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(54) **Method for controlling an engine**

(57) A method for controlling torque aberrations of an engine, wherein said engine comprises at least one actuator for controlling engine management, generally comprises the steps of

- a) recording rotating speed data of said engine,
- b) processing said rotating speed data for detecting a variation of rotating speed of said engine and calculating an actual rotating speed variation,
- c) determining an actuator correction value from

said actual rotating speed variation, and

d) feeding said actuator correction value into an actuator controller of said actuator.

EP 1 227 235 A1

Description

Introduction

[0001] The present invention relates to a method for controlling an engine, and more specifically a method for controlling non-requested torque changes in an engine.

[0002] Direct injection gasoline engines (DIG) have become commercially available in recent years and are projected to form an increasingly important segment in the automotive powertrain market in Europe and in Japan.

[0003] DIG engines generally operate with a number of combustion modes other than the homogeneous stoichiometric/rich mode used by traditional multiple port fuel injection (MPFI) engines. These additional modes will include stratified lean and homogeneous lean modes. Switching between the different modes will generally occur in response to different driver torque requests (more desired torque might mean switching from homogeneous lean to homogeneous rich mode). These torque requests might be defined in terms of acceleration in some systems.

[0004] In addition to this load-speed considerations, the combustion mode is based on a variety of other engine condition criteria such as the need for NOx regeneration or catalyst protection. Consequently, the combustion mode with which the DIG engine is operating may change frequently throughout a drive cycle, even if no change in torque is requested by the driver. The driver will not wish to feel a change in torque during these mode-switching periods. Furthermore, in view of component durability, fuel economy and emission requirements, each abrupt torque change is of course potentially undesirable. Accordingly, there is a need for controlling the engine to ensure as smooth a transition between combustion modes as possible, so that each abrupt torque change is prevented.

[0005] In order for high quality driveability to be maintained, it is important that torque aberrations resulting from mode switching are minimised. In order for these to be controlled, some form of measurement of torque would be required. An obvious means for doing this would be the use of a torque sensor. Such a sensor would be able to provide absolute values of torque. However, these torque sensors are very expensive and very heavy.

[0006] Consequently the control of the engine is generally achieved through the use of look-up tables, e.g. spark timing maps or fuelling maps, with desired torque as an input, in order to determine actuator settings for the new combustion mode. The transition strategies generally do not actively control the torque during and/or immediately after a transition. Consequently, there exists a risk that the transition may not always be smooth. Should there be a deterioration in the relative quality of the look-up tables utilised for the mode trans-

mission, e.g. as a result of plant ageing, a torque mismatch between combustion modes will occur and a smooth transition will no longer take place.

Object of the invention

[0007] The object of the present invention is to provide an improved method for controlling an engine.

General description of the invention

[0008] In order to overcome the above-mentioned problem, the present invention generally proposes a method for controlling torque aberrations of an engine, wherein said engine comprises at least one actuator for controlling engine management, which comprises the steps of

- a) recording rotating speed data of said engine,
- b) processing said rotating speed data for detecting a variation of acceleration of said engine and calculating an actual acceleration variation,
- c) determining an actuator correction value from said actual acceleration variation,
- d) feeding said actuator correction value into an actuator controller of said actuator.

[0009] The method of the present invention is based on the fact, that torque and rotating speed of an engine are related by the following simple relationship $T = J \times \omega$, where T represents net torque, J represents rotational inertia and $\dot{\omega}$ represents angular acceleration. A torque aberration can therefore be defined as follows:

$$\Delta T = J \times \Delta \dot{\omega}$$

i.e. the torque aberration (ΔT) is linearly related to an acceleration aberration ($\Delta \dot{\omega}$). It follows that by determining a correction value based on actual acceleration variation, a torque aberration can be effectively corrected. Such an algorithm based upon engine speed is particularly apt because of the fact that it is the change in engine/vehicle speed that is actually noticed by the occupants of the vehicle and is directly responsible for any perception of transition roughness. This equation also demonstrates that such a method is applicable to control systems where driver commands are expressed in terms of acceleration as well as in terms of torque.

