A computational study of compressor surge during transient operation of turbocharged diesel engines

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Abstract: The transient response of turbocharged diesel engines is of great importance, especially when compressor surge is likely to occur. Turbocharger compressor surge is a dynamic instability mode that can generate air mass flow rate and pressure rise oscillations of large amplitude, and may also induce severe torsional loading to the turbocharger shaft. In this paper, an effort is made to study by simulation the effect of various parameters on the compressor surge behaviour. By varying these parameters, it can be revealed how compressor surge is affected and how surge intensity is influenced. To this aim, a simulation program developed is used for transient operation runs, which are critical for surge appearance. This has been applied on a six-cylinder, diesel engine of marine duty. The model has a control-design oriented philosophy and is capable of identifying the impact of each parameter on compressor surge. Thus, it can prove a useful tool to the engine designer.

Keywords: compressor surge; turbocharged diesel engine; transient operation; governor; inlet manifold; load change; parametric study.


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1 Introduction

The matching of a turbocharger to a diesel engine is a crucial task, as it is associated with the efficient and reliable operation of the power unit. Regarding compressor performance, it is essential that, for a wide range of engine operating conditions, the compressor functions within the region of its maximum efficiency. Moreover, for both steady-state and transient operation of the turbocharged diesel engine, the compressor operating point should lie as far as possible from the surge line, so that the surge margin is always adequate for avoiding compressor instability.

However, there are certain occasions when the engine operates under dynamic conditions, especially for engine performance close to or at full load, which can drive the compressor into its unstable region. At this point, simulation codes can be a valuable tool for predicting both engine and turbocharger compressor transient performance, since experimental and measuring techniques are usually costly and time-consuming. In fact, simulation tools, when used in parallel to some actual testing, can direct to specific experimental investigations. This way, the usually tedious and expensive (owing to the size of the power plant) test-bed measurements are greatly reduced.

For the simulation of the compression system, an initial approximate trial was attempted by Yano and Nagata (1971), who studied surge phenomena produced by the opening and closing of the cylinders’ scavenging ports. In a similar approach, Greitzer (1976a) developed a theoretical model that was afterwards coupled with experimental validation (Greitzer, 1976b). The results reported by that analysis concerned the time-dependent behaviour of a compression system of axial flow, which is more typical of gas-turbine installations. However, when either surge or rotating stall occurs, the developed model was generally accepted by later works as adequately representative of an engine air-charging system. Hansen et al. (1981) have shown that the model developed by Greitzer for axial compressors gave satisfactory description of surge oscillations in a centrifugal compression system too. The assumption in previous treatments of constant compressor rotor speed during surge was relaxed by Fink et al. (1992), by taking into account the compressor shaft speed dynamics, thus improving the model’s agreement with the observed surge behaviour.

Although several refined transient diesel engine simulation codes have been developed over the past years (Rakopoulos and Giakoumis, 2004, 2006a,b; Rakopoulos et al., 2004; Watson, 1984; Winterbone, 1986), few of them have been combined with models predicting turbocharger compressor surge. In one of these attempts, Tuzia et al. (1998) developed an engine simulation code, based on the ‘filling and emptying’ method, for investigating the transient performance of a sequentially turbocharged marine diesel engine.
engine. In a similar attempt, a zero-dimensional engine simulation code was utilised by Theotokatos and Kyrtatos (2001, 2003).

In general, transient operations responsible for surge initiation depend on various engine and turbocharger parameters specific to the current engine. These are mainly design or dynamic rather than thermodynamic parameters, such as for example the engine-load mass moment of inertia and the turbocharger mass moment of inertia, the constants that describe the operation of the speed governor, the duration of the transient load application and the volume of the engine’s inlet manifold.

It is therefore interesting to investigate the influence of these parameters on surge initiation and development; as such, a research can prove of great importance on the one hand to the understanding of the relevant phenomena and on the other to the armoury of the engine designer. Therefore, the main objective of the present work is the parametric study of the effect of various parameters on the compressor surge behaviour, filling an apparent gap in the open literature.

In a previous publication by the present research group (Rakopoulos et al., 2005), a model was developed predicting the time-dependent behaviour of the engine air-charging system when compressor surge occurs, in conjunction with a linearised, quasi-steady, diesel engine simulation code. The present work is based on that model. The authors opted to use a quasi-steady engine code in order to take advantage of the simplicity of such models, which moreover do not require an excessive number of input data. The implementation of such codes of limited complexity is justified by their capability of identifying key controlling variables and providing guidelines in order to determine trends of engine control design accompanied with adequate accuracy of predicted results.

