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ABSTRACT

The use of variable geometry turbocharging (VGT) as an aid to performance enhancement has been the subject of much interest for use in high-speed, light-duty automotive diesel applications in recent times (4). One of the key benefits anticipated is the improved transient response possible with such a device over the conventional fixed geometry turbine with wastegate.

The transient responses of two different types of variable geometry turbocharger have been investigated on a dynamic engine test bed. To demonstrate the effect of the turbocharger on the entire system a series of step changes in engine load at constant engine speed were carried out with the turbocharger and exhaust gas recirculation (EGR) systems under the control of the engine management microprocessor.

Results are presented which compare the different performance and emissions characteristics of the devices. Some control issues are discussed with a view to improving the transient response of both types. Of particular importance is the interaction between the turbocharger system and the EGR system.

INTRODUCTION

In order to satisfy the requirements of increasingly severe world wide emissions control legislation (1,2) modern drive by wire Diesel engines are fitted with a large number of control devices allowing a wide range of engine characteristics to be altered during operation (3). In addition to the primary control variables of fuel quantity and injection timing the engine controller also actively sets EGR level, boost pressure and swirl ratio.

The flexibility offered by these systems needs to be carefully controlled so as to maximize the benefits to emissions and fuel economy while at the same time enhancing drivability, particularly during transients. Typically much engine calibration work is carried out on steady state engine dynamometers with only limited transient validation.

Transient testing is an important part of a systematic engine development program and is particularly useful in checking the action of the many control algorithms in use on a modern engine.

A series of tests have been performed on a dynamic test rig at the University of Bath. The aim was to investigate the response of the various engine systems, particularly that of two alternative types of VGT to step pedal inputs. A selection of the results are presented which highlight the relatively rich excursion following a step up in pedal from low to high load. The effect of the EGR system is crucial to the successful action of the VGT system as their behaviors are mutually dependant.

ENGINE HARDWARE AND CONTROL STRATEGY

The engine used was an experimental direct injection Diesel engine equipped with turbocharger and intercooler. Tests were carried out using two different variable geometry turbochargers differing only in the turbine assembly used. Turbine A used a swiveling vane design. Turbine B used a sliding nozzle arrangement. Both turbines were vacuum actuated under the PWM control of the engine management ECU. The EGR system also used a vacuum operated valve under the PWM control of the ECU. Inlet throttling and swirl control by port deactivation were available for the engine but not used in this work.

The control strategy was based on experimental code and used the well proven open loop with three term feedback correction method to achieve good, stable control in steady state through careful calibration and testing. The turbine and the EGR system were controlled separately except for some linkage for fault handling. The turbine was used to control the inlet manifold pressure according to a steady state map of demanded inlet manifold pressure against engine speed and fuelling. With knowledge of engine characteristics a further map was produced which set the open loop PWM demand for the turbine control system. This open loop path was designed to achieve approximately the correct boost pressure, any error being sensed by a pressure sensor mounted off the inlet manifold and used by a closed loop PID controller to trim the PWM demand.

Essentially the same strategy was used for both turbochargers although they operated in the opposite sense mechanically. Turbine A reduced its effective flow area (increasing boost) on the application of vacuum to its actuator. Turbine B increased its flow area with increasing vacuum.

The EGR valve was used to control mass air flow (MAF) through the engine in a similar manner, here the feedback loop being closed by the signal from a hot film mass air flow meter mounted upstream of the inlet air filter. The EGR rate was set implicitly by setting the demanded MAF map to the level shown experimentally to deliver the optimum EGR rate at each engine speed and load. The mass flow through the air filter is reduced as the EGR valve is opened further to allow more flow to pass from the exhaust to the inlet manifold.

The control strategy described above is simplified for clarity. There are numerous modifiers to the behavior of the system to account for effects such as temperature, altitude and so on. Controller gains were also mapped to enable changing performance at different operating points. The tests described below attempted to analyze the suitability of the strategy for effective transient control. Of particular interest was the case where a rapid increase in driver power demand results in a need for rapidly increasing fuelling and thus MAF and boost pressure and a rapid decrease in EGR fraction to allow clean combustion at the new operating point.

TEST PROCEDURE

Tests were conducted on a dynamic engine and transmission test rig as part of a larger investigation into the drive cycle performance of the powertrain although a simpler engine only rig would have been sufficient for the work described here.

The dynamometer was operated in speed control mode with the engine speed set to 2500 rev/min. An initial pedal setting corresponding to 10Nm engine torque was selected. Sufficient time was allowed for engine variables to stabilize, including saturation of the integral term of the boost pressure (VGT) controller where appropriate. A second pedal potentiometer was then selected using a changeover switch to move to a condition previously set to deliver 170Nm engine torque.

Data were recorded at 25Hz on the test cell acquisition computer and 50Hz with the fuel pump calibration software to capture controller variables.

