Design and validation of the reconfiguration strategy for a redundantly actuated intelligent autonomous vehicle

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Abstract: This article deals with the reconfiguration of an intelligent autonomous vehicle which implements automatic navigation method and online supervision system to improve the safety, traffic management and space optimization inside the confined space of sea ports. The bond graph model of the vehicle dynamic system is developed in a modular and hierarchical modelling environment. The over actuated vehicle with redundant actuators has four independent driven wheels, four independent braking wheels and four wheel steering system. This vehicle can be safely operated with appropriate control law restructuring even when some of its actuators are unusable due to some fault. For actuator fault detection, Analytical Redundancy Relations (ARRs), which are constraint relations describing nominal system behaviour and are written in terms of the measured system variables, are derived from the bond graph model. ARRs are continuously evaluated to generate residual signals and the symptoms in these signals are monitored for actuator fault detection and isolation. Once one or more actuator faults are isolated, the system is reconfigured with selection of a proper operating mode to prevent critical or accidental situations. This procedure is validated by considering a fault scenario with two reconfiguration options.

Keywords: intelligent autonomous vehicle, path tracking, fault diagnosis, fault tolerant control, reconfiguration, bond graph.

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1 INTRODUCTION

The world seaborne trade has been developing considerably in the last decade due to globalization and continuous development of emerging countries. This growth has an impact on the competitiveness, diverting some road traffic elements to maritime coastal highways, reducing the environmental impact, reducing the disparity between economically developed and less developed regions and development of ports and maritime terminals. Few ports in the coastal area (stretching from Ireland to the Netherlands) of North West Europe (NWE) are able to keep pace with this growth. The main problem for the development of the ports and terminals in the NWE area depends on the internal traffic management and space optimization inside a confined space. A solution proposed for a section of major ports such as Rotterdam, Düsseldorf and Hamburg, is to automate the handling of goods using intelligent autonomous vehicle (IAV).

The vehicle is called intelligent vehicle due to its automatic navigation method and online supervision system to improve the safety, traffic management and space optimization inside the confined space of sea ports. Autonomous vehicles are used for various purposes such as material transfer in industries and ports. An IAV can adapt its speed and trajectory according to the traffic status and environmental changes. For this reason, the IAV is equipped with a large number of embedded sensors (GPS, laser based sensors, inertial gyroscope, etc.) allowing it to be safely operated and navigated inside a congested environment. This article focuses on the development of the reconfiguration strategy for the IAV.

The IAV considered in this study is in development stage. This vehicle can be operated in two modes: autonomous and manual. It is designed to handle containers of 20 ft and 40 ft length. For transporting a 40 ft container, two identical vehicles will be coupled together. This IAV can carry either a directly deposited container handled by an external handling system such as a crane or transport the container placed on a table taking the whole table with the container.
Various types of controllers based on kinematic and dynamic models [1] have been developed for the trajectory tracking of autonomous vehicles. The autonomous vehicle needs a fault tolerant and adaptive control because a faulty vehicle may hamper the continuous flow of material inside a confined space in an industry.

A hybrid system combines continuous behavior and discrete transitions. A fault adaptive controller based on system reconfiguration, i.e., selection of a operating mode which uses the set of available healthy actuators, needs switching (discrete transition) between control laws on occurrence of every actuator fault. A hybrid fault adaptive controller implements a robust control strategy for small faults or disturbances, and a switching control strategy for large faults such as complete actuator failures. In this article, the IAV implements pure pursuit algorithm combined with proportional-integral position control for four wheel steering and two-wheel steering when a steering motor is faulty, and drive motor speed control for differential steering when more than one steering motors are faulty. The reconfiguration module switches between these steering and drive motor speed control laws. Development of hybrid fault adaptive control for mobile robots needs a model of the physical processes of the mobile robot for online fault detection and isolation during robot operation so that the mode transition algorithm can be implemented based on the availability of healthy actuators [2]. Bond graph modelling [3–4] as a unified approach for the physical modelling of various disciplines has been extensively used in vehicle dynamics studies [5] and it is used in this article for model development, ARR generation as well as for simulation.

The IAV is a complex mechatronic system. Margolis and Shim [6] developed various bond graph sub-models incorporating sensors, actuators and vehicle dynamics and used them for controller development. Pacejka [7] developed the basic framework and the multiple transformations required to model multibody systems in bond graph form. In this paper, a full
vehicle model which has 6-DOF for the vehicle chassis and also 5-DOF for each wheel and includes tyre forces [8] is considered for the vertical, longitudinal and lateral vehicle dynamics. Bond graph modelling is well suited for modular modelling of large physical systems residing in several energy domains and is also suitable for mechatronic systems [9–11]. The IAV being a complex mechatronic system is a suitable candidate for use of bond graph modelling so that the full leverage of the bond graph model can be taken for various activities such as the formal model simulation, fault indicator generation, control integration, etc.

The first phase of bond graph modeling of complex systems is a word bond graph which shows the major components of the system and their interconnections. Power and signal are exchanged between the components detailed in a word bond graph. Each component’s sub-model has a set of ports to which bonds can be connected. Unlike a block diagram model, a bond graph sub-model does not have separate input and output ports because a bond carries two counter-oriented information. The sense of information exchange is determined from causality assignment in the bonds. Therefore, a bond graph model composed of several sub-models has a neater look as compared to a modular block diagram model of the same system where more number of connections is required to establish feedbacks due to passive interaction between the components. Moreover, the modular bond graph model clearly reveals the physical structure of the system and the energy exchange between the components of the system.

In an object-oriented bond graph modeling approach, each component sub-model is treated as an object. Each object contains a sub-graph structure composed of normal bond graph elements and possibly other objects, and a set of ports for connection to other objects. This approach is sometimes referred to as port-based modeling. Sometimes, simple lower level objects are used to create higher level objects. This form of modeling with sequentially increasing complexity of sub-models is called hierarchical object oriented modeling. A sub-
model object can be developed in many forms. Modelica as an object-oriented modeling language can be used to develop equation level bond graph sub-model objects [12]. Other examples of such equation level hierarchical modelling languages are SIDOPS (textual modelling tool in 20-sim) and ACSL (with use of Macros). Dedicated bond graph modelling software like 20-sim, Symbols and MS1 allow development of graphical level bond graph objects. These object oriented modelling tools and the methodology are detailed in Chapter-11 of the book by Borutzky [4] where BGML (a XML representation of bond graph model) has been proposed as a unified syntax for exchange of bond graph models developed in different port-based modelling software.