[0010] Thus, the method of the present invention represents a closed loop correction algorithm, which actively considers the actual situation in order to correct torque aberrations. Furthermore, this correction is based on readily available data in current engines. In fact, in existing engine control software, a time value is recorded

every time the flywheel is at a specific angular position. In some software, mainly relating to 4 cylinder engines, this might e.g. be every time the flywheel rotates through 180 degrees. From these time values, a velocity (averaged over the 180-degree sampling period) is calculated. The algorithm can therefore take these time data used to calculate velocity in addition to the results of the pre-existing velocity calculation. The algorithm being based on a detection of flywheel speed acceleration variation, no sensing of absolute torque values and accordingly no torque sensors are required. This algorithm is therefore useful on the current generation of systems that are currently being developed without torque estimators or sensors. The algorithm uses only current sensors and therefore not requires the implementation of any additional hardware.

[0011] Furthermore, the output of the algorithm is fed into an existing actuator for controlling engine management. The actuator can e.g. comprise an engine control unit for controlling spark timing and/or fuelling and/or air intake and/or exhaust gas recirculation and/or port deactivation. It follows that the method of the present invention does not rely on hardware other than that already present in current engines.

[0012] The method according to the present invention is particularly suitable for controlling torque aberrations due to combustion mode transitions in a direct gasoline injection engine. In this case, a transition status flag indicative of the occurrence of a combustion mode transition is preferably set and the actuator correction value is determined and fed into an actuator controller of the engine at least if said transition status flag is set. A sudden drop or increase in torque due to the introduction of a new combustion mode would cause an acceleration or deceleration of flywheel speed. This might then be used to introduce a change in fuelling (DIG being fuel lead control in most cases) or spark in order to rapidly compensate the torque.

[0013] A potential problem with respect to the implementation of such an algorithm would be the consideration of torque changes during a combustion mode transition that are not associated with the mode switch itself. Such torque changes might be caused by an increase/decrease in desired torque for purposes of vehicle acceleration/deceleration, changes in road gradient/surface and vehicle drag, or gearshifts. These torque changes would also affect engine speed, and therefore their effects upon flywheel speed would have to be taken into account by the algorithm.

[0014] In a more preferred embodiment of the method, a change in demanded torque will therefore be considered. In this case, a change in demanded torque ΔT_{dem} is determined and a demanded acceleration variation $\Delta \dot{\omega}_{dem}$ is calculated based on the change in demanded torque. The difference between said actual acceleration variation $\Delta \dot{\omega}_{act}$ and said demanded acceleration variation $\Delta \dot{\omega}_{dem}$ is then calculated for obtaining an undesired acceleration variation $\Delta \dot{\omega}_{und} =$

$\Delta \dot{\omega}_{act} - \Delta \dot{\omega}_{dem}$. Finally, the actuator correction value will be determined based on said undesired acceleration variation. In some control systems the demanded parameter may already be defined in terms of acceleration. It should be clear that in such a case a torque to acceleration conversion is not necessary.

[0015] In order to also compensate torque changes due to engine load, a variation of acceleration $\Delta \dot{\omega}_{load}$ of said engine prior to the setting of said transition status flag can further be determined, said variation being indicative of a change in engine load, and a load based acceleration variation calculated. The difference between the actual acceleration variation $\Delta \dot{\omega}_{act}$ and said demanded acceleration variation $\Delta \dot{\omega}_{dem}$ and said load based acceleration variation $\Delta \dot{\omega}_{load}$ is then calculated for obtaining an undesired acceleration variation $\Delta \dot{\omega}_{und} = \Delta \dot{\omega}_{act} - \Delta \dot{\omega}_{dem} - \Delta \dot{\omega}_{load}$. Finally, the actuator correction value will be determined based on said undesired acceleration variation.

[0016] In an alternative embodiment, the acceleration variation can be converted into an actual torque change value ΔT_{act} using the relationship $\Delta T = J \times \Delta \dot{\omega}$. The actuator correction value is then determined based on said actual torque change value ΔT_{act} . If a demanded torque change ΔT_{dem} is to be considered, a difference between said actual torque change value ΔT_{act} and the demanded torque change value ΔT_{dem} can be calculated for obtaining an undesired torque change value $\Delta T_{und} = \Delta T_{act} - \Delta T_{dem}$, and the actuator correction value can be determined based on said undesired torque change value ΔT_{und} . If a load based torque change ΔT_{load} should be taken into account, a load based torque change value ΔT_{load} can be calculated based on a variation of acceleration $\Delta \dot{\omega}_{load}$ of said engine prior to the setting of said transition status flag and the undesired torque change value ΔT_{und} may be calculated by $\Delta T_{und} = \Delta T_{act} - \Delta T_{dem} - \Delta T_{load}$.