For tests with no EGR required the system was disabled by removing setting both the maximum and minimum EGR valve PWM duty cycles to zero within the calibration software. It should be noted that where EGR was used the EGR levels selected were higher than those normally used in a production calibration, and on the limit of acceptable drivability. This was in order to highlight the differing controller responses in differing extremes of operation.

A matrix of tests was performed, four of which are reported here, they are summarized in **table 1** below.

Table 1. tests performed

Test	Turbocharger	EGR
1	A	Off
2	A	On
3	B	Off
4	B	On

Additionally, a set of tests aimed at investigating the step response of the turbines and the vacuum system were carried out. Here the engine was operated at 2500 rev/min and 10Nm while the turbine PWM duty cycle was manipulated using the engine calibration software. A step up from minimum to maximum demand was followed by a step back down to minimum demand.

RESULTS

STEP RESPONSE OF TURBINES – **Figure 1** shows the results of the step changes in turbine demand. Table 2 below summarizes the delay and rise times of the vacuum system and turbine position.

The time taken for the vacuum to rise to 90% of its final value (T_{90}) for turbine A was less than that for turbine B. This difference is due solely to the larger actuator volume on turbine B. The rest of the vacuum circuit was unchanged. For the corresponding step down in vacuum demand the T_{90} for turbine A was again less than that for turbine B although both times were much shorter than the rise times as the actuator was vented to atmosphere, leading to very fast response.

Table 2. Vacuum and Turbine position response times

	Vacuum T_{90} (s)		**Position T_{delay} (s)**		**Position T_{90} (s)**	
Turbine	**A**	**B**	**A**	**B**	**A**	**B**
Step Up	0.36	0.56	0.08	0.04	0.38	0.57
Step Down	0.12	0.14	0.04	0.00	0.08	0.14

The delay between the start of change of vacuum and the start of actuator movement was small in all cases, undetectable at the 25Hz sampling frequency used in the case of decreasing vacuum for turbine B. The T_{90} for the turbine position rapidly followed the change in vacuum, suggesting that the actuators are well sized in relation to the operating force required. There may be some scope for decreasing actuator volume in order to speed up response. One important feature of this test is that since the actuation mechanisms of the turbines are operated in opposite directions the vacuum rise time of turbine A must be compared to the vacuum decay time of turbine B and vice versa. Hence the time to move turbine A fully in the direction of increasing boost is 0.46 seconds (T_{delay} + T_{90} for turbine position) but only 0.14 seconds for turbine B. This could have significant effect on the system performance since time to increase boost is crucial during a step increase in power demand. The corresponding increased time taken to open up turbine B and reduce the boost pressure is unlikely to be problematic.

LOAD STEPS

Engine response – Data from the step changes in load are presented in **figures 2 to 5**. Each figure has a number of variables plotted against time for the two cases, EGR off to the left of the page and EGR on to the right. **Figure 2** shows at the top (**Figure 2a**) the fuel demand as seen by the fuel injection pump after drivability filters have modified the raw demand from the driver's pedal. The fuel levels differ slightly between tests due to small differences in turbine efficiency and normal experimental variability.

The second pair of graphs (**Figure 2b**) show the torque produced by the engine. The engine torque produced using turbine A is quite similar to that using turbine B. The most significant difference is between the EGR off tests and those with EGR on. Although the fuelling was only slightly delayed by boost limits the torque produced with EGR on is significantly delayed, full torque being produced some time later than the previous case and the rate of change being significantly slower. This behavior is likely to be noticeable to the driver and is a consequence of the high EGR levels used.

The third pair of plots (**Figure 2c**) shows the boost (or inlet manifold) pressure. Here the first significant differences between the two devices can be seen. The tests using turbine A exhibit some undesirable characteristics in both cases. Without EGR the boost level fails to reach a steady level for some four seconds following the pedal step. In the EGR on case there is a large initial overshoot in boost pressure lasting over one second followed by further oscillations of lower magnitude. In contrast the tests with turbine B are much more satisfactory. A steady boost pressure is achieved within one second in the EGR off case and only slightly longer once EGR is introduced. There is a slight drop in boost immediately following the step but not so large as to be serious.

The same behavior can be seen in the MAF data (**Figure 2d**). Tests with turbine A show large delays in achieving a steady mass flow The non EGR case is not a fair test of the MAF control system as the only means of controlling MAF is via the EGR valve, which is closed. Here the mass flow through the engine is simply the result of the pressure in the inlet manifold and the engine's natural characteristics. The MAF is hence clearly linked with the boost and not under the control of the ECU. When EGR is used, however, there is a large overshoot in mass flow with turbine A, suggesting that the inevitable physical interactions between of the two loops are insufficiently decoupled by the control strategy in this case. The test using turbine B exhibits better performance although there is still a moderate overshoot of around two seconds duration.

