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Assessment of Fault Tolerance for Actively Controlled Railway Wheelset

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Abstract— **this paper studies the key issue of fault tolerance for actively controlled railway wheelsets. It assesses failure modes in such systems, with a focus on actuator failures, and consequence of those hardware failures. It seeks to establish the necessary basis for control reconfiguration to ensure system stability and performance in the event of a faulty, without the need for hardware redundancies. A number of control schemes (with and without faults) are included in the study. Both analytical and simulation results are presented.**

Keywords-Railway; Wheelset; Active control; Stability; Actuator; Fault tolerance.

I. INTRODUCTION

Conventional wheelset for the railway vehicle is composed of the two coned (or profiled) wheels rigidly fixed to a common axle to rotate at the same angular velocity. When an unconstraint wheelset rolling along the track it is displaced laterally due to track irregularities, the rolling radii therefore are different because of the profiles of these two wheelset. Consequently, different forward speeds obtained for each wheelset due to the difference rolling radii to provide a natural centering/curving action. However, an unconstrained wheelset also presents a problem of kinematic instability known as the “Kinematic Oscillation” or wheelset “hunting” [1,2].

Traditionally the wheelset is stabilized by using passive suspensions on conventional rail vehicle, but such additional stiffness affects the pure rolling action of the wheelset around the curve. It has been theoretically proven that to this design conflict between stability and curving performance can be solved by applying active control instead of conventional passive components within the primary suspension of railway vehicle [2].

Passive components in the primary suspensions can be designed in such a way not to fail in order to maintain the stability and steering performance of railway vehicle and they are generally accepted as “safe” in railway industry. However, any new technology must prove that it can cope with any failures to demonstrate that any component faults would not lead to the system failure such that passenger safety is not compromised under such conditions. From a practical point of view, any active control scheme must be also able to maintain an effective operation of a rail system in order to meet the necessary standard of reliability [3]. Hardware redundancy technique may be used in the system to guarantee safety operation of such a system. Whilst it may be acceptable to apply the above technique in sensors due to their relatively low cost, it is far more difficult to justify the use of multiple actuators in a cost effective manner for redundancy or accommodate those within the limited space of railway bogie [3]. There are two main approaches for fault tolerant control systems. The first philosophy relies on the existing system redundancies to achieve acceptable performance in the event of component failures. In this type of systems, once the controllers designed, it will remain stable. It should be noted that the redundancies in such a system are usually in hardware forms. The second methodology takes a completely different approach to achieve fault tolerance. It involves such procedures as real-time fault detection, isolation, and control system reconfiguration. The redundancy in such a system may be an analytical form [4] and help to minimize the use of the hardware redundancies in order to keep the overall cost down [5].

The object of this study is to develop the fault tolerance approaches without using redundant actuators to provide stability across a range of operation conditions with different failure modes. It investigates the possibilities/feasibilities of re-configuring the controller based on the use of remaining actuator(s) in the system. For this study, the paper will review first a number of different control methods for railway wheelsets in the normal condition to understand how control for stability and/or curving performance is achieved. A thorough assessment of failure modes and adverse effect of the faults on the system stability and performance is then carried out, followed by an investigation into control re-tuning/re-configuration for fault tolerance. The paper is organized as following. The mathematical dynamic model of railway vehicle is presented in Section п. Consideration of basic control scheme is given in section ш. Section V demonstrates the different fail modes of railway vehicle with the actuator faults and re-configuration of the controller based on the remaining actuator Finally, conclusion and future work will be discussed.

