mechanical Engineers, 3rd International Conference on 'Computers in Reciprocating Engines and Gas Turbines', Paper C499/021, 1996, pp. 163–72.

- [29] Wijetunge RS, Hawley JG, Vaughan ND. Application of alternative EGR and VGT strategies to a diesel engine. SAE Paper No. 2004-01-0899, 2004.
- [30] Guzzella L, Amstutz A. Control of diesel engines. IEEE Control Syst 1998;18:53–71.
- [31] Dekker HJ, Sturm WL. Simulation and control of a HD diesel engine equipped with new EGR technology. SAE Paper No. 960871, 1996.
- [32] Weber O, Joergl V, Shutty J, Keller P. Future breathing system requirements for clean diesel engines. Aachener Kolloquium, Fahrzeug- und Motorentechnik, Aachen, Germany, 2005.

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