Turbocharger control variables – To explain some of the reasons for this behavior various control variables were logged. Those related to the control of the turbine are presented in **figure 3**. The first pair of plots (**Figure 3a**) shows the demanded boost pressure. The sharp step in the graphs is due to the relation of boost pressure demand to engine speed (not varying here) and fuel demand. The demands are similar in all four cases. The second pair of plots (**Figure 3b**) show the open loop demand to the turbine vacuum regulator. Turbine B had only recently been fitted and hence had a fixed open loop value of 50% until sufficient data were available for a realistic calibration. The PID part of the structure is therefore performing all of the control. The resulting boost error is shown in the next pair of plots (**Figure 3c**). Notable features are the sustained error in the EGR off case for turbine A and the negative error when EGR is used with the same turbine corresponding to the overshoot described above. Also significant is the small but steady error in initial boost pressure for both turbines when EGR is used. This is a consequence of the high levels of EGR used. Neither turbocharger can maintain the demanded boost with a high EGR flow at such low exhaust manifold pressures. Turbine B is worse in this regard as it has a slightly larger effective area.

The fourth pair of plots (**Figure 3d**) show the proportional term of the controller. This is, as expected, proportional to the error plotted above although the sense of operation of the turbines is inverted as discussed above, necessitating the gain used to be opposite sign. The integral terms are plotted in **Figure 3e**. Clearly evident is the build up in integral action for turbine A following the step which eventually corrects the boost pressure. In the EGR on case the integral term is initially saturated for both turbines due to the excessive combination of boost and EGR demand at the initial condition. Some degree of wind-up is evident in both cases as the integral term takes up to one second to come out of saturation. This condition should not occur in a production calibration since demands would not be set so as to drive the hardware to the limits of possible operation. Finally the derivative term is presented for each case (**Figure 3f**). The noisy response typical of a derivative action is evident, leading in part to a small gain being used. The resulting action has little effect on the transient control of the turbines.

Turbine responses – The **Figure 4a** shows turbine actuator vacuum resulting from the sum of open loop, proportional, integral and derivative actions controlling the vacuum regulator. This vacuum is translated in the next pair of plots (**Figure 4b**) into turbine position. Here the saturation of the turbines in the EGR on condition is clear. Each turbine is hard up on its respective closed limit. These limits are set in the software to correspond to a position very close to the actual end of mechanism travel. Turbine A responds quite slowly to the initial drop in vacuum due to an over-center effect in the mechanism. Once the turbine is less than 80% closed the response speeds up considerably. Turbine B is already fully closed at the initial condition so the small drop in vacuum immediately following the step has no effect. On the contrary, as the exhaust pressure rises rapidly (**Figure 4c**) the resulting gas force on the sliding vane assembly tends to open the turbine up. Exhaust pressure is the variable most directly affected by turbine position. The large excursion in pressure for both turbines in the EGR on case is clear. This is due to the turbines being in their most closed positions and only decreases once they have opened significantly.

The speed of turbine A was measured and is the next variable plotted (**Figure 4d**). The lower initial turbine speed when EGR is used is due to the lower mass flow through the device and means that a larger acceleration is required to reach the final speed of around 120,000 rev/min. An overshoot in speed is observed which matches well with the overshoot in boost seen in **Figure 2c**. One of the most critical variables in determining exhaust emissions is air fuel ratio (AFR). The fifth pair of plots (**Figure 4e**) shows AFR as calculated from the ECU data for instantaneous fuel and air mass flows. The accuracy of these data is not as great as for typical steady state test cell instrumentation but the relative changes are reflected well for such fast changing operating conditions. Here the contrast between tests conducted with EGR on and those without is stark. The initial AFR in the EGR off case is around 85 for turbine A and 112 for turbine B. The difference is due to slightly different fuel flow rates at the low load condition. Following the pedal step the AFR drops rapidly as fuelling is increased. There is only minor undershoot and the AFR quickly reaches a steady level of around 25. When EGR is used the initial AFR is much lower, around 40, reflecting the high levels of EGR used. Following the pedal step the fuelling again increases rapidly and although the mass air flow increases rapidly the residual proportion of EGR in the inlet system is sufficient to cause a marked undershoot in AFR. In this extreme case the AFR drops to around stoichiometry. The consequence of this can be seen clearly in **Figure 4f** - exhaust opacity. In the no EGR tests smoke was barely noticeable following the step. The rich spike in the EGR case causes large exhaust opacity spikes with both turbines. Such behavior would be unacceptable in a production calibration. There are a number of ways to avoid this situation:

- Increase mass air flow more rapidly - this is difficult since the response of the turbocharger system is already rapid and a very large amount of excess air would be required.
- Reduce EGR by increasing the MAF demand map - this could compromise the steady state NOx performance of the engine although the calibration used here certainly uses excessive EGR and could usefully be reduced significantly. Later hardware configurations of the engine can comfortably tolerate increased EGR levels in the steady state and to a lesser degree in the transient case. The effect of steady state calibration changes on the transient performance needs to be carefully assessed at each stage.
- Slow down the fuelling transient - the strategy does this to a degree but if the fuelling were slowed sufficiently to avoid the undershoot in AFR the vehicle drivability would be seriously impaired in the case illustrated.
- Reduce delay times in the inlet system - reducing actuator delays, transport delays and component volumes are all beneficial here. The use of internal EGR through variable valve phasing is a very fast way of varying EGR fraction although does not allow external EGR cooling. Most of these steps are costly in terms of engine hardware complexity.