II. Mathematical model of the rail way vehicle

A railway vehicle mainly consists of a vehicle body and two bogie frames, and each bogie frame consists of a bogie frame and the two wheelsets. The wheelsets are connected to the bogie frame with springs and dampers in the longitudinal and lateral directions. For this study, only the plan-view dynamics of a half vehicle is used to analysis stability and steering performance of the vehicle, which is the accepted practice in railway industry [6]. Fig.1 gives a plan-view diagram of the half body vehicle model used for this study. The equations of motion for a railway vehicle when running along a track are mainly determined by the creep forces between wheel and rail contact patches. In this paper, a linear model has been considered, which is justified as the active control tends to reduce the effect of non-linearity in the wheelsets [7]. The linear model of the motion contains seven degrees of freedom, i.e. the lateral and yaw motions for each wheelset and for the bogie frame, and a lateral displacement for the vehicle body defined by (1) to (7). The model is therefore 14th order in total, and can be represented in the state space model by (8) [6].

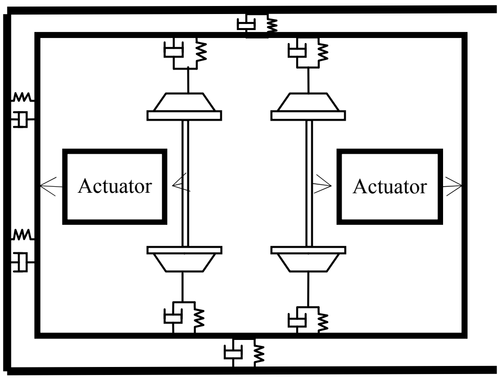

Figure 1: Plane-view of the vehicle

The input vector u represents the control inputs to the wheelsets, and the vector μ is used to represent the inputs from the railway track, including the lateral displacement, cant, and curvature. The lateral track displacement is a random input, which represents track irregularities along the path track, whereas the track curvature and cant are the deterministic inputs [6]. More details of the vehicle parameters in the equations are provided in the Appendix A.

$$\ddot{y}_{w_1} = -\frac{1}{m_w}[(\frac{2f_{11}}{V_s}+C_s)\dot{y}_{w_1} - K_s y_{w_1} + 2f_{22}\Psi_{w_1} + C_s\dot{y}_g + K_s y_g + C_s L_v\dot{\Psi}_g + K_s L_v\Psi_g + m_w(\frac{V_s^2}{R_1} - g\cdot\theta_{c_1})], \tag{1}$$

$$\ddot{\Psi}_{w_1} = -\frac{1}{I_w}[\frac{-2f_{11}L_g^2}{I_w V_s}\dot{\Psi}_{w_1} - \frac{-2f_{11}\lambda L_g}{I_w r_0}y_{w_1} + \frac{2f_{11}L_g^2}{I_w R_1} - \frac{-2f_{11}\lambda L_g}{I_w r_0}y_{t_1} + \frac{\tau_{w_1}}{I_w}], \tag{2}$$

$$\ddot{y}_{w_2} = -\frac{1}{m_w}[(\frac{2f_{11}}{V_s}+C_s)\dot{y}_{w_2} - K_s y_{w_2} + 2f_{22}\Psi_{w_2} + C_s\dot{y}_g + K_s y_g - C_s L_v\dot{\Psi}_g - K_s L_v\Psi_g + m_w(\frac{V_s^2}{R_2} - g\cdot\theta_{c_2})], \tag{3}$$

$$\ddot{\Psi}_{w_2} = -\frac{1}{I_w}[\frac{-2f_{11}L_g^2}{I_w V_s}\dot{\Psi}_{w_2} - \frac{-2f_{11}\lambda L_g}{I_w r_0}y_{w_2} + \frac{2f_{11}L_g^2}{I_w R_2} - \frac{-2f_{11}\lambda L_g}{I_w r_0}y_{t_2} + \frac{\tau_{w_2}}{I_w}], \tag{4}$$

$$\ddot{y}_g = \frac{1}{m_g}[-(2\cdot C_s + C_{sc})\cdot\dot{y}_g - (2\cdot K_s + K_{sc})y_g + C_s\dot{y}_{w_1} + K_s y_{w_1} + C_s\dot{y}_{w_2} + K_s y_{w_2} + C_{sc}\dot{y}_v + K_{sc}y_v + m_g V_s^2(\frac{1}{R_1}+\frac{1}{R_2}) - m_g g(\frac{\theta_{c_1}}{2}+\frac{\theta_{c_2}}{2})], \tag{5}$$