The model is used for the parametric investigation and explicit diagrams will be presented, which show the effect of each variable parameter on compressor surge behaviour. It is within the main scopes of this study to define the really significant variables for avoiding surge generation or decreasing surge intensity.

The present theoretical and experimental (as regards steady-state and ‘normal’ transient operation) analysis was conducted using the geometrical and experimental data at steady-state and transient conditions of a six-cylinder, four-stroke, turbocharged and aftercooled, medium-high speed diesel engine of marine duty, located at the authors’ laboratory. It was decided to choose this specific engine, as detailed simulation analysis, accompanied with extensive experimental work, has already been conducted, both at steady-state and transient conditions (Rakopoulos and Giakoumis, 2004, 2006a,b; Rakopoulos et al., 2004). Besides, it is quite typical for the compressor of a turbocharged marine diesel engine to experience surge, as sudden changes of the engine loading may often occur, especially during ship manoeuvrings, driving the compressor away from its stable operating region.

2 Compressor system dynamic modelling

2.1 Engine air-charging model

The model for the compression system that was proposed by Greitzer (1976a) is adopted here; its equivalent arrangement is depicted in Figure 1. The centrifugal compressor is represented by an actuator disc with zero length, which accounts for the air pressure rise,
while the compressor outlet ducting is replaced by an equivalent pipe of constant area for simulating the air dynamics in the compressor passages and ducting. For the sake of simplicity of the model, the compressor inlet ducting is not included, thus ignoring compressor inlet temperature and pressure pulsations that are experienced during surge cycles (Greitzer, 1976b). Then, it is supposed that the air, which is supplied to the compressor impeller for raising its pressure, is constantly at ambient conditions. This assumption does not introduce significant errors to the analysis, especially when the compressor inlet is relatively short and of small volume (Theotokatos and Kyratatos, 2003).

Figure 1  Equivalent arrangement for compression system model

Regarding compressor performance, this is computed by its steady flow characteristics, provided by the compressor manufacturer and derived under steady-state conditions; the compressor pressure ratio is calculated using a two-dimensional data interpolation. The steady-state compressor map applies at all the engine operating conditions, while it is assumed that the compressor characteristics are not altered under dynamic operation. This assumption relieves the model of the necessity, incorporated in previous works (Fink et al., 1992; Greitzer, 1976a), to include a compressor transient response representation in order to simulate the lag noticed in compressor dynamic operation. Thus, it is implied that the compressor responds quasi-steadily to dynamic changes in mass flow rate and that the compressor characteristics do not depart from their steady state form, when the compressor functions under transient conditions. The compressor rotational speed during surge is variable and it is calculated from the turbocharger shaft speed dynamics. Application of the conservation of angular momentum on the turbocharger shaft gives the following equation:

$$ G_T C \cdot \frac{d\omega_{tc}}{dt} = \tau_t - \tau_c $$

(1)

with $G_{TC}$ the mass moment of inertia of the turbocharger and $\tau$ is torque (N m).

The compressor air duct is treated as a column of air and no consideration is taken of changes in velocity or pressure of airflow along the duct, at any instant of time. The length of this equivalent duct is determined by requiring that a given rate of change of mass flow produces the same unsteady pressure difference in the actual duct and in
the model, which yields the relation (the integral extends over the compressor and its outlet duct)

\[
\left( \frac{L_c}{A} \right)_{\text{model}} = \int \frac{dt}{A(t)}
\] (2)

while the constant flow-through area of the model duct can be matched with a characteristic area of the actual system. Here, \( l \) is length (m), \( A \) is compressor equivalent duct area (m²) and \( L_c \) is compressor equivalent duct length (m). This can be taken, in the case of a centrifugal compression system, as the compressor impeller inlet-eye area.

Assuming one-dimensional incompressible flow and applying the conservation of momentum equation, while neglecting pressure losses along the duct, it leads to the following differential equation:

\[
\frac{d\rho L_c}{dt} = p_c - p_m
\] (3)

where

\[
p_c = \rho r \cdot p_o
\] (4)

is the compressor discharge pressure, \( p \) is pressure (N/m²), and ‘\( \rho r \)’ is the compressor pressure ratio. This way, the inertia of the air contained inside the compressor impeller, diffuser, volute and outlet ducting is taken into account.

The compressor discharge plenum, which is the inlet manifold in the case of a turbocharged engine, is modelled via the ‘filling and emptying’ method. Only its volume is considered, with its geometrical profile dropped out of consideration, while pressure losses at its entrance are disregarded too. The equations that govern such a constant volume open thermodynamic system are the conservation of mass and the conservation of energy.