EGR system response – **Figure 5** shows EGR related variables. The first pair of plots (**Figure 5a**) show the demanded mass air flow. This is a mapped demand based on engine speed and fuelling. For the EGR on case the open loop demand (**Figure 5b**) is based on expected steady state conditions. In the EGR off case the control variables have been omitted since the valve position is forced to be fully closed at all times. The third pair of plots (**Figure 5c**) show the MAF error, of course for the no EGR condition the initial error is large for both turbines and is still appreciable at the end of the test. When EGR is used there is still a small initial MAF error for both turbines as although the EGR valve is fully open (**Figure 5f**) there is insufficient EGR flow to reduce the MAF through the air filter. Immediately following the pedal step the error and hence the proportional control action for turbine A (**Figure 5d**) become large. The EGR valve closes quickly but subsequently opens as the proportional term diminishes and the integral action fails to increase sufficiently. There is then some interaction with the boost pressure control algorithm as the overshoot in boost for turbine A causes the EGR system to close up for a second time around one second after the pedal step. This is evident in the open loop demand (**Figure 5b**) and also in the plot of valve position (**Figure 5f**) and the falling integral action at a time when the MAF error is positive, normally the integral action would be increasing to allow the EGR valve to open. These effects cause the large MAF overshoot during this period, the time taken for the integral term to recover causes the MAF control to oscillate somewhat for several seconds. The test using turbine B does not exhibit this effect, the integral action keeps the EGR valve closed early in the transient and since the boost error is never sufficiently large to cause override of the EGR system the valve is allowed to open again to properly control MAF later in the transient with only minor overshoot.

CONCLUSION

The use of PID control with open loop for the control of boost pressure and mass air flow is effective and allows tight control, particularly at steady state. Transient performance is generally good but some situations result in excessive delay in achieving the set points. Such behavior is dependant on a number of factors.

As with any control system, it is important to avoid areas where the turbocharger and the EGR valve are forced into saturation. If this is unavoidable the integral action must be carefully treated to avoid wind-up. Actuator and other mechanical delays should be minimized as they cause problems for traditional feedback controllers. It is noticeable that turbine B performs well in comparison with turbine A in almost all respects despite a relatively incomplete controller calibration. This is in large part due to its faster response when increased boost is required.

The treatment of large transients may be improved by an additional term designed to be active only when the error is large following a rapid demand change. This may allow the effect of the derivative action to be imitated without the associated noise and stability concerns. This could, for example, ensure the maximum speed of response of the EGR valve and minimize the effect of inlet system delays on emissions and drivability. A number of hardware modifications could be introduced to achieve similar benefit through the reduction of delays and lags. Model based control strategies would also be valuable in this respect, allowing the system dynamics to be described effectively. Although treated largely separately in the control strategy used here, the EGR and turbocharger circuits are heavily dependant on each other. The control strategy succeeds for the must part through careful calibration, although a more advanced multi-variable strategy (5) would improve performance when extreme conditions are encountered and reduce calibration time through accurate representation of the system dynamics.

This work underlines the importance of proper evaluation of the dynamic performance of the system, including the control strategy, at an early stage in development. The work would ideally be carried out in parallel with the steady state investigation. It is unlikely that sufficient repeatability would be achievable in a vehicle due to uncertainties in engine torque measurement. A dynamic engine test facility, as used here, overcomes this difficulty and provides essential calibration data for the system simulation.

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NOTATION

ECU: Electronic control unit
EGR: Exhaust gas recirculation
MAF: Mass air flow
Nm: Newton Metres
PID: Proportional, integral, derivative
PWM: Pulse width modulation
T_{90}: Time to achieve 90% of final value
VGT: Variable geometry turbocharger