$$\ddot{\Psi}_g = \frac{1}{I_g}[-2L_v^2 C_s\dot{\Psi}_g - 2L_v^2 K_s\Psi_g + L_v C_s\dot{y}_{w_1} - L_v C_s\dot{y}_{w_2} + L_v K_s y_{w_1} - L_v K_s y_{w_2} - (\tau_{w_1}+\tau_{w_2})] \tag{6}$$

$$\ddot{y}_v = \frac{1}{m_v}[-C_{sc}\dot{y}_v - K_{sc}y_v + C_{sc}\dot{y}_g + K_{sc}y_g + m_v V_s^2(\frac{1}{2R_1}+\frac{1}{2R_2}) - m_v g(\frac{\theta_{c_1}}{2}+\frac{\theta_{c_2}}{2})], \tag{7}$$

$$\dot{x} = A\cdot x + B\cdot u + \mu\cdot w, \tag{8}$$

$$x = [\dot{y}_{w_1}\ \ y_{w_1}\ \ \dot{\Psi}_{w_1}\ \ \Psi_{w_1}\ \ \dot{y}_{w_2}\ \ y_{w_2}\ \ \dot{\Psi}_{w_2}\ \ \Psi_{w_2}\ \ \dot{y}_g\ \ y_g\ \ \dot{\Psi}_g\ \ \Psi_g\ \ \dot{y}_v\ \ y_v]$$

$$u = [\tau_{w_1}\quad \tau_{w_2}],$$

$$\mu = \begin{bmatrix} y_{t_1} & \theta_{c_1} & \frac{1}{R_1} & y_{t_2} & \theta_{c_2} & \frac{1}{R_2} \end{bmatrix},$$

III. BASIC WHEELSET CONTROL SCHEMES

The railway wheelset can be stabilized by using either passive suspension or through the use of active control. For active approaches, it is possible to achieve this by applying either a yaw torque or lateral force between the bogie and the wheelset, but the yaw control is preferred as it also tends to improve the ride quality experienced by passengers [8]. Therefore, this study only discusses approaches that apply control in the yaw direction to provide desired damping to stabilize the system. The review of the control strategies is to provide a background for the study of fault tolerant control issues and more detail of the controls can be found in the references provided [9,6,10]. The suspension/control schemes are considered in the study are:

- Passive Suspension that uses conventional passive yaw stiffness in the primary suspensions.
- Active Yaw Damping where the two wheelset of the bogie are controlled by applying a yaw torque proportional to the lateral velocity of the wheelset.
- Sky-hook Yaw Stiffness where the control output of each actuator is set to be proportional to the absolute yaw motion of each wheelset
- Optimal Control where the controllers for the two inputs (actuators) are designed with the use of full state feedback (from either direct measurements or through the use of an estimator).

The track input used in the simulation, to study the control performance on curves for both active controllers and passive suspension, represents a curved track with radius of 1250m connected to straight track via a transition of 2sec. The curved track is canted inward by 6 degrees to reduce the lateral acceleration experienced by the passengers (a normal features of railway track). The vehicle speed of 50m/s is used – parameters of the vehicle are provided is Appendix A. The simulation result in Fig.2 and Fig.3 clearly illustrates that active control can provide good curving performances to reduce the longitudinal and lateral creep forces, compared with passive suspension, when both leading and trailing actuator functioning normally.