The conservation of energy in the inlet manifold reads

\[
\frac{dQ_{\text{in}}}{dt} = \frac{dW_{\text{in}}}{dt} + \frac{d(mu)_{\text{in}}}{dt} + \dot{m} \dot{h}_{\text{m}} - \dot{m} \dot{h}_c
\] (5)

with \( \dot{W}_{\text{in}} / dt = p_m \dot{V}_{\text{in}} / dt = 0 \) since this is a constant volume system, and \( \dot{Q}_{\text{in}} / dt = 0 \) since the heat transfer through the manifold walls is (as is the usual case in such models) neglected. Here, \( Q \) is heat (J), \( W \) is work (J), \( V \) is volume (m³), \( m \) is mass (kg), \( \dot{m} \) is mass flow rate (kg/s) and \( h \) is specific enthalpy (J/kg). The air is assumed a perfect gas, hence

\[
p_m \dot{V}_{\text{in}} = m \dot{R}_m \dot{T}_{\text{in}}
\] (6)

where \( R \) is specific gas constant (J/kg K) and \( T \) is absolute temperature (K). Moreover, it is assumed that the air has constant specific heat capacities, with specific enthalpy and internal energy set to zero at zero absolute temperature, that is, \( h_{\gamma=0} = u_{\gamma=0} = 0 \), which yields \( h_a = c_p a T \) and \( u_a = c_v a T \). Here, \( u \) is specific internal energy (J/kg), \( c_p \) is specific heat
A computational study of compressor surge during transient operation

capacity at constant pressure (J/kg K) and \( c_v \) is specific heat capacity at constant volume (J/kg K). Therefore, Equations (5) and (6) give

\[
\frac{dt}{dt} \left[ \left( \frac{p_{in} V_{in}}{R T_{in}} \right) c_v T_{in} \right] = c_p (m_c T_s - m_s T_{in})
\]  

(7)

Differentiating Equation (7) with respect to time yields, for the normal compressor flow operation,

\[
\frac{dp_{in}}{dt} = \frac{\gamma R}{V_{in}} \left( \dot{m}_c T_s - \dot{m}_s T_{in} \right) \quad \text{with } \dot{m}_c > 0
\]  

(8a)

In the case of reverse flow, we have

\[
\frac{dp_{in}}{dt} = \frac{\gamma R}{V_{in}} \left( \dot{m}_c T_{in} - \dot{m}_s T_{in} \right) \quad \text{with } \dot{m}_c < 0
\]  

(8b)

Finally, the application of the conservation of mass inside the inlet manifold reads

\[
\frac{dm_{in}}{dt} = \dot{m}_c - \dot{m}_s
\]  

(9)

2.2 Compressor characteristics to the left of the surge line

The estimation of the compressor characteristics for the regions to the left of the surge line is necessary for the present model in order to describe the oscillating surge cycles, especially when reverse flow occurs. A simple method is presented for the assessment of the whole compressor map, when only the useful operational range between the surge line and the choking region is known. It is pointed out that the following estimation procedure is justified in order to avoid recourse to be made to costly and time-consuming experiments on a compressor test bed. The latter approach is, however, the only unequivocally reliable method to acquire the whole compressor map.

According to this method, for the area between compressor zero flow and the surge line, a third-order polynomial equation is proposed, as for example by Hansen et al. (1981), in an attempt to adequately fit experimental data for a small centrifugal compressor. Moore and Greitzer (1986) used the following equation, based on a physically realistic choice among other available equations:

\[
pr = p_{in} + \beta \left[ 1 + \frac{3}{2} \left( \frac{m_{pu}}{\varphi} - 1 \right) - \frac{1}{2} \left( \frac{m_{pu}}{\varphi} - 1 \right)^2 \right]
\]  

(10)

Here, \( \beta \) and \( \varphi \) are compressor characteristic parameters. It is clear that the adoption of such an approximation may induce some kind of inaccuracy, the degree of which could only be estimated if the actual measured curves were available. However, the accurate representation of this part of the characteristic is not essential, as the compression system operating point passes through this zone very rapidly during surge (Hansen et al., 1981).
Parameters $\beta$ and $\varphi$ are explained in Figure 2, while $\text{pr}_0$ is the compressor pressure ratio at zero flow as stated in the paper of Botros (1994)

$$\text{pr} = \text{pr}_0 = \left[1 + \frac{\gamma - 1}{2\gamma R_s T_o} \omega_c^2 \left(r_2^2 - r_1^2\right)^{\gamma / \gamma - 1}\right]$$

(11)

where $r_1$ and $r_2$ are the compressor eye and impeller outer radii (m), respectively, $\gamma$ is ratio of specific heat capacities and $\omega$ is angular velocity (rad/s). The previous equation can be derived applying radial equilibrium theory for the compressor impeller blades and assuming isentropic flow processes (Theotokatos and Kyrtatos, 2001). The compressor eye radius is defined as the compressor impeller eye mean geometric radius, which divides the impeller eye area into two sections of equal area

$$r_e = \sqrt{\frac{r_{hi}^2 + r_{ti}^2}{2}}$$

(12)

where $r_{hi}$ and $r_{ti}$ are the impeller eye hub and tip radii, respectively.