In addition to object-oriented modelling support, a bond graph model structure can be analyzed and modified at a graphical level to implement efficient control algorithms [13–15]. Moreover, bond graph models of complex multibody systems such as the IAV can be represented in a very concise and revealing form by using the multi bond graph representation [16–17]. Bond graph modelling is also suitable for fault detection and isolation of complex systems [18]. The causal and structural properties of the bond graph [19] can be analyzed to determine not only the control theoretic properties like controllability and observability, etc. but also for determination of fault sensitivities, i.e., structural relations between faults and outputs or residuals [20]. In the context of the application considered in this article, bond graph model has been used to by various researchers to model mechatronics and vehicle dynamic systems such as humanoids [21], parallel manipulators [22], tyre road contact force estimation [23], under water vehicles [24], walking robots [25], space vehicles [26], and autonomous vehicles [27].

This paper presents a model-based approach for reconfiguration of an intelligent autonomous vehicle using bond graph models. Bond graph modelling technique has been adopted here for the modelling of the vehicle which consists of components residing in different
energy domains such as mechanical components (vehicle body, wheel etc), electrical components (traction and steering motors, sensors etc) and pneumatic components (air suspension). Bond graph model is used for generating the analytical redundancy relations (ARRs), which are evaluated to generate residuals for structural fault isolation. The main idea is to use more actuation elements, so that a fault of one actuating element can be accommodated with minor effect on the overall system dynamics. In this way, redundantly actuated vehicle can be functional with different faulty elements. This approach is certainly effective though it becomes heavy and its manufacturing cost increases. However, to increase reliability and safety of operation, especially in an automated environment, the vehicle has to be equipped by redundant actuators. The developed methodology is validated by considering a fault scenario with two probable reconfigurations. Only actuator faults are considered in this article.

The structure of this paper is as follows: First, models of various subsystems of the intelligent autonomous vehicle along with their bond graph models are explained. Then the pure pursuit algorithm for the trajectory tracking is described. After that the organisation of operating modes, fault detection and reconfiguration based on selection of proper operating modes are described. Finally, some simulation results from the reconfigured vehicle are provided.

2 MODELLING OF INTELLIGENT AUTONOMOUS VEHICLE

The intelligent autonomous vehicle (IAV) is a vehicle with four drive wheels and four braking wheels. Rear and front wheels can be independently steered by four steering motors. Each driving wheel is driven independently and brakes can also be independently applied on braking wheels. This design has been adopted for intelligent vehicles under the European commission’s InTrade project [28].
2.1 The word bond graph for the intelligent autonomous vehicle

The word bond graph representation of the intelligent autonomous vehicle (IAV) is shown in Fig. 1.
This word bond graph and the other bond graph models are developed based on the models developed in references [5, 7, 29–31] with necessary extensions. In the word bond graph, the IAV is decomposed into six subsystems. These are: vehicle body, air suspension, traction and braking wheel, motor driven steering, motor driven driving system and braking system.

The four wheel frames are connected with the vehicle body through air suspension. Each wheel frame is coupled with a pair of wheels—one traction wheel and the other one braking wheel. The coordinate transformation (CTF) block indicates coordinate transformation from one frame to the other (either inertial frame to the body-fixed frame or inverse transformation). The traction motor and brakes are connected by scalar bonds with driving and braking wheel respectively. Each wheel frame is connected through a pair of toothed gear and pinion with the steering motor separately. The body weight and aerodynamic forces acting in the inertial frame are acting on the vehicle body after coordinate transformation. In the Fig. 1, only the flow variables are shown and the effort variables are not shown due to splendor of the picture.

2.2 Vehicle body

The CAD figure of the IAV is shown in Fig. 2.

![Fig. 2 Intelligent autonomous vehicle without the container vessel.](image)

The chassis is formed of two main longitudinal steel bars welded to transversal bars. This structure is supported on air springs. The battery packs are secured to this structure. This
The technique is used for construction of industrial vehicles to avoid any kind of deformation or bending of the structure due to load. The battery packs are installed in sliding drawers under the frame and are accessible from the side. This vehicle is equipped with a 360 V DC battery pack.

The vehicle body is considered as a six-degrees-of-freedom rigid body. The Newton-Euler equations of the vehicle body [29] with attached body fixed axes aligned with the principal axes of inertia are as follows:

\[
\sum F_x = m_{cb} \ddot{x}_{cb} + m_{cb} (\dot{z}_{cb} \dot{\theta}_y - \dot{y}_{cb} \dot{\theta}_z) \quad (1)
\]
\[
\sum F_y = m_{cb} \ddot{y}_{cb} + m_{cb} (\dot{x}_{cb} \dot{\theta}_y - \dot{z}_{cb} \dot{\theta}_x) \quad (2)
\]
\[
\sum F_z = m_{cb} \ddot{z}_{cb} + m_{cb} (\dot{y}_{cb} \dot{\theta}_x - \dot{x}_{cb} \dot{\theta}_y) \quad (3)
\]
\[
\sum M_x = J_{cbx} \dot{\theta}_y - \dot{\theta}_z (J_{cby} - J_{cbz}) \quad (4)
\]
\[
\sum M_y = J_{cby} \dot{\theta}_x - \dot{\theta}_z (J_{cbx} - J_{cby}) \quad (5)
\]
\[
\sum M_z = J_{cbz} \dot{\theta}_x - \dot{\theta}_y (J_{cby} - J_{cbz}) \quad (6)
\]