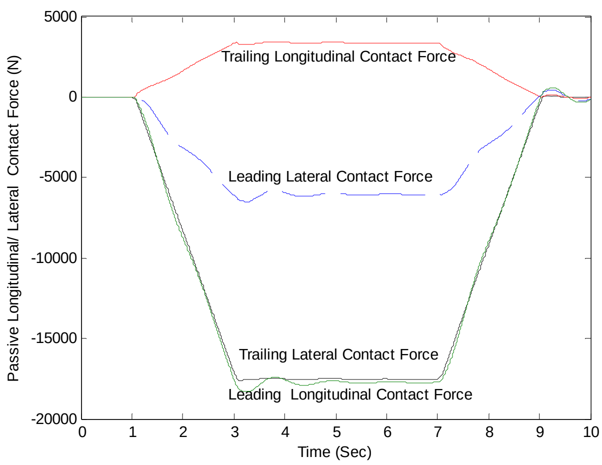

Figure 2: Passive Longitudinal/ Lateral creep forces

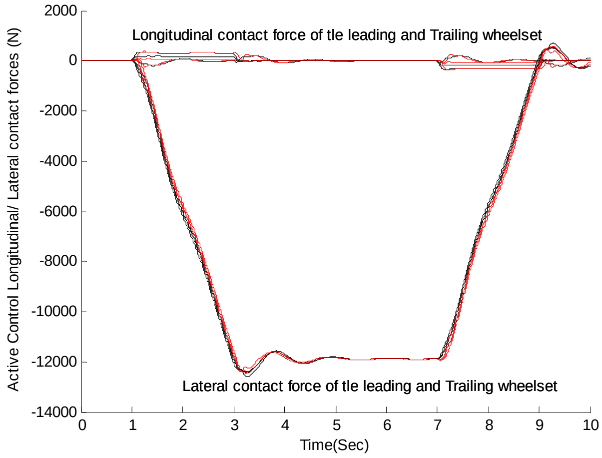

Figure 3: Active control Longitudinal/ Lateral contact forces

However, the active controllers are designed based on the assumption that both actuators function as expected and extra measures (possibly through the use of fault tolerance) will be needed in order to maintain the stability and if possible curving performance if one of the control channels fails to deliver. Therefore, it is essential to establish first a full understanding how the fault(s) affect the bogie dynamics in the system and then define what corrective actions can be taken [3].

IV. CONTROL ANLYSIS IN FAULT CONDITIONS

In the normal condition with both actuators functioning, the bogie is designed to be stable. Fig.4 shows the minimum damping ratio of the wheelset modes with the different controllers where the stability is achieved across a wide range of speed with a critical speed of over 100m/s in the three of controllers except active yaw damper [3].

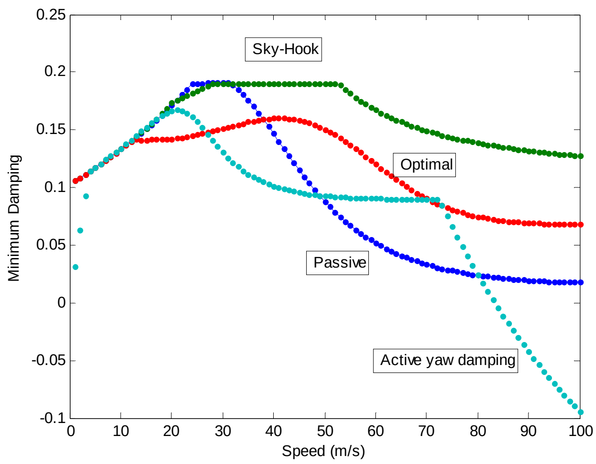

Figure 4: Comparison of different control scheme

However, this is expected to change dramatically, when one of the actuators fails. In this study, two failure modes are considered – one is fail-hard and the other fail-soft, representing an actuator jam and free-motion respectively.

The aim of fault tolerance for actively control system is to preserve stability conditions and maintain the current curving performance close to desired ones (or at least not worse than the passive system in the normal condition) in the presence of actuator faults. In this study, the full state feedback is considered as a start point and a control gain matrix is designed to control the remaining actuator in order to explore the fault tolerant control possibilities