**Figure 2** Compressor characteristic for the definition of parameters $\beta$ and $\varphi$

For the estimation of the negative flow branch of a centrifugal compressor characteristic, a second-order polynomial equation can be used (Hansen et al., 1981)

$$\text{pr} = \text{pr}_0 + 2\beta \left(\frac{m_{pu}}{2\varphi}\right)^2$$

(13)

with $m_{pu} = \dot{m}_c \sqrt{T_o / p_o}$. It is noted that this equation is adopted here because adequate fitting to experimental data was achieved by its use in previous works. Again, some degree of inaccuracy may be introduced at this part, which is also passed over for the sake of simplicity of the compressor map model.

### 2.3 Estimation of compressor torque

The developed model requires the calculation of compressor torque for each operating point of the compressor map. Neglecting heat losses and taking into account
the definition of compressor isentropic efficiency (Heywood, 1988), the following equation is derived

\[ \tau_c = \frac{30\gamma_a R_m C_o \dot{m}_c T_o}{\pi N_c (\gamma_a - 1)} \eta_c^{-1/\gamma_a} - 1 \] (14)

Therefore, in order to calculate compressor torque, we need to estimate the compressor pressure ratio and isentropic efficiency from its performance map, for certain values of compressor shaft speed and mass flow rate. Although the model estimates compressor pressure ratio using a two-dimensional interpolation of data from the digital representation of the compressor map, it would be further time-consuming to do so for the calculation of isentropic efficiency. Therefore, the usual compressor characteristics are converted into compressor torque parameter to compressor speed parameter.

For each point of the compressor stable region, the compressor torque is calculated using Equation (14). Then, the compressor torque parameter \( \tau_c/p_o \) is divided by the compressor speed parameter \( N_c/\sqrt{T_o} \), forming the variable \( \left( \tau_c/p_o \right) / N_c/\sqrt{T_o} \). Plotting the previous variable against the compressor mass flow rate parameter for all the lines of constant compressor speed parameter results in a straight line that can easily be fitted to the calculated points, as proposed by Winterbone (1986) and Winterbone et al. (1977)

\[ \frac{\tau_c/p_o}{N_c/\sqrt{T_o}} = a \left( \frac{\dot{m}_c \sqrt{T_o}}{p_o} \right) + b \] (15)

with a and b being constants, computed by the least squares fitting method. The values of the compressor mass flow rate parameter in the above relation extend from the choking area to the compressor zero flow point. For the reverse compressor flow operation, the even extension of Equation (15) is assumed to apply.

In Figure 3, values of the compressor torque parameter to compressor speed parameter against values of compressor mass flow rate parameter are presented, showing that a straight line can indeed successfully fit the resultant points, which were computed using the previously described method.

3 Description of the quasi-linear engine code

The quasi-linear model used is described in detail in a previous publication (Rakopoulos et al., 2005), so that only a brief outline will be given here. Nonetheless, it should be pointed out that although all the assumptions of the code are reasonably justified by its scope, some kind of inaccuracy is inevitably introduced to the predicted results. This is mainly due to the fact that the quasi-steady approximation simulates the various components of the system as if they pass successively from one steady-state operating point to another. The intermittent nature of the engine cycle is thus neglected and many cyclic oscillatory phenomena (intake and exhaust flows) are simplified through representation by time-averaged values. The mean-value simulation model used here is more event-based rather than time-based, as the time step of computations is much longer than that used in analyses based on a degree crank angle basis.
Figure 3  Compressor torque to speed parameter against compressor mass flow rate parameter

The air mass flow rate from the inlet manifold into the engine cylinders is calculated by the speed–density equation (Ferguson, 1986; Stone, 1999)

$$\dot{m}_a = \lambda \frac{p_{IVC}}{R_a} \frac{T_{IVC}}{z} \frac{N_e}{120}$$

(16)

where the scavenge ratio $\lambda$ is assumed to have a fixed value, $p_{IVC}$ and $T_{IVC}$ are the cylinder air pressure and temperature at inlet valve closing event, respectively, $V_t$ is the trapped volume per cylinder and $z$ is number of cylinders.