The equations for three linear velocities of left–front suspension reference point in the moving system of axes are as follows:

\[
\dot{x}_i = \dot{x}_{cb} + z_i \dot{\theta}_y - y_i \dot{\theta}_z \quad (7)
\]
\[
\dot{y}_i = \dot{y}_{cb} + x_i \dot{\theta}_z - z_i \dot{\theta}_x \quad (8)
\]
\[
\dot{z}_i = \dot{z}_{cb} + y_i \dot{\theta}_x - x_i \dot{\theta}_y \quad (9)
\]

The bond graph model of the vehicle body is shown in Fig. 3. The inertias are coupled by a pair of gyrator rings which are formed using the equations (1) to (6). These gyrator rings are modeled inside the EJS (Euler junction structure) sub-model [3, 9]. Details of the EJS sub-model can be found in [29, 30]. The equations (7) to (9) are used to find the velocities at the upper and lower reference points of the suspension. The weight of the vehicle and the aerodynamic forces in the inertial frame are acting on the vehicle through coordinate transformations (CTF blocks). The coordinate transformation is done through a set of modulated transformers. It requires the values of the Euler angles which are computed Euler angle rates determined from body-fixed angular velocities. Details of the CTF and Euler angle computation can be found elsewhere [5, 7,
The reaction torques from all the four motors which are mounted on the wheel frame, act on the centre of the gravity of the vehicle body after double coordinate transform, i.e., from the wheel body-fixed frame to inertial frame and then from inertial frame to the vehicle body-fixed frame.

![Bond graph model of the vehicle body.](image)

**Fig. 3** Bond graph model of the vehicle body.

### 2.3 Wheels

The two wheels with a diameter of 0.36 m are twinned as shown in Fig. 4. One of the two wheels is integral with the traction motor and the gearbox and the other wheel is free and is equipped with hydraulic disc brakes. The steering motor is coupled with the wheel frame through a pair of toothed pinion and crown. The vertical axis of rotation of each wheel frame passes through the centre of dual wheels. With this arrangement, the vehicle can move in all directions as well as the efforts required to pull the wheel over the pavements minimizes. The pneumatic tyre and tube with radial or diagonal version is adopted here.
The bond graph models of the traction wheel and braking wheel are shown in Fig. 5(a) and Fig. 5(b), respectively.

A wheel is modeled as rigid body with five degrees of freedom in the wheel body-fixed frame considering the fact that wheel does not roll or rolling is quite negligible about its longitudinal axis. Moreover, the wheel or axle body-fixed frame does not rotate as the wheel spins about its axle. Thus, the gyrator ring for Euler’s equation completely disappears and that
for Newton’s equations reduces to a single gyrator. The longitudinal force \( F_x \), transverse force \( F_y \) and self aligning moment \( M_z \) are modeled as per Pacejka’s *magic formula* \(^8\) given as

\[
y_o = D \sin \left[ C \tan^{-1} \left\{ B x_i - E \left( B x_i - \tan^{-1} (B x_i) \right) \right\} \right]
\]

(10)

where output variable, \( y_o : F_x, F_y \) or \( M_z \) and input variable, \( x_i : \sigma_x \) or \( \sigma_y \). The 4-port modulated MR element models Pacejka’s *magic formula*.

The longitudinal and cornering dynamics are modulated by the suspension force \( (F_z) \) and instantaneous radius of the wheel \( (r_w) \) \([30–31]\). Ports 1 to 3 of the traction wheel and ports 11 to 13 are connected to the corresponding points of the lower part of suspension reference. The steering torque is applied through ports 4 and 14. The braking torque applied through port 15 for the braking wheel. The torque from the motor is applied at the rotational port \( \dot{\theta}_y \) of the traction wheel. The rolling resistance moment acting at the \( \dot{\theta}_y \) port of both the wheels is proportional to suspension force, \( F_z \) and radius of the wheel, \( r_w \). The four transformers at the top the figure (modulated by \( \gamma \) value although it is not shown explicitly) are used for coordinate transformation from inertial to wheel frame where \( \gamma \) is the yaw angle.

### 2.4 Four wheel steering system

The four wheel steering system is used for the improvement of vehicle maneuverability and stability. To minimize the turning radius during low speed maneuvering or parking, the rear wheels steer in the opposite direction of the front wheels \([32]\). But the rear wheels track the front wheels during high speed maneuvering. The major problem of four wheel steering is the power delivery to the wheels. To avoid this problem, the steering motor is mounted on the wheel frame. For the proper alignment, the steering angle must be tangent to the inner and outer circles as shown in Fig. 6 \([32]\).
The inner and outer wheels make angles $\delta_i$ and $\delta_o$, respectively, at the center of rotation and these angles are calculated as

\[
\tan \delta_i = \frac{a + b}{a + b + \tan \delta_{st} \mp 2c}
\]

(11)

and

\[
\tan \delta_o = \frac{a + b}{a + b + \tan \delta_{st} \pm 2c}
\]

(12)

where $\mp$ and the $\pm$ signs are appropriately used depending on the front or the rear wheel. Note that for two-wheel steering, either the front or the rear wheels remain straight and these formulae are accordingly revised to the usual Ackerman’s steering formulae.

The functions $\Phi_i$ and $\Phi_o$ to be used in the bond graph model determine the inner and outer wheel angles from a given reference angle ($\delta_{ref} = \delta_{st}$) by using equations (11) and (12), respectively. The bond graph model of the steering system is shown in Fig. 7. The steered wheels angular position errors modulate the voltage of the steering motors through two proportional-integral (PI) controllers. The output of these PI controllers are given as
\[ v = \xi_p \delta_e + \xi_I \int \delta_e dt \]  

where \( \delta_e \) is the angular position error, \( \xi_p \) and \( \xi_I \) are proportional and integral gains respectively, and \( v \) is the voltage supplied to the steering motor.

Fig. 7 Bond graph model of steering system.