[0017] One potential flaw in the overall strategy of utilising change in flywheel speed in order to facilitate smooth transition between combustion modes is that the algorithm will only operate once a speed change, implying a torque change, has already been detected. It could therefore be desirable for the algorithm to correct for torque before the changes become perceptible to the driver. In a preferred embodiment of the method, said actuator correction value is therefore stored and said stored correction value is used for controlling torque aberration if said combustion mode transition is reversed. The magnitude of torque discrepancy due to transition from one mode to another may be indicative of the torque discrepancy that may occur when the mode switch is reversed. This is providing that the switching to and from modes occur relatively close to one another in time, e.g. during Nox regeneration transitioning with respect to a lean Nox catalyst. Furthermore, look-up tables or the like can be used to provide different actuator correction values for different engine conditions. These tables should preferably be adaptive

in order to provide some learn capability over time. This would also allow for deterioration in the relative quality of the look-up tables utilised for the mode transmission, e.g. as a result of plant ageing.

[0018] In order to disable the torque correction when the brakes of the vehicle are applied, a brake applied flag may be set when vehicle brakes are commanded and the actuator correction value is determined and fed into an actuator controller of the engine only if said brake applied flag is not set.

[0019] It has to be noted, that the determination of the actuator correction value is preferably executed by means of a PID controller or a PI controller or a derivative thereof.

[0020] It should further be noted that, in addition to DIG, the algorithm of the present invention could also be applicable within the context of cylinder deactivation. This is because cylinder deactivation could also lead to abrupt torque changes as well.

Detailed description with respect to the figures

[0021] The present invention will be more apparent from the following description of a not limiting embodiment with reference to the attached drawings, wherein

Fig.1: a fundamental representation of the control algorithm;

Fig.2: the base algorithm architecture; and

Fig.3: the position of the algorithm within the overall control software.

[0022] The utilisation of the flywheel speed for correction of engine torque is essentially a feedback technique and may be represented in its most basic form as shown in fig.1.

[0023] The algorithm might be considered to be a type of state estimator. This is true in as much as that a measured state (engine speed) is used to estimate a torque change, i.e. an engine state that cannot be measured on current production vehicles. Torque and speed are related by the following simple relationship $T = J \times \dot{\omega}$ [eq.1], where T represents net torque, J represents rotational inertia and ω represents angular speed. A torque aberration related to mode switching might therefore be defined as follows: $\Delta T = J \times \Delta \dot{\omega}$ [eq.2], i.e. the torque aberration (ΔT) is linearly related to a flywheel acceleration aberration ($\Delta \dot{\omega}$). If the control algorithm is to be used in conjunction with a torque based software, this ΔT value may be fed directly back into the software as a desired torque increase or decrease; thereby leaving the actuator setting commands to the core software.

[0024] It is clear from eq.2 that two flywheel speed samples must be taken in order to obtain one value of flywheel acceleration. In order to identify an acceleration aberration comparison must be made with respect to previous values. Additionally, the algorithm also has to take into account that a change in acceleration may well

be occurring independently, but simultaneously with a mode change. This may either be as a consequence of a change in load torque or a change in actual torque commanded by the engine controller.

[0025] Fig.2 outlines the basic architecture of the control algorithm. The algorithm is basically made out of the following components:

PID Controller Bloc

[0026] With respect to particular control techniques, the problem may be approximated to a 'parameter uncertainty' problem. In such cases, integral control is often utilised as a method of correction. As a consequence the principal control technique utilised is that of a PID or a PI controller. The input to this PID controller is the error between the desired signal and the actual signal. This error signal could easily have been the torque signal, if the respective acceleration values were multiplied by an engine inertia term J_{eng} according to eq.2. However, they may be maintained as acceleration term and the constant J_{eng} can effectively be combined with the PID proportional gain constant, K_p . This can be done because the value of K_p would have to be tuned and the combination of K_p and J_{eng} would eradicate any requirement for the calibrator to know the value of J_{eng} if the algorithm is to be used in steady state operation. Nonetheless, if it is to be used during non-steady state conditions, the constant J_{eng} must be used elsewhere anyway.