For the complete development of the engine code, engine temperature rise data must be provided from experimental work. It is reminded that engine temperature rise is defined as the difference of the temperature of the air inducted into the engine block from the temperature of the hot exhaust gas leaving the engine cylinders. This engine variable is strongly dependent on the overall air/fuel ratio, with engine speed having only a minor effect, and it is also based on engine design and configuration issues as described by Benson and Whitehouse (1979).

For the transient engine runs, an engine speed governor has been included in the model. Its role is to control the fuel pump rack position $q$, which, in turn, determines the amount of fuel mass injected into the engine cylinders in order to stabilise the engine speed around a desired point. The output of the governor is regulated using the PI (proportional–integral) control law with first order dynamics (Winterbone, 1986)

$$q_t = q_{t-1} + \left[ k_p \left( N_{e_{des}} - N_e \right) - q_{t-1} \right] \Delta t + \int_0^t k_i \left( N_{e_{des}} - N_e \right) dt$$

(17)
Here, \( k \) is PI control law constant and \( \tau_{\text{gov}} \) is speed governor time constant (s). The turbine of the turbocharger is modelled in the code using digital representations of the manufacturer’s data, derived under steady-state conditions. The turbine expansion ratio and isentropic efficiency are evaluated using the turbine swallowing capacity (or turbine mass flow rate parameter) and turbine shaft rotational speed, interpolating between given points of the turbine maps. The turbine torque is

\[
\tau_T = \frac{30\gamma_s R_s \dot{m}_t T_3}{\pi N_1 (\gamma_s - 1)} \cdot \eta_T \left[ 1 - \left( \frac{p_a}{p_3} \right)^{(\gamma_s - 1)/\gamma_s} \right]
\] (18)

Here, \( \eta \) is isentropic efficiency. In order to take into account the pulsating nature of the exhaust gas flow exiting the cylinders and entering the turbine blades (Benson et al., 1973), as the engine employs the pulse turbocharging system, the turbine expansion ratio is corrected by a pulse factor that is evaluated after experimental work from an empirical relation.

### 4 Experimental procedure

The experimental investigation was conducted on a six-cylinder, turbocharged and aftercooled diesel engine of marine duty (Rakopoulos et al., 2004). The basic data for the engine are given in Table 1.

<table>
<thead>
<tr>
<th>Table 1</th>
<th>Engine basic data</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Engine type</strong></td>
<td>6-Cylinder, 4-stroke, water-cooled, turbocharged and aftercooled, marine-duty diesel engine</td>
</tr>
<tr>
<td>Speed range</td>
<td>1000–1500 rpm</td>
</tr>
<tr>
<td>Bore/stroke</td>
<td>140 mm/180 mm</td>
</tr>
<tr>
<td>Maximum power</td>
<td>236 kW at 1500 rpm</td>
</tr>
<tr>
<td>Maximum torque</td>
<td>1520 N m at 1250 rpm</td>
</tr>
<tr>
<td>Moment of inertia</td>
<td>Engine+brake: 15.60 kg m²</td>
</tr>
</tbody>
</table>

Since the particular engine is one with a small speed range, mainly load changes (increases) with constant governor setting were examined. The engine is coupled with a hydraulic dynamometer. This is a variable fill brake, with the loading accomplished via the brake lever, which controls the amount of water swirling inside it. The application of the final load was affected by the movement of the brake lever, which in turn increased the amount of water inside the brake. However, this hydraulic brake is characterised by a high mass moment of inertia, resulting in long, abrupt and non-linear actual load-change profiles.

The overall matching between experimental and predicted transient responses was satisfactory for both engine and turbocharger values. A typical example is illustrated in
Figure 4 for a 10–50% load increase. This was a ‘normal’ transient operation case, where no surge was initiated. The degree of inaccuracy produced by the equations of the quasi-linear code is revealed. It is observed that the quasi-linear model’s response is faster than the experimental one for this particular transient case, as the governor reaches its maximum position earlier and so leading to a smaller speed droop. The difference in the quasi-linear results from the experimental ones is always less than 10%. A similarly acceptable agreement was reached for the other transient cases examined.

Figure 4 Experimental and predicted engine transient response to an increase in load

5 Results and discussion

5.1 Engine ‘nominal’ case for surging

Having gained confidence in the model’s predictive capabilities under ‘normal’ transient conditions, the examination of compressor surge behaviour was the next step. To this aim, a case of engine transient operation was simulated that was considered prone to inducing compressor surge, as it included very rapid changes of engine loadings. This case will be referred to from now on as the ‘nominal’ case.