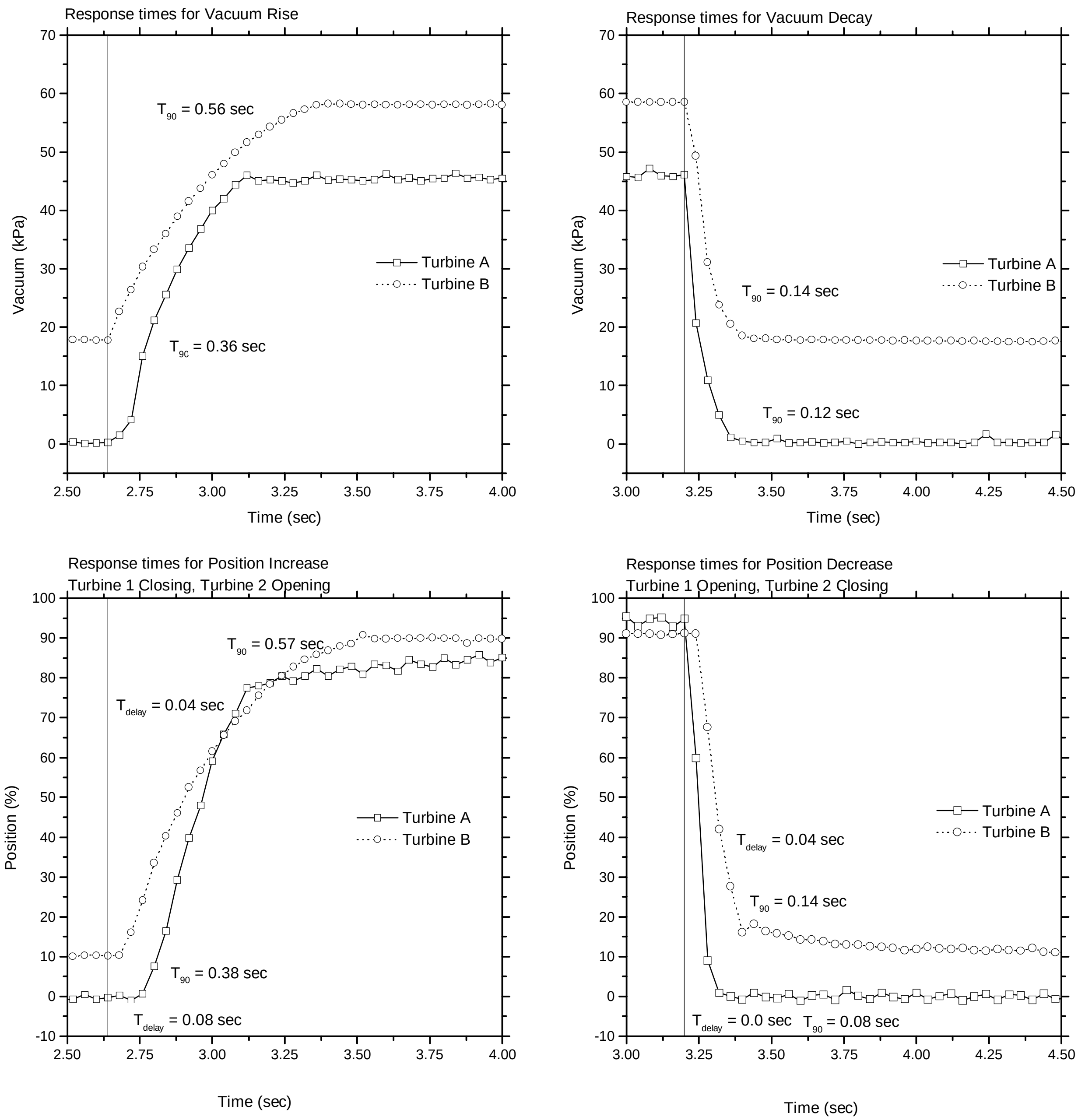

Figure 1. Vacuum and turbine position step response

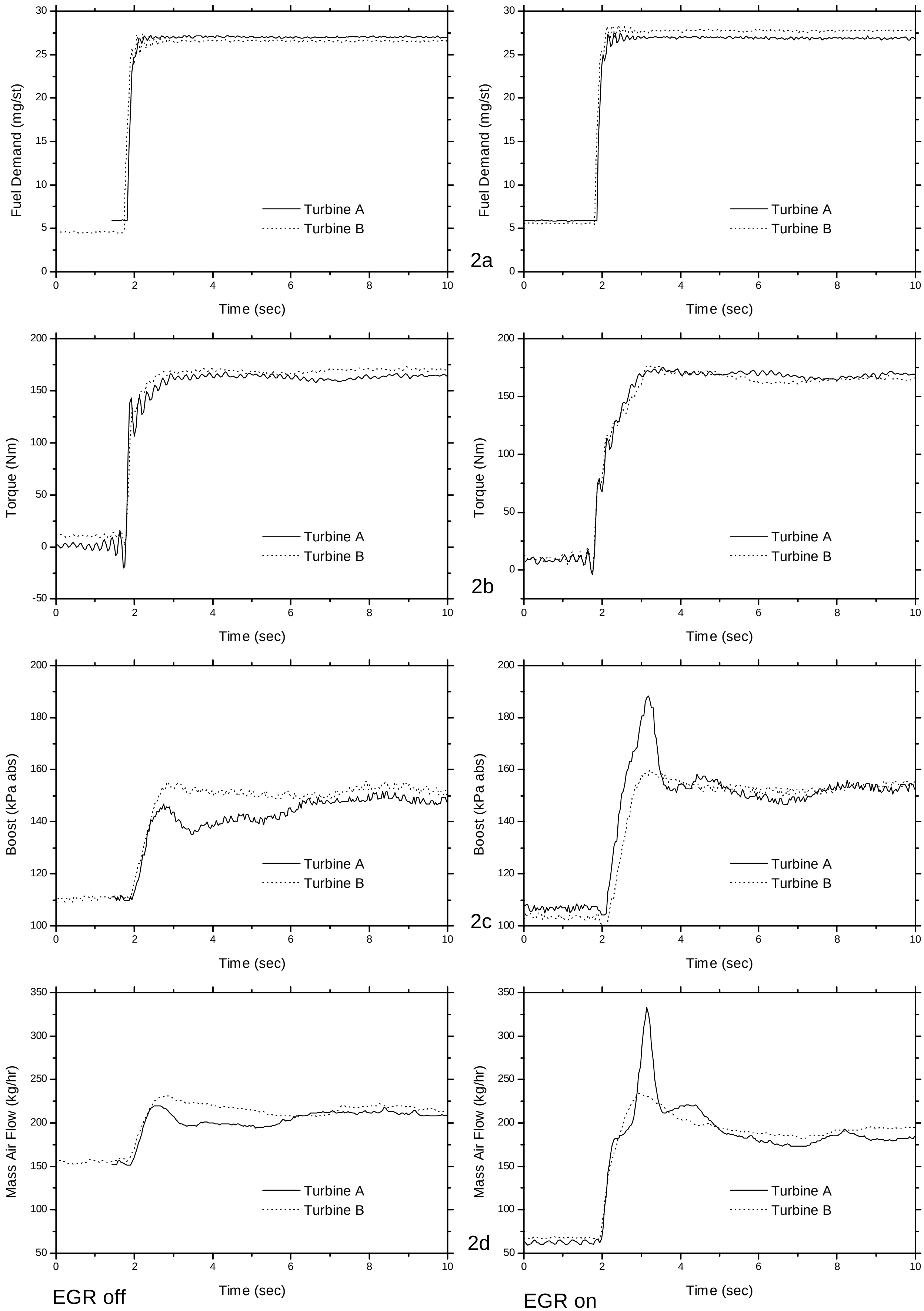