Feed Forward Compensation

[0027] The magnitude of torque discrepancy due to transition from one mode to another may be indicative of the torque discrepancy that may occur when the mode switch is reversed. This is providing that the switching to and from modes occur relatively close to one another in time, e.g. during Nox regeneration transitioning with respect to a lean Nox catalyst. Such feed forward compensation is included in the algorithm. Only half of the magnitude of the first torque peak is applied for reasons of conservatism - in case the transition reversal torque is different. The feed forward compensation is of course also applied in the opposite direction to the initial torque peak.

Data Processing and Sampling

[0028] The preliminary block for the algorithm is in fact the data process and sampling block. Into this block enter the flywheel data samples. Ultimately the commanded torque from the engine control unit ECU will also enter.

[0029] The outputs from this block are:

- i) Time Interval - the sampling period is in CA degs and therefore varies in time depending upon engine

speed. This output is used in determining the rate of change in commanded torque.

ii) Velocity - used for determining combustion delays.

iii) Flywheel acceleration.

iv) Rate of delta torque change - with no transition occurring. This is for use in applying control under non-steady state conditions.

Non-Steady State Conditions

[0030] Under steady state torque conditions control is relatively easy because the controller simply has to maintain the torque at the value corresponding to the situation just before transition. However, such an approach is clearly not appropriate when that torque value has been in the process of change prior to the transition. Indeed it is also not appropriate if that torque value were to change during the period during which control is applied as a consequence of factors unrelated to the transition. Such factors that contribute to changes in torque, unrelated to transitions, consist of changes in load torque and changes in commanded torque. Their changes are taken into account by the algorithm, as may be seen in fig.2, by subtracting their respective changes during the control period from the error value fed into the PID controller.

[0031] It should be noted that any changes in commanded torque have to be passed through a delay corresponding to a combustion delay before they are applied to the error. It is also worth noting that depending on the actual algorithm structure the error may be an acceleration term. In this case the value J_{eng} has to be used to convert it into an acceleration value before application to the error signal. Additionally, commanded torque signals would have to have any torque correction component removed from them in order to prevent the controller 'correcting for a correction' in commanded torque.

[0032] Changes in load torque are far harder to account for. This is because they are not easily determined outputs of an ECU but are in fact determined by changes in external factors such as wind resistance and road gradient etc. To take such changes into account, the rate of change in load torque is determined before control is applied. This is done by finding the average change in flywheel acceleration (proportional to rate of change in torque according to eq.2), for example over a period of 150 sampling periods. Such a period is required in order to filter out instantaneous fluctuations. However it still remains relatively short in time terms (engine speed being in the order of thousands of rpm) and load torque changes, being due to physical conditions, should not change too rapidly. When this value has the value for rate of change of commanded torque subtracted from

it, rate of change of load torque prior to control is obtained. This value is then integrated throughout the period of control via a discrete integrator. It should be noted that this method does not take into account changes in load torque during control (unlike the commanded torque consideration). This is not possible to determine because torque changes due to transition and load torque cannot be separately identified.

[0033] The transition corrector algorithm can be designed to be compatible with a torque based control software as shown in fig.3. Such software essentially consists of a torque controller that is used to determine engine actuator settings such as spark timing, throttle position, external EGR rate and amount of fuel injected. The determination of such actuator settings are based upon inputs of desired indicated torque - an ordinary torque and extraordinary torque input. The extraordinary torque represents special torque requests, such as those from a traction control system or during idle, when non-optimum torque conditions are required. These non-optimum conditions may be quickly obtained through spark control. It should be noted that the described input parameters are only given as an example. Other forms of architecture are possible and compatible with the algorithm.

[0034] These torque inputs are calculated e.g. within the torque co-ordinator where torques from various sources are combined in order to provide the torque inputs to the controller. Such sources include driver pedal position, air-conditioning, traction control etc. It is appropriate to include the output of the transition corrector algorithm amongst them. In order to do so, the output of the algorithm must be defined as a torque value. The torque co-ordinator will then be responsible for deciding whether the torque correction takes the form of an extraordinary torque request or whether it should be fed into a torque controller as a corrected ordinary torque request. Negative torque corrections may well be best applied via extra-ordinary torque. Positive torque requests, when spark is already at optimum, would not be suited to spark control and would have to be achieved via correction of ordinary torque. During idle, when spark is not usually optimised, there may be an option to still apply the torque correction via extra-ordinary torque. In a software architecture that defines torque requests in terms of the potential speed of engine actuator application, then for the same reasoning, positive torque requests would usually be slow torque requests and torque reductions might usually be defined as fast torque requests.