The steering motor is coupled with the wheel frame by a pair of gears which is presented by a transformer element of modulus \( \mu_g \). The combined effective stiffness and damping of the output gear shaft and the wheel frame is denoted by \( K_p \) and \( R_p \). The final output steering motor torques (from both steering motors) are applied on the wheel frames through ports numbered 4 and 14 in the bond graph models given in Figs. 5(a) and 5(b), respectively. Note that each wheel frame holds two wheels, a traction wheel and a braking wheel as shown in Fig. 4. The reaction torque is directly applied on the vehicle body to induce a reactive yaw.

The comparison of Y-displacement of the vehicle center of gravity with 4 wheel steering and 2 wheel steering during a lane change process is shown in Fig. 8(a). It is seen from the picture that the radius of rotation for the vehicle with 4 wheel steering is lesser than the vehicle with 2 wheel steering. The angle of rotations of the wheels are lesser in case four wheel steering (this result is not shown). It is found from the results shown in Fig. 8(b) that at the vehicle
operating speed (6.883 m/s) and constant steering angle $\delta_{st} = 0.15$ rad, the vehicle moves in a steady circular path of radius of 16.25 m under four wheel steering condition which means that there is a mild under steering characteristics because the vehicle should move on a circle of radius of 15.22 m under neutral steer. The parameter values are given in Table 1.

<table>
<thead>
<tr>
<th>Sub-system</th>
<th>Parameter values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle body</td>
<td>$m_{cb} = 2917.2$ kg $J_x = 1925.6$ kg.m$^2$ $J_y = 16396.7$ kg.m$^2$ $J_z = 17561.6$ kg.m$^2$ $a = 2.3$ m $b = 2.3$ m $c = 0.86$ m $h = 0.486$ m</td>
</tr>
<tr>
<td>Suspension</td>
<td>$L_{sn} = 0.165$ m $c_i = -3.448 \times 10^9$ $c_2 = 1.762 \times 10^9$ $c_3 = -3.565 \times 10^8$ $c_4 = 3.32 \times 10^7$ $c_5 = -1.523 \times 10^6$ $c_6 = 8.238 \times 10^4$ $k_s = 10^7$ N/m $r_s = 2000$ N.s/m</td>
</tr>
<tr>
<td>Wheel frame</td>
<td>$m_w = 42$ kg $J_{wy} = 2$ kg.m$^2$ $J_{wz} = 1$ kg.m$^2$ $r_w = 0.38$ m $k_p = 10^8$ N/m $R_p = 10^3$ N.s/m</td>
</tr>
<tr>
<td>Tyre</td>
<td>$B_t = 10.19$ $C_t = 1.2$ $D_t = 3191.409$ $E_t = -2$ $B_{tr} = 4$ $C_{tr} = 1.2$ $D_{tr} = 0.0304$ $E_{tr} = -40$ $K_{t1} = 305$ kN/m $R_t = 200$ N.s/m $C_{tr} = 0.03$</td>
</tr>
<tr>
<td>Traction motor</td>
<td>$v_1 = 360$ V $v_2 = 360$ V $v_3 = 360$ V $v_4 = 360$ V $R_{tm} = 0.2$ $\Omega$ $\mu_{tm} = 19.71$ N.m/A</td>
</tr>
</tbody>
</table>

**Fig. 8** (a) Y-displacement of the vehicle during 4 wheel steering and 2 wheel steering and (b) trajectory of the centre of the mass of the vehicle for a steering angle of 0.15 rad under four wheel steering condition.
2.5 Active differential steering

For active differential steering, the traction motor supply voltages (MSe elements) are regulated. If no wheel is steered and the vehicle is supposed to take a turn of radius $r$ with vehicle c.g. tangential velocity $V$ (or yaw rate, $V/r$) then the inner wheels must move at a linear speed of $(1-c/r)V$ and the outer wheels must move with a linear speed of $(1+c/r)V$. The equivalent steering angle becomes $\delta = \tan^{-1}\left(\left(a+b\right)/r\right)$. Thus, for a given equivalent desired steering angle or yaw rate, the desired wheel angular speeds of inner and outer wheels can be computed. The wheel supply voltages are then modulated to achieve this speed and a feedback loop with a PI controller maintains the reference angular velocities. This control architecture is shown in the bond graph model given in Fig. 9. Note that the blocks $\Phi_i$ and $\Phi_o$ implement functions to compute the reference inner and outer wheel angular speeds. The output of the PI controllers is $\pm \Delta \nu$, which is the variation of the average supply voltage over the average nominal voltage used for the specified vehicle speed. The voltage modulation can be achieved through proportional input regulation or pulse-width modulation.

![Differential steering controller and bond graph model of active differential steering system.](image)

**Fig. 9** Bond graph model of active differential steering system.
In this case, the output torque is applied to the $\dot{\theta}_y$ port in the traction wheel bond graph model given in Fig. 5(a). The reaction torque is applied to the vehicle body after coordinate transformation from wheel frame to the vehicle frame to induce a reactive pitch motion. This is because the motors are mounted on wheel axis and pivoted to the vehicle body.

2.6 Air suspension

Air suspension is used here to prevent shock and vibration that is transmitted from the wheel to the frame and to keep the wheels in contact with the road surface during acceleration, braking or cornering. The pressure inside the air spring is regulated as the function of load carried by the vehicle. High pressure is applied to the spring when the vehicle is loaded with more loads which is called active ride height management. For the optimum use of this spring, it is operated in the range of 55 mm to 165 mm with a pressure of 6 bar. So, the maximum load varies between 24 kN to 54 kN as shown in Fig. 10(a).

The bond graph model of the air spring is shown in Fig. 10(b). The force acting on the air spring modelled by a Se element is a function of the change in length of the spring. The force-deformation relation is obtained from curve fitting around the operating point and is given as

$$F = c_1 L_5^5 + c_2 L_4^4 + c_3 L_3^3 + c_4 L_2^2 + c_5 L_1 + c_6,$$

where $c_1 = -3.448 \times 10^9$, $c_2 = 1.762 \times 10^9$, $c_3 = -3.565 \times 10^8$, $c_4 = 3.32 \times 10^7$, $c_5 = -1.523 \times 10^6$, $c_6 = 8.238 \times 10^4$.