The predicted engine and compressor variables that describe this simulated transient event are presented in Figure 5. The initial steady-state engine operating point corresponds to 96% of full load and 1200 rpm. The engine load is then reduced to 9% and soon increased again to 87% of the full load. Owing to the original engine speed increase, the engine airflow is increased and the compression system operating point moves towards the choke region. The predicted trajectory of the compression system operating point on the compressor map is also depicted in Figure 5, but in order to explain more clearly what is happening during the surge cycles, Figure 6 is introduced, which is
actually an enlargement of the bottom-right panel of Figure 5. After that, the load is increased, causing the rapid drop of engine speed, and the speed governor takes over in order to cause more fuel mass to be injected into the engine cylinders by the fuel pump. The relative delay though of the governor response, to adjust the engine torque to the load one, has as a result the continued engine speed droop, which at about 0.6 s reaches values even lower than 800 rpm. In this way, compressor surge starts. Finally, after almost 0.3 s from the surge initiation, while the governor still increases the injected fuel mass in order to meet the final demand of 1200 rpm at 87% of full load, the engine torque, and thus the speed and airflow too, are increased, causing the compression system operating point to move back again into the stable compressor region.

Figure 5  Predicted engine and compressor variables during a transient event initiating compressor surge
The path the compression system operating point undergoes is illustrated in Figure 6. The initial drift of the operating point towards the choke region due to the respective engine airflow increase is represented by the movement from point SS\textsubscript{i} (i.e. the initial steady-state operating point) to point A. During this movement, the inlet manifold pressure decreases as the engine torque decreases, the latter being larger than the respective load one, producing the corresponding engine speed increase. The subsequent steep load increase, which causes a relatively abrupt speed decrease, results in an equally large airflow decrease corresponding to movement from point A to point B, the latter being marginally to the left of the compressor surge line. From this point onwards, the oscillating surge loops are initiated, which consecutively evolve along the route B-C-D-E-B, with consecutively decreased values of inlet manifold pressures, as already depicted in Figure 5. After the final engine speed increase, which was caused by the engine torque increase, the air flow is increased just enough to move the compressor operating point to the right of the surge line, thus terminating the surge cycles. The final equilibrium steady-state point is represented by point SS\textsubscript{f} in Figure 6.

A deeper analysis of the surge cycles reveals that these can be divided into four different parts: E-B, B-C, C-D and D-E (Rakopoulos et al., 2005). Parts E-B (inlet manifold pressure building up) and C-D (inlet manifold discharging) last longer than the other two parts, determining the filling and emptying time, respectively, of the inlet manifold and defining in this way the surge cycle period. During these two parts, the compressor mass
flow changes gradually and the operating point moves mainly along steady-state points of the compressor characteristic. The other two parts of the surge cycle, that is, B-C and D-E, last much shorter; here the mass flow changes very rapidly and the operating point moves mainly along unsteady flow regions of the compressor characteristic.

As shown in Figure 5, during the compressor surge cycles, the torque absorbed by the compressor impeller oscillates rapidly, mainly due to airflow reversals, while the torque produced by the turbine presents a much smoother variation. In this way, severe torsional loadings are introduced to the turbocharger shaft. It is therefore evident that, in the case of compressor surge event, the air mass flow, inlet manifold pressure and turbocharger shaft torque fluctuations that are generated can be very harmful, both for the engine and the turbocharger operation. Of course, in the examined case, compressor surge lasted approximately only 0.3 s, so it cannot practically affect the engine safe running. However, if compressor surge occurs repeatedly, the engine and turbocharger secure running can be adversely affected.

Regarding the intensity of the surge phenomenon, it is noted that this can be characterised by its duration, $\Delta t_{\text{sur}}$, and the width and depth appearing during the surge cycles. The terms ‘width’ (‘delta compressor mass flow rate’, $\Delta \dot{m}_c$) and ‘depth’ (‘delta inlet manifold pressure ratio’, $\Delta p_{\text{im}}$) refer to the difference (amplitude), throughout the fluctuations generated by surge, between the maximum and minimum values of the compressor mass flow rate and inlet manifold pressure ratio, respectively. Additionally, a representative feature of the intensity of the surge phenomenon is the ‘delta compressor torque’, $\Delta \tau_c$, defined in a similar way to width and depth. In the following subsections, the effect of the various examined parameters on the four previously mentioned surge features ($\Delta t_{\text{sur}}, \Delta \dot{m}_c, \Delta p_{\text{im}}$ and $\Delta \tau_c$) will be exposed for the present engine configuration.