[0035] A number of inputs to the algorithm are required. Foremost of these inputs is the data related to the flywheel. In some existing control software, a time value is recorded every time the flywheel rotates through 180 degrees. From these time values, a velocity (averaged over the 180 degree sampling period) is calculated. The algorithm would therefore take this time data used to calculate velocity in addition to the results of

the pre-existing velocity calculation.

[0036] Other data can include a transition status flag in order for the algorithm to recognise when a transition begins and ends. Additionally a similar flag representing brake applied can be fed into the algorithm in order to disable it when braking is required. This ensures that breaking is always applied as effectively as possible.

[0037] Whilst the controller sets engine actuators in direct response to a commanded desired torque input, the resultant actual torque delivered by the engine at a given instant will not always correspond to the commanded torque. This is a consequence of delays in the response of engine actuators (principally EGR valve and throttle) and other delays inherent to the operation of the engine itself (e.g. combustion delays). However the value of this actual torque is calculated within the torque controller and is also delivered as an input to the algorithm. This is in order for compensation of the transition torque correction value when changes in commanded torque occur during the transition.

Claims

1. Method for controlling torque aberrations of an engine, wherein said engine comprises at least one actuator for controlling engine management, comprising the steps of
 - a) recording rotating speed data of said engine
 - b) processing said rotating speed data for detecting a variation of acceleration of said engine and calculating an actual acceleration variation,
 - c) determining an actuator correction value from said actual acceleration variation,
 - d) feeding said actuator correction value into an actuator controller of said actuator.
2. Method according to claim 1, wherein the step of determining an actuator correction value comprises the steps of
 - a) determining a change in demanded torque,
 - b) calculating a demanded acceleration variation based on the change in demanded torque,
 - c) calculating a difference between said actual acceleration variation and said demanded acceleration variation for obtaining an undesired acceleration variation, and
 - d) determining said actuator correction value based on said undesired acceleration variation.
3. Method according to claim 1, wherein the step of determining an actuator correction value comprises the steps of
 - a) determining a demanded acceleration variation,
 - b) calculating a difference between said actual acceleration variation and said demanded acceleration variation for obtaining an undesired acceleration variation, and
 - c) determining said actuator correction value based on said undesired acceleration variation.
4. Method according to claim 1, wherein the step of determining an actuator correction value comprises the steps of
 - a) calculating an actual torque change value based on said acceleration variation, and
 - b) determining said actuator correction value based on said actual torque change value.
5. Method according to claim 1, wherein the step of determining an actuator correction value comprises the steps of
 - a) calculating an actual torque change value based on said acceleration variation,
 - b) calculating a demanded torque change value indicative of a change in demanded torque,
 - c) calculating a difference between said demanded torque change value and said actual torque change value for obtaining an undesired torque change value, and
 - d) determining said actuator correction value based on said undesired torque change value.
6. Method according to any one of claims 1 to 5, wherein said actuator correction value is determined by means of a PID controller and/or a PI controller and/or a derivative thereof.
7. Method according to any one of claims 1 to 6, wherein said actuator comprises an engine control unit for controlling spark timing and/or fuelling and/or air intake and/or exhaust gas recirculation and/or port deactivation.
8. Method for controlling torque aberrations due to combustion mode transitions in a direct gasoline injection engine, wherein said engine comprises at least one actuator for controlling engine manage-

ment, comprising the steps of

a) recording rotating speed data of said engine

b) setting a transition status flag indicative of the occurrence of a combustion mode transition; and
if said transition status flag is set

c) processing said rotating speed data for detecting a variation of acceleration of said engine and calculating an actual acceleration variation,

d) determining an actuator correction value from said actual acceleration variation, and

e) feeding said actuator correction value into an actuator controller of said actuator.

9. Method according to claim 8, wherein a brake applied flag is set when vehicle brakes are commanded and wherein said steps c) to e) are only executed if said brake applied flag is not set.