3 CONTROL SCHEME BASED ON PURE PURSUIT ALGORITHM

For path tracking, many algorithms are available in the literature. The pure pursuit algorithm and Follow the Carrot are generally used for the vehicle to approximately follow a predefined path [34]. In this article, pure pursuit algorithm is used for trajectory tracking.
The trajectory control is achieved through one of the three mechanisms: four wheel steering, two wheel steering or active differential steering (wheel speed control). The control architecture is shown in Figure 11.

The overall control logic is a hybrid control. There are a set of cascaded proportional-integral (PI) controllers and discrete mode changes (transitions) that switch between the cascaded controllers through an if-then-else logic loop. For active differential control to follow a reference vehicle trajectory, the set-point for wheel-speed controllers are computed from another controller based on pure-pursuit algorithm. Likewise, pure pursuit algorithm decides the set-point for steering wheel position controller. Both steering wheel position and drive wheel angular speed controllers are PI controllers whose set-points are determined by another controller. Note that at
any point of time, only one form of the trajectory control, i.e., steering position or active
differential, is used.

Fig. 11 Structure of the steering position and wheel speed controllers.

The measurements used for trajectory control with fault tolerance are the vehicle’s
forward velocity and yaw rate, the motor currents (four drive motors and four steering motors),
supply voltage from a common source (battery pack), and wheel angular speeds. For actuator
fault detection, only measurements from actuators are necessary.

A reference trajectory is specified and the steering controller drives the vehicle along this
trajectory. The steering control is a cascaded control system. The pure pursuit algorithm is used
in the first controller to find the desired steering orientation from the actual and target positions
of the vehicle. The calculated steering angle is the set-point (reference steering angle) for the
steering angular position controller, which is a PI controller. For steering through active differential control of drive motors (changing left and right wheel speeds to take a turn), the pure-pursuit algorithm specifies the set-point to wheel speed controllers which are PI controllers.

The measurements from drive and steering motors are fed to a monitoring system (shown as reconfiguration module in Fig. 11). This monitoring system monitors the time evolution of residuals. The residuals are evaluation of certain constraint equations called analytical redundancy relations which are written in terms of measurements only. When any residual deviates from zero there is some fault. The actual fault is isolated by looking at which residuals have deviated and which have not. This part is implemented in a computer program.

If a fault is isolated in one or more actuators then the decision support system in the computer selects the next best suited operating mode. The steering control law and drive motor control laws are switched to new control laws. These control laws are implemented in if-then-else blocks. For example, the four-wheel steering can change to two-wheel steering or the steering motors can be locked and active differential control of drive motors can be applied.

3.1 Pure pursuit algorithm

The vehicle starts moving from the point o and the path is straight along the X-axis as shown in Fig. 12(a). Now the vehicle deviate from the trajectory at point A due to some reason, say a fault, and point B is the instantaneous position (given by GPS) of the vehicle after some time. The closest point of the vehicle to the defined path is point C. Point D is the look-ahead point and the CD is the look-ahead distance \( l \). The look-ahead angle \( \phi_L \neq 0 \) is the characteristic if the vehicle deviates from its path for some reason. If the deviation of the vehicle is very less compared to the look-ahead distance then \( \phi_L \) is proportional to the distance [34], i.e.,

\[
\phi_L = \tan^{-1} \left( \frac{Y}{l} \right) \quad \text{or} \quad \phi_L = \frac{Y}{l} \quad \text{for} \quad Y \ll l.
\]  

(15)
3.2 Controller for path tracking

The block diagram of the controller is shown in Fig. 12(b). When fault is introduced to the vehicle there is a difference between the reference trajectory position and the actual vehicle position. This positional error needs to be removed to track the actual trajectory. During faulty period, the loss in angular position and deviation from the track is accumulated. The proportional gain term is derived from pure-pursuit algorithm and it is given as $\zeta_1 = 1/l$ where $l$ is the look ahead distance. The integral gain $\zeta_2$ is chosen depending on how much time delay is there between fault occurrence and the reconfiguration, i.e., how much positional error is accumulated. The output of the controller is the desired steering angle, $\delta_{ref}$. This information is applied to steering position controller (Fig. 7) or traction wheel speed controller for active differential steering (Fig. 9). The decision on which motors will be used depends on the current operating mode (reconfiguration state by taking into account the availability of healthy actuators). This is discussed in the next section.

4 FAULT DIAGNOSIS

A mathematical model of the system is required for the model-based quantitative Fault Detection and Isolation (FDI). Analytical FDI checks the consistencies between the actual process and the predicted behaviour of the system described by Analytical Redundancy Relations (ARRs). These
differences are called residuals. The number of ARRs equals the number of sensors. A structurally independent residual has unique fault signature, i.e., it becomes sensitive to a certain fault and insensitive to other faults. The ARRs should be robust, i.e., insensitive when no fault is present and should be sensitive and structured during faulty situation [35]. The value of each residual should be zero or near to zero during normal operation and should deviate appreciably during faulty situation. Fault in a component is monitorable if at least one residual is affected by it. A binary vector indicating the deviation and non-deviation of residuals from their nominal values or thresholds is called a coherence vector [35]. If the coherence vector is non-null (i.e., at least one residual has deviated substantially from its nominal behaviour), a fault is identified. The component fault can be isolated if its fault signature is different from fault signatures of all other components so that the non-null coherence vector in case of a fault can be uniquely matched to a predefined fault signature derived from a model, experiment, experience, etc [35]. In this case, the set of predefined fault signatures, organized into a tabular form called Fault Signature Matrix (FSM), is derived from the bond graph model of the system.