### 5.2 Effect of engine-load mass moment of inertia

Figure 7 illustrates the influence of the total engine-load mass moment of inertia on the compressor surge behaviour during the engine transient response. As the inertia is reduced, surge is experienced more severely since surge duration, width, depth and delta compressor torque are increased. This is due to the bigger engine speed droop that occurs after the abrupt engine load application, resulting in greater decrease of engine and compressor airflow. This is clearly shown on the sub-diagram that depicts the transient path of the engine air mass flow rate. For this reason, three representative values of the engine-load inertia have been examined, that is, 50, 125 and 150% of the nominal one. For values of the inertia higher than 1.5 times the nominal one, surge is totally eliminated, as the compressor air mass flow rate transient path is practically without any distinguishable fluctuation, indicated on the respective sub-diagram.

### 5.3 Effect of turbocharger mass moment of inertia

The influence of the turbocharger mass moment of inertia on the compressor surge is illustrated in Figure 8. For a clear representation, three cases are examined, namely, 50, 75 and 200% of the nominal value of the turbocharger inertia. The smaller the turbocharger inertia, the greater the possibility is for the compressor to avoid surge. On
the contrary, as turbocharger inertia increases, its transient response becomes slower, as it is depicted on the engine air mass flow rate sub-diagram. As a result of this lag, engine speed droops further, and so does the compressor air mass flow rate. Only when the inertia is reduced to 50% of its nominal value, the turbocharger of the present engine set-up manages to respond quickly enough, thus avoiding surge initiation.

Figure 7  Effect of engine-load mass moment of inertia on compressor surge

5.4 Effect of speed governor P-constant

Figure 9 examines the impact of the governor P-constant, \( k_p \), on the compressor behaviour during surge. This constant is also known as ‘the governor gain’, and appears in the numerator of the second term on the right-hand side of Equation (17). The difference in the surge pattern, generated by the governor gain variation, is depicted using three values, namely, 87.5, 125 and 175% of the nominal one. It is obvious that an
increase in the governor gain leads to surge elimination, as a consequence of the smaller engine speed droop. On the other hand, as its value decreases surge intensity is amplified, rendering the system more unstable. In particular, reduction of the governor gain at half its nominal value results in an almost doubling of the surge duration. Besides, excessive decrease in the governor gain value results in unacceptable pulsating engine response, coupled with a very long duration, until the final engine equilibrium point is reached. Values of $k_P$ higher than 150% of the nominal one keep the compressor operating point away from the surge area. It is noted that further increase in the governor gain does not contribute to the stability of the system, as the engine dynamics are changed over its whole operating range, rendering the engine unstable at low load and low speed (the most sensitive area) (Winterbone, 1986).

Figure 8  Effect of turbocharger mass moment of inertia on compressor surge
5.5 Effect of speed governor time constant

Figure 10 focuses on the effect of the speed governor time constant, $\tau_{\text{gov}}$ (Equation (17)), on the compressor surge. Three different values of the time constant, namely, 50, 75 and 125% of the nominal one, are examined. Decreasing $\tau_{\text{gov}}$ results in faster engine speed recovery with smaller speed droops. As a result, the engine and compressor air mass flow rates remain at values high enough to prevent surge. On the contrary, as the time constant increases, it takes more time for the engine speed to reach its desired value, while its transient fluctuation presents greater amplitude. As a result, the compressor operating point is driven into the detrimental surge area. It is also observed that surge duration increases as long as the time constant increases. It seems that, for the examined speed governor, values of the time constant less than 50% of the nominal one result in surge avoidance.
A computational study of compressor surge during transient operation

Figure 10  Effect of speed governor time constant on compressor surge

5.6 Effect of speed governor I-constant

The speed governor I-constant, $k_I$, which appears in the last term of the right-hand side of equation (17), was included for achieving zero final engine speed droop during transients. Its effect is illustrated in Figure 11 for three different values, namely, 50, 100 and 150% of the nominal one. As it is shown, this parameter has no traceable effect on the compressor surge behaviour. The only significant remark is that an increase of the I-constant results in an unacceptable pulsating engine transient response. The transient engine speed droop remains practically unaffected by the I-constant, and therefore compressor surge occurs independently of its value.
Figure 11  Effect of speed governor I-constant on compressor surge

5.7  Effect of load increase duration

The effect of the duration of load increase on compressor surge behaviour is exposed in Figure 12. The influence of three different values of load increase duration, namely, 50, 100 and 1000% of the nominal one, are depicted in this figure. The more rigid the load application is, the greater the possibility of compressor surge. The fast application of engine load increases crankshaft deceleration thus affecting the transient speed droop, resulting in low engine speed values. Therefore, the engine and compressor air mass flow rates reach low values too, driving the compressor into the surge area. It is observed that when the load increase duration is ten times the nominal one, the compressor surge is marginally avoided.
5.8 Effect of inlet manifold volume