10. Method according to claim 8 or 9, wherein the step of determining an actuator correction value comprises the steps of

a) determining a change in demanded torque,

b) calculating a demanded acceleration variation based on the change in demanded torque,

c) calculating a difference between said actual acceleration variation and said demanded acceleration variation for obtaining an undesired acceleration variation, and

d) determining said actuator correction value based on said undesired acceleration variation.

11. Method according to claim 8 or 9, wherein the step of determining an actuator correction value comprises the steps of

a) determining a change in demanded torque,

b) calculating a demanded acceleration variation based on the change in demanded torque,

c) determining a variation of acceleration of said engine prior to the setting of said transition status flag, said variation being indicative of a change in engine load, and calculating a load based acceleration variation,

d) calculating a difference between said actual

acceleration variation and said demanded acceleration variation and said load based acceleration variation for obtaining an undesired acceleration variation, and

e) determining said actuator correction value based on said undesired acceleration variation.

12. Method according to claim 8 or 9, wherein the step of determining an actuator correction value comprises the steps of

a) determining a demanded acceleration variation,

b) calculating a difference between said actual acceleration variation and said demanded acceleration variation for obtaining an undesired acceleration variation, and

c) determining said actuator correction value based on said undesired acceleration variation.

13. Method according to claim 8 or 9, wherein the step of determining an actuator correction value comprises the steps of

a) determining a demanded acceleration variation,

b) determining a variation of acceleration of said engine prior to the setting of said transition status flag, said variation being indicative of a change in engine load, and calculating a load based acceleration variation,

c) calculating a difference between said actual acceleration variation and said demanded acceleration variation and said load based acceleration variation for obtaining an undesired acceleration variation, and

d) determining said actuator correction value based on said undesired acceleration variation.

14. Method according to claim 8 or 9, wherein the step of determining an actuator correction value comprises the steps of

a) calculating an actual torque change value based on said acceleration variation

b) determining said actuator correction value based on said actual torque change value.

15. Method according to claim 8 or 9, wherein the step of determining an actuator correction value comprises the steps of

- a) calculating an actual torque change value based on said acceleration variation
 - b) calculating a demanded torque change value indicative of a change in demanded torque, 5
 - c) calculating a difference between said demanded torque change value and said actual torque change value for obtaining an undesired torque change value 10
 - d) determining said actuator correction value based on said undesired torque change value
- 16.** Method according to claim 8 or 9, wherein the step of determining an actuator correction value comprises the steps of 15
- a) calculating an actual torque change value based on said acceleration variation 20
 - b) calculating a demanded torque change value indicative of a change in demanded torque,
 - c) calculating a load based torque change value based on a variation of acceleration of said engine prior to the setting of said transition status flag, said variation being indicative of a change in engine load, 25
 - d) calculating a difference between said actual torque change value and said demanded torque change value and said load based torque change value for obtaining an undesired torque change value 30
 - e) determining said actuator correction value based on said undesired torque change value. 35
- 17.** Method according for any one of claims 8 to 16, wherein said actuator correction value is stored and said stored correction value is used for controlling torque aberration if said combustion mode transition is reversed. 40
- 45
- 18.** Method according to any one of claims 7 to 17, wherein said actuator correction value is determined by means of a PID controller and/or a PI controller and/or a derivative thereof. 50
- 19.** Method according to any one of claims 7 to 18, wherein said actuator comprises an engine control unit for controlling spark timing and/or fuelling and/or air intake and/or exhaust gas recirculation and/or port deactivation. 55