4.1 Organization of operating modes

The reconfiguration of the vehicle is possible if redundant devices are available in good health during failure of one or more components [27]. The equipment availability chart with operating modes [25, 36] is shown in Fig. 13. In this tree-structure, the associated devices for the performing of basic functionalities are shown. Here, the number of devices is more than the requirement of a particular task to be performed. Any particular task can be operated normally if at least one device is available for that purpose. Note that Fig. 13 gives only a partial list of operating modes and missions; for example one of the operating modes for lane change mission not shown in the figure is steering of the vehicle by active differential action which is achieved by modulating wheel speeds on the inner and outer sides of the curve.
Note that when the motor of a geared DC motor fails (e.g., electrical power supply to the motor is somehow disconnected) then it requires a huge torque to rotate the motor and it acts as a brake on the vehicle. Thus, this vehicle system is equipped with the necessary clutching mechanisms in the transmission system which disconnects or disengages the wheel axle from the motor on occurrence of any such event. This disengagement can be automatic or triggered by a control signal.

![Equipment availability chart with operating modes.](image)

**Fig. 13** Equipment availability chart with operating modes.

When the failure of a particular device is detected by the FDI approach, the branches associated with that device or equipment are removed and the system can be reconfigured using the remaining available device(s). For example, if the front left motor fails, the vehicle can be reconfigured during lane changing process either by rear-wheel drive with front wheel steering or by rear wheel drive with rear wheel steering. This conclusion is drawn from Fig. 13 where cross marks indicate unavailable equipment and operating modes (those operating modes which contain the unavailable equipment). Now, if front left motor, front steering motor and rear steering motor are not available, it is apparent that the vehicle cannot be operated during lane changing process. But, the lane changing is possible by rear wheel drive with differential motor
speeds which is not shown in the picture. So, the vehicle cannot be operated during lane change mission if front left motor, both the steering motors as well as any of the rear wheel motors fail. Of course, there are other such fault combinations under which lane change mission cannot be accomplished. The same is true for other missions.

4.2 Fault detection

The sensors in the bond graph model given in Fig. 3 and Fig. 4(a) are dualized into sources and the preferred differential causality is assigned. Based on the algorithms presented in [37], the ARRs are derived. As there are three sensors (to measure \( \dot{\theta}_y \), \( \dot{\theta}_z \) and traction motor current \( i \) ) in each the traction wheel, one gets three ARRs for each wheel. The three ARRs for the rear traction wheels are given in equations (16-18). The ARRs for the other three traction wheels (ARR\(_4\) to ARR\(_{12}\)) can be found similarly. The longitudinal force \( F_x \), transverse force \( F_y \) and self aligning moment \( M_z \) are calculated from the Pacejka magic formula (Eq. (10)). The suspension force \( F_z \) can be measured and the coefficient of rolling resistance \( C_{rr} \) for the car tyre on the asphalt road is considered. Note that the tyre-road contact forces can be more realistically modeled with experimental identification of contact model parameters [23]. The six sensors for the vehicle body lead to six ARRs (ARR\(_{13}\) to ARR\(_{18}\)) which are given in equations (19-24). Let us assume that the vehicle is equipped with inertial sensors and gyros from which angular and linear velocities in the body-fixed frame can be estimated [38]. We will discuss more about these sensors and the corresponding ARRs at a later stage.

\[
\begin{align*}
\text{ARR}_1 : R_m i + \mu_m \dot{\theta}_y - \nu &= 0 \quad (16) \\
\text{ARR}_2 : J_{wy} \ddot{\theta}_y + F_x r_w C_{rr} - \mu_m i + F_x r_w &= 0 \quad (17) \\
\text{ARR}_3 : J_{wz} \ddot{\theta}_z + M_z - \tau_z &= 0 \quad (18) \\
\text{ARR}_{13} : \sum_{j=1}^{4} F_{x,j} + m_c \ddot{x} - m_c \dot{\theta}_z \dot{y} + m_c \dot{\theta}_y \dot{z} + F_1 &= 0 \quad (19)
\end{align*}
\]
In an ideal situation, evaluation of the LHS of ARRs is zero. However, when real measurements are used to evaluate the LHS of ARRs, the result is called a residual. The residual is ideally zero for no fault and non-zero when there is a fault. To account for modeling uncertainties like errors in parameter estimation, sensor noise and bias, etc., it is assumed that the residual has deviated from its nominal value only when it exceeds a static or adaptive threshold \[35\]. Such an approach to account for uncertainties is called a passive approach. The ARRs are structurally analyzed to generate the fault signature matrix (FSM) \[39\] as shown in Table 2. This table shows variations in parameters of which components cause changes in which residuals. It is seen that fault in any component affects at least one residual, so all component faults are monitorable. It is also observed that fault signature (the way a fault influences some residuals and does not influence others \[35\]) for each component is unique. Thus, faults in all of the listed components can be isolated.

Note that the ARRs are derived using bond graph modelling software Symbols \[40\]. The FDIPad toolbox \[41\] of Symbols not only derives the ARRs in symbolic form, but also generates the fault signature matrix and allows changes in instrumentation architecture (sensor and actuator placement) to optimize the fault isolation performance.
The fault simulation is performed using the system equations and ARRs derived from the bond graph model developed in Symbols software. Note that for compatibility with other research activities in the InTrade project, the ARRs derived in Symbols software are implemented in Matlab software to evaluate the residuals for online supervision of the IAV.