Figure 2. Engine response to step change in pedal demand

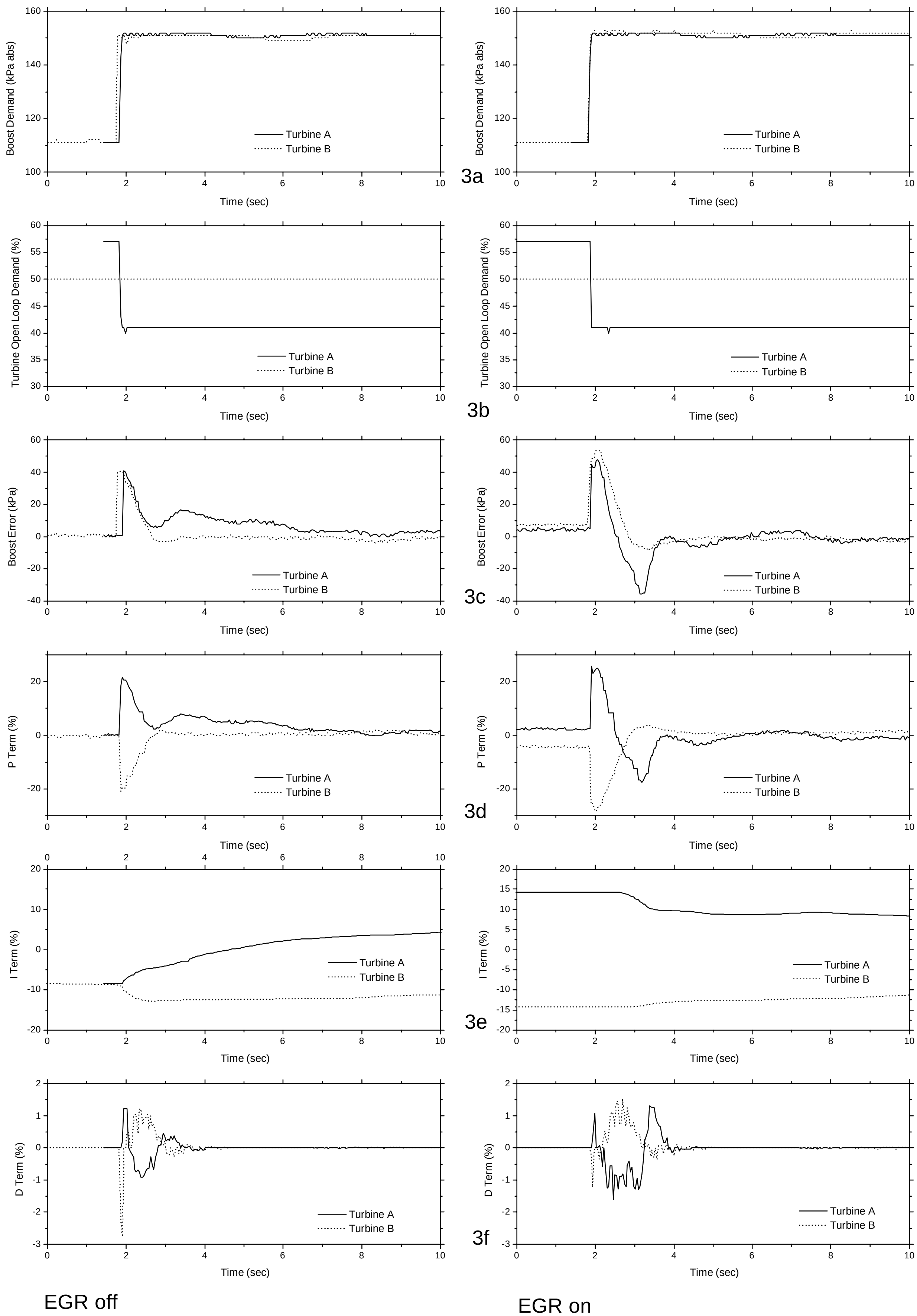

Figure 3. Turbocharger controller