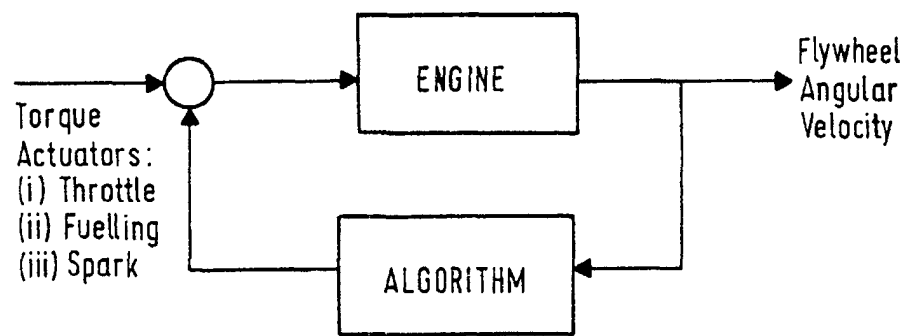
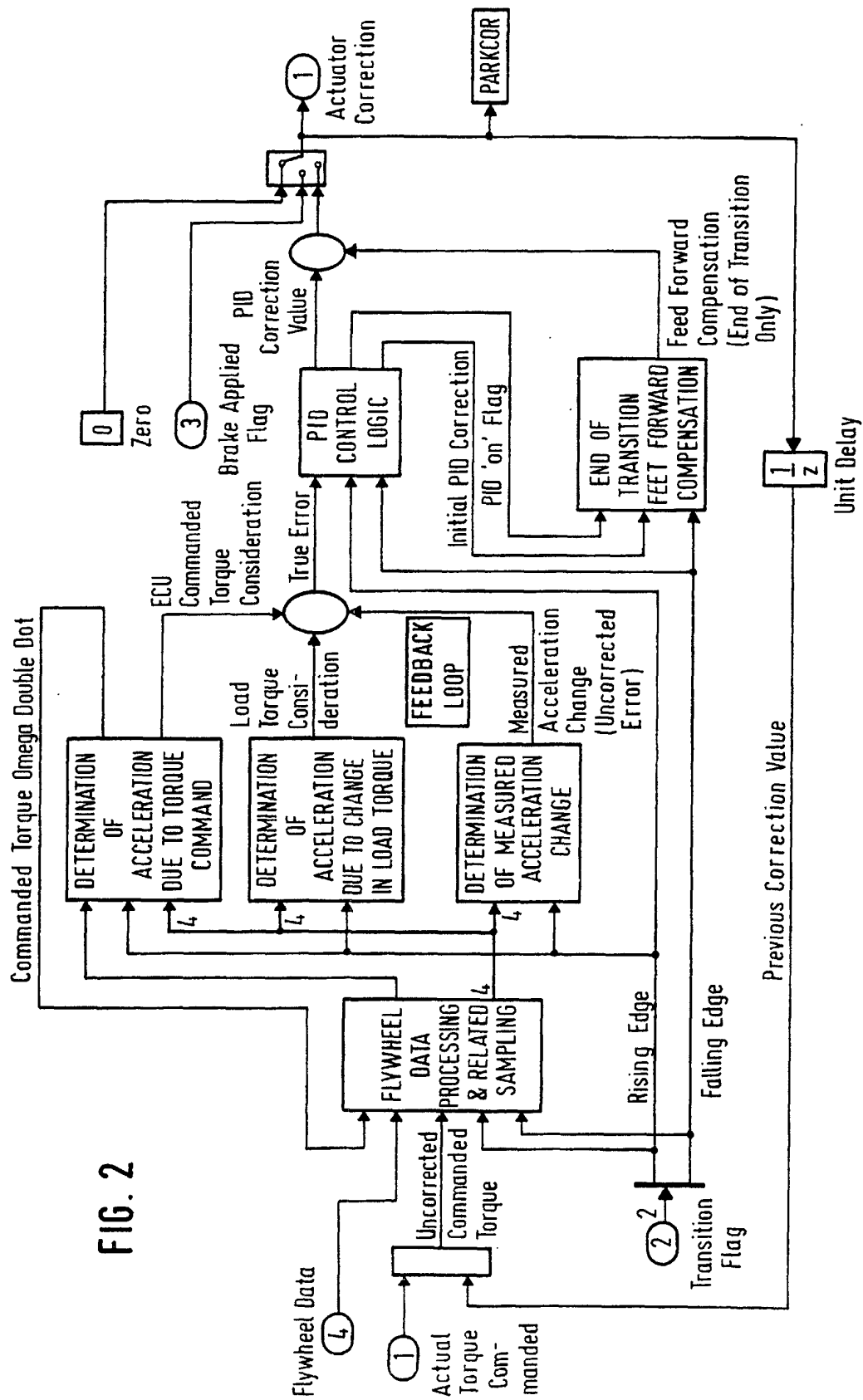