Finally, Figure 13 illustrates the effect of the engine’s inlet manifold volume on compressor surge. Three values of the inlet manifold volume, namely, 100, 300 and 500% of the nominal one, are examined here. The size of the inlet manifold does not affect compressor surge initiation for the current engine set-up. Moreover, the volume of the inlet manifold affects only the surge oscillations frequency, becoming lower with volume increase. This is due to the fact that in case of volume increase, the surge loops need longer time in order to be developed, as the inlet manifold needs more air to be filled and thus more time. It is also observed that a decrease in inlet manifold volume results in greater surge depth.
6 Conclusions

A model predicting the transient response of a turbocharged diesel engine compression system during surge has been developed and incorporated into a linearised quasi-steady engine code. The simulation program was used for studying the effect of various parameters on compressor surge behaviour. The developed model, having a control-design orientation, is applied for the simulation of a marine diesel engine transient response, subjected to abrupt engine loading schedules. The following conclusions are derived concerning the present engine-load configuration bearing also in mind the various assumptions and limitations introduced and explained in the main text regarding the modelling of the engine and of the compressor map to the left of the surge line:

1. The engine-load mass moment of inertia has a significant effect on the compressor surge behaviour. In order to avoid generation of compressor instability, increased values of inertia are favoured. However, a compromise has to be made since this will have a negative impact on the desirable engine response rate.
A computational study of compressor surge during transient operation

2 A small turbocharger mass moment of inertia leads to elimination of compressor surge, as the well-known unfavourable turbocharger lag is reduced and the turbocharger response becomes faster. In this way, the engine’s transient performance is improved too.

3 The speed governor gain (P-constant) must be increased, in order to avoid compressor surge. However, its increase should not be excessive, as this would destabilise the engine performance at the low-load and low-speed operating range.

4 The speed governor time constant has a direct effect on the engine transient response, affecting compressor surge behaviour too. Reduced values of this constant are desired to prevent surge initiation.

5 The speed governor I-constant does not affect compressor surge. This constant is related only to the engine transient response, rendering it quite unstable if increased.

6 The duration of the load increase affects the compressor behaviour seriously under surge conditions. Abrupt load applications are generally not desired, as they can result in steep engine speed droops and therefore lead to compressor surge.

7 The volume of the engine’s inlet manifold affects only the frequency of the surge cycles, becoming lower with volume increase. The occurrence of compressor surge or not, does not depend on the size of the inlet manifold.

Acknowledgements

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Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>$A$</td>
<td>Compressor equivalent duct area, m$^2$</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat capacity at constant pressure, J/kg K</td>
</tr>
<tr>
<td>$c_v$</td>
<td>Specific heat capacity at constant volume, J/kg K</td>
</tr>
<tr>
<td>$G$</td>
<td>Mass moment of inertia, kg m$^2$</td>
</tr>
<tr>
<td>$h$</td>
<td>Specific enthalpy, J/kg</td>
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<tr>
<td>$k$</td>
<td>Proportional–integral control law constant</td>
</tr>
<tr>
<td>$l$</td>
<td>Length, m</td>
</tr>
<tr>
<td>$L_c$</td>
<td>Compressor equivalent duct length, m</td>
</tr>
<tr>
<td>$m$</td>
<td>Mass, kg</td>
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<tr>
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<td>Mass flow rate, kg/s</td>
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<tr>
<td>$N$</td>
<td>Rotational speed, rpm</td>
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<tr>
<td>$p$</td>
<td>Pressure, N/m$^2$</td>
</tr>
<tr>
<td>$pr$</td>
<td>Compressor pressure ratio</td>
</tr>
</tbody>
</table>
Fuel pump rack position, mm
Heat, J
Radius, m
Specific gas constant, J/kg K
Time, s
Absolute temperature, K
Specific internal energy, J/kg
Volume, m$^3$
Work, J
Number of cylinders

Compressor characteristic parameter
Ratio of specific heat capacities
Isentropic efficiency
Scavenge ratio
Torque, N m
Speed governor time constant, s
Compressor characteristic parameter,
Angular velocity, rad/s

Conditions at compressor zero flow
Turbine inlet
Turbine exit
Air
Compressor
Desired
Engine
Gas
Inlet manifold
Increase
A computational study of compressor surge during transient operation

I I-constant
IVC Inlet valve closing
o Ambient
P P-constant
sur Surge
tot Total
tr Trapped
T Turbine
TC Turbocharger

Abbreviation

PI Proportional–integral

References