response to step change in pedal demand

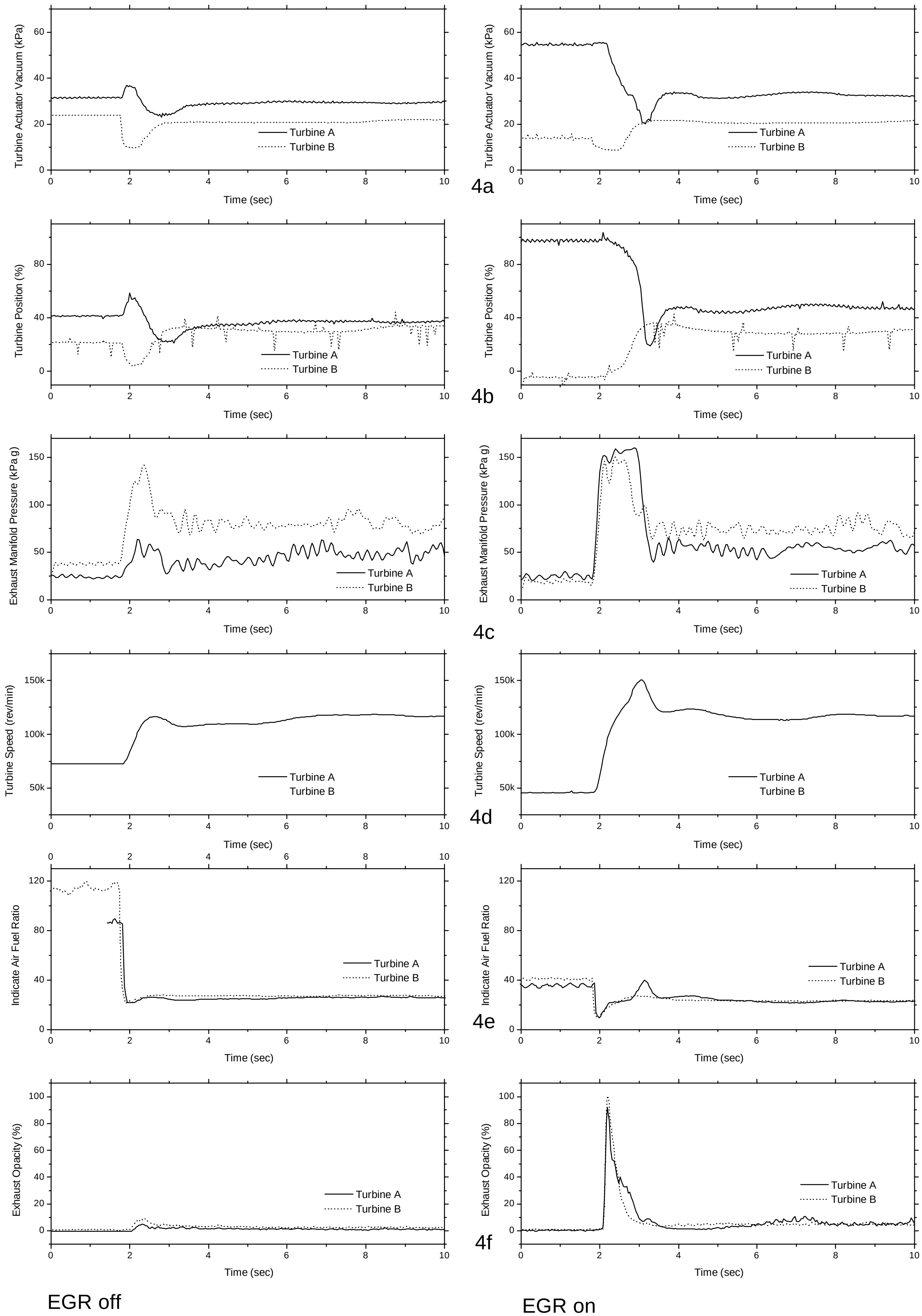

Figure 4. Turbocharger and emissions response to