FIG. 1



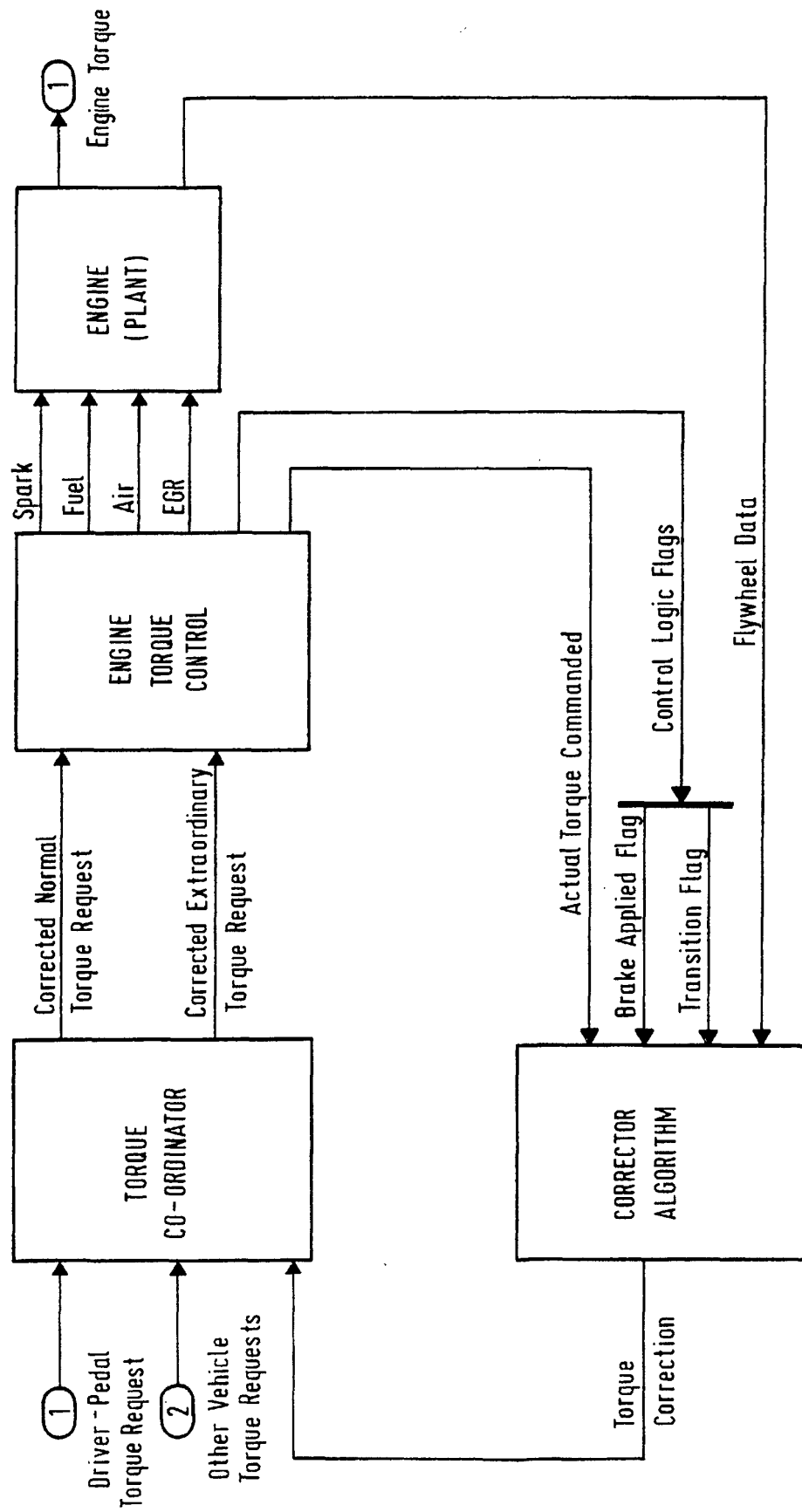


FIG. 3



European Patent
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EUROPEAN SEARCH REPORT

Application Number
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The present search report has been drawn up for all claims			
Place of search THE HAGUE		Date of completion of the search 23 April 2002	Examiner Röttger, K
CATEGORY OF CITED DOCUMENTS X : particularly relevant if taken alone Y : particularly relevant if combined with another document of the same category A : technological background O : non-written disclosure P : intermediate document		T : theory or principle underlying the invention E : earlier patent document, but published on, or after the filing date D : document cited in the application L : document cited for other reasons & : member of the same patent family, corresponding document	

EPC FORM 1503 03/92 (P04C01)

**ANNEX TO THE EUROPEAN SEARCH REPORT
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This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report. The members are as contained in the European Patent Office EDP file on
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