**Table 2. Theoretical Fault Signature Matrix**

<table>
<thead>
<tr>
<th>Component</th>
<th>Residuals</th>
<th>Mo</th>
<th>Io</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>r1  r2  r3  r4  r5  r6  r7  r8  r9  r10  r11  r12  r13  r14  r15  r16  r17  r18</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rear left traction motor</td>
<td>1  1  0  0  0  0  0  0  0  0  0  0  0  0  0  0  1  1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rear left wheel</td>
<td>0  1  0  0  0  0  0  0  0  0  0  0  0  0  0  0  0  1  1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rear left steering motor</td>
<td>0  0  1  0  0  0  0  0  0  0  0  0  0  0  0  0  0  0  1  1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rear right traction motor</td>
<td>0  0  0  1  1  0  0  0  0  0  0  0  0  0  0  0  0  0  1  1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rear right wheel</td>
<td>0  0  0  0  1  0  0  0  0  0  0  0  0  0  0  0  0  0  1  1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rear right steering motor</td>
<td>0  0  0  0  0  1  0  0  0  0  0  0  0  0  0  0  0  0  1  1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Front left traction motor</td>
<td>0  0  0  0  0  1  1  0  0  0  0  0  0  0  0  0  0  0  1  1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Front left wheel</td>
<td>0  0  0  0  0  0  1  0  0  0  0  0  0  0  0  0  0  0  1  1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Front left steering motor</td>
<td>0  0  0  0  0  0  0  1  0  0  0  0  0  0  0  0  0  0  1  1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Front right traction motor</td>
<td>0  0  0  0  0  0  0  1  1  0  0  0  0  0  0  0  0  0  1  1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Front right wheel</td>
<td>0  0  0  0  0  0  0  0  1  0  0  0  0  0  0  0  0  0  1  1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Front right steering motor</td>
<td>0  0  0  0  0  0  0  0  0  1  0  0  0  0  0  0  0  0  1  1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Body</td>
<td>0  0  0  0  0  0  0  0  0  0  0  1  1  1  1  1  1  1  1  1</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
solver (modified 4th order Runge-Kutta method) and the outputs sampled at finite intervals are fed to the residual evaluator programmed in Matlab. Note that the residual evaluator is not a simulation and Symbols software also supports residual evaluation. However, Symbols software does not support interface with data acquisition system and therefore Matlab is preferred here for implementation of residual evaluation, fault isolation (comparing the fault signature with the FSM), reconfiguration system (decision on mode change) and the overall decision support system.

Let us assume a specific fault scenario to validate the theoretical FSM. The initial configuration of the vehicle is considered as rear wheel drive with front wheel steering. The vehicle is initially moving with a constant speed of 6.811 m/s along a straight line. After 10 s movement of the vehicle, power supply to the motor of the rear left traction wheel is disconnected intentionally for the fault simulation.

![Figure 14](image)

**Fig. 14** Corresponding deviations in the two residuals for a specific traction wheel upon occurrence of a fault (fault in both front traction wheels at $t = 0$ s, fault in rear left traction wheel at $t = 10$ s and fault in rear right traction wheel at $t = 20$ s). The ARR numbers are suitably offset.

It is seen from Fig. 14 that the first two residuals which are sensitive to the rear left traction wheel, deviate just after 10 s. The other residuals which are not shown in the figure do not deviate during this faulty situation. The coherence vector (deviation and no deviation of a
residual are indicated by binary numbers 1 and 0, respectively, in a vector form) is $\begin{bmatrix} 1 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}$. This coherence vector has a unique match with the fault signature in the first row of Table 2 and thus the rear left wheel traction motor fault is isolated. Introduction of faults in other wheels at different times shows similar sensitivity to faults.

4.3 Practical Considerations

The theoretical FDI analysis done so far is difficult to implement. The first problem is to estimate the body parameters and aerodynamic loads. The container will be loaded and unloaded in arbitrary manner and the placement or removal of objects from the container frame would change the location of its center of mass, distance from center of mass to suspension reference points, mass and weight, rotary inertias, etc. Likewise, the aerodynamic load and ground profile are unpredictable. It is impossible to fix those parameters every time. Thus, use of ARRs related to body dynamics for FDI is simply impractical. Likewise, determining tyre and road parameters at all times is a difficult task because tyres will wear over time, the unpredictable weather (rain, snow, etc.) would change the tyre-road grip (Pacejka magic formulae parameters) unpredictably, the tyre pressure can change as the vehicle operates (tyre temperature is an important factor) and suspension health influences normal load and tyre grip significantly. Thus, $\text{ARR}_{15}$ and three other such ARRs for other wheels are also best left out of the analysis.

These technological specifications eliminate many components and residuals from the theoretical FSM. Moreover, the sensitivity of residuals to faults is tested to formulate the practical fault signature matrix $[39]$. This leaves us with only four ARRs for electrical motor part of four wheels and four ARRs for the steering motors. Four additional ARRs for breaking wheels may as well be included provided brake pad friction is properly estimated. Note that failure of any of the front steering motors hampers front wheel steering. The same is true for rear wheel
steering. Thus, we can linearly combine two ARRs for the two front wheel steering motors to one ARR and the two ARRs for the two rear wheel steering motors to another ARR. This gives us a total of six ARRs for the four drive wheels and front and rear steering motors. We will leave braking wheels out of the analysis at this stage and thus a revised FSM given in Table 3 will be used for monitoring driving motor faults. In Table 3, residuals have been renumbered and they are structured, i.e., diagonal. Structured residuals allow us to isolate multiple faults. Structured residuals also ensure that fault effects cannot cancel out symptoms associated with each other. For example, if the coherence vector is $[1 \ 0 \ 0 \ 0 \ 1 \ 0]$ then from Table 3, it can be concluded that the rear left traction motor and front steering motor have failed. Thus, it is possible to independently detect and isolate the actuator faults in the considered intelligent vehicle under the assumption that the sensors in the electrical motor are not faulty. This assumption is unnecessary if additional redundant sensors are used [20].

<table>
<thead>
<tr>
<th>Component</th>
<th>Residuals</th>
<th>Mo</th>
<th>Io</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rear left traction motor</td>
<td>r₁  r₂</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Rear right traction motor</td>
<td>r₃  r₄</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>Front left traction motor</td>
<td>r₅  r₆</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>Front right traction motor</td>
<td>r₁  r₂</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>Front Steering motors</td>
<td>r₃  r₄</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>Rear Steering motors</td>
<td>r₅  r₆</td>
<td>0</td>
<td>1</td>
</tr>
</tbody>
</table>

If fault in left rear traction wheel motor occurs then only residual $r₁$ deviates (exactly the same way as shown in Fig. 14(a)) and leads to a coherence vector $[1 \ 0 \ 0 \ 0 \ 1 \ 0]$ which is sued to isolate the fault. For this isolated fault, two ways of reconfiguration based on equipment availability chart with selection of proper operating modes are considered next.
5 SIMULATION RESULTS