step change in pedal demand

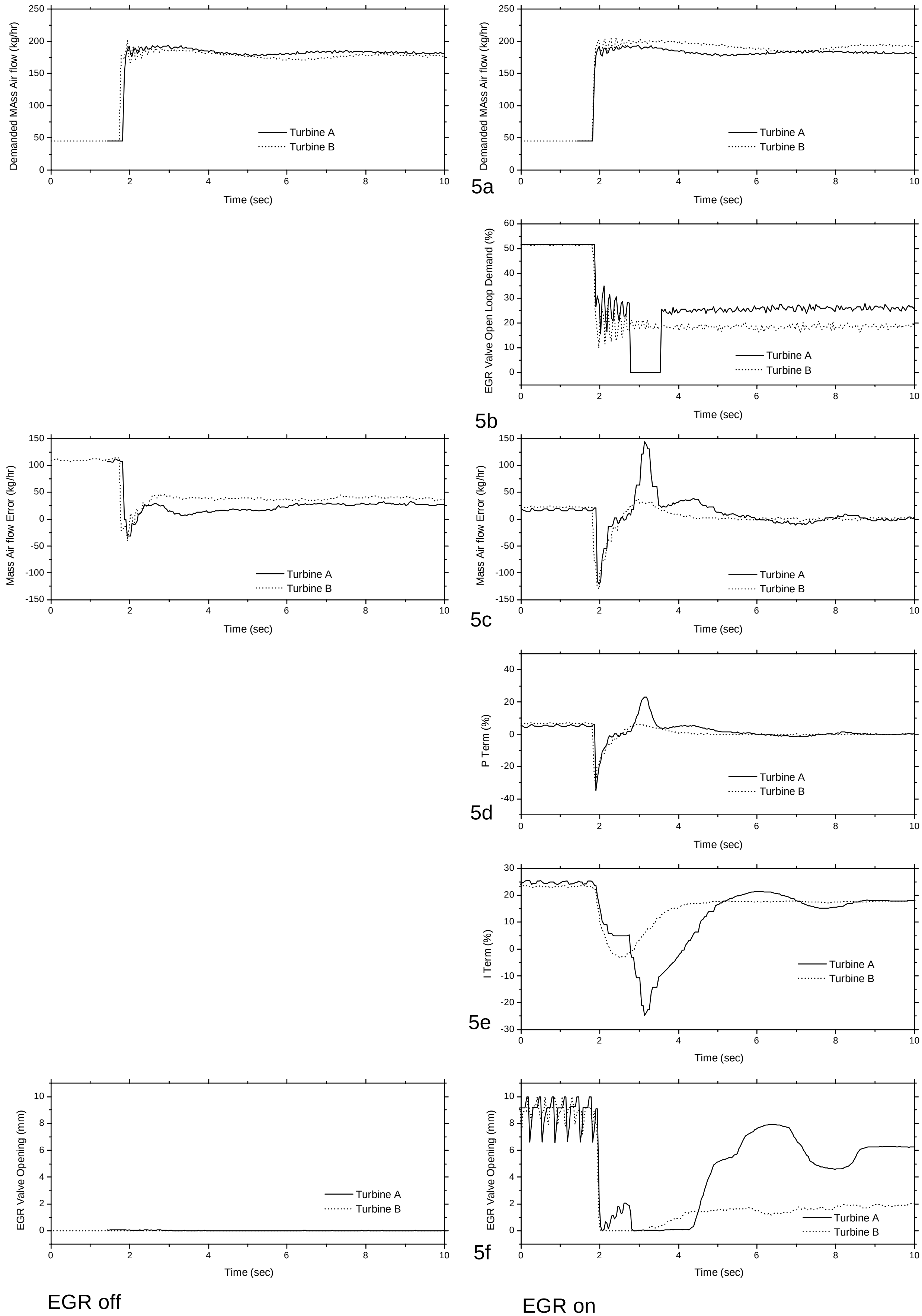

Figure 5. EGR controller response to step change in pedal demand