The model parameters in the simulations for the intelligent autonomous vehicle model are given in Table 1. The steering and drive motor controller gains are $\zeta_1 = 0.05$ (20m look ahead distance) and $\zeta_2 = 0.004$. The vehicle moves with a constant speed of 6.811 m/s along a straight line by using both rear traction wheels and four wheel steering. After 10 s movement of the vehicle, the motor of the rear left traction wheel is disconnected for the fault simulation. As a result of this the vehicle deviates from its straight line path and its linear speed falls. The vehicle can be reconfigured by two ways. Centre of mass position in X and Y frame in the top view and vehicle centre of mass velocity in x and y frame versus time when front steering is used after reconfiguration are shown in Fig. 15. The longitudinal and lateral velocity of the vehicle changes slightly due to the fault in the motor of the rear left traction wheel. Both the rear wheels are disengaged from the drive at 20 s and the new configuration adopted is front wheel drive with front wheel steering. The residuals ARR$_1$ and ARR$_2$ for different wheels are shown in Fig. 14. It is seen from Fig. 15(a) that the vehicle starts to follow the desired path after the reconfiguration.

![Fig. 15](image)

**Fig. 15** (a) Centre of mass position in X and Y frame in the top view and (b) vehicle centre of mass velocity in x and y frame versus time when front steering is used after reconfiguration.
The same faulty vehicle can be reconfigured to front wheel drive with differential front wheel speeds (active differential configuration). Note that this approach is generally not preferred because the wheels must slip significantly in the lateral direction for the vehicle to turn and thus it leads to rapid tyre wear. Therefore, usually a preference or weight is given to reconfiguration options and the available operating mode with highest preference is selected \[28\]. In this least preferred operating mode, the vehicle steers due to application of differential front speed of the vehicle without using the front steering motor. Centre of mass position in X and Y frame in the top view as shown in Fig. 16(a) is almost similar to the earlier case. But vehicle centre of mass velocity in x and y frame versus time as shown in Fig. 16(b) is slightly different from the earlier case.

![Fig. 16](image)

**Fig. 16** (a) Centre of mass position in X and Y frame in the top view and (b) vehicle centre of mass velocity in x and y frame versus time when differential speed is applied to front wheels after reconfiguration.

Note that when both the rear wheel and front wheel steering motors fail, and one of the rear wheel traction motors fails then the only remaining option to accomplish lane change mission is to use front active front wheel differential drive which becomes the default. Thus, the preference of a specific operating mode is dynamic and it depends on the availability of equipments.
6 CONCLUSIONS

This paper contributes to modelling of intelligent autonomous vehicle system using bond graph tool which is a multi-energy domain approach to integrated system modelling. The over-actuated intelligent autonomous vehicle offers many reconfiguration options for fault in one or more actuators. We have created bond graph models of different subsystems of the intelligent autonomous vehicle and integrated with controllers to achieve specified control objectives. The controller for path tracking has been developed based on the pure pursuit algorithm. The advantage of four wheel steering over the two wheel steering was shown with simulation results. The fault detection and isolation is implemented through structural analysis of the residuals. The directional handling performance of the vehicle with different actuator combinations in reconfiguration options after a specific actuator fault has happened has helped us to sort out these methods according to their performance and to give them an order of merit. This sorting was performed for failure hypothesis in individual actuators and combination of actuator faults. This information is then stored in a decision support system as equipment availability chart with order of preference of operating modes. An appropriate reconfiguration option is chosen on the fly when a fault or set of faults are detected.

ACKNOWLEDGEMENTS

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REFERENCES


**APPENDIX**

### Notation

- **a**: Distance of front axle from the vehicle cg (m)
- **b**: Distance of rear axle from vehicle cg (m)
- **B**: Stiffness factor
- **C**: Half of track width (m)
- **C₀**: Shape factor
- **Cᵣ**: Coefficient of rolling resistance
- **D**: Peak value
- **E**: Curvature factor
- **F**: Force (N)
- **h**: Height of vehicle cg from ground (m)
- **i**: Current
- **J**: Polar Moment of inertia (kg.m²)
- **k, K**: Stiffness (N/m)
- **L**: Length (m)
- **m**: Mass (kg)
- **M**: Moment (N.m)
- **r**: Effective radius (m)
- **R**: Damping (N.s/m)
- **x, y, z**: Displacements in three directions (m)
- **X, Y, Z**: Inertial displacements in three directions (m)
- **x, y, z**: Velocities in three directions (m/s)
- **x, y, z**: Accelerations in three directions (m/s²)
- **v**: Voltage
- **δ**: Yaw angle (rad)
- **δᵢ**: Transformer modulus
- **δᵢ**: Look ahead angle
- **ζ**: Proportional gain

### Subscripts

- **aero**: Aerodynamic
- **BI**: Body fixed to inertial
- **IB**: Inertial to body fixed
- **cb**: Vehicle body
- **f**: Front
- **g**: Gear
- **i**: Inner
- **o**: Outer
- **r**: Rear
- **ref**: Reference
- **s**: Suspension
- **sm**: Steering motor
- **st**: Steering
- **t**: Tyre
- **tm**: Traction motor
- **tr**: Trailing
- **w**: Wheel
- **wb, wy, wz**: x, y, z direction of wheel
- **zE**: Euler angle in z direction
- **1b, 2b, 3b, 4b**: Body fixed 1, 2, 3, 4 suspension reference point
- **1i, 2i, 3i, 4i**: Inertial 1, 2, 3, 4 suspension reference point
- **1b, 2b, 3b, 4b**: x-direction body fixed 1, 2, 3, 4 suspension reference point
- **1b, 2b, 3b, 4b**: y-direction body fixed 1, 2, 3, 4 suspension reference point
- **z1b, z2b, z3b, z4b**: z-direction body fixed 1, 2, 3, 4 suspension reference point
- **z1b, z2b, z3b, z4b**: x-direction inertial 1, 2, 3, 4 suspension reference point
- **z1b, z2b, z3b, z4b**: y-direction inertial 1, 2, 3, 4 suspension reference point
- **z1b, z2b, z3b, z4b**: z-direction inertial 1, 2, 3, 4 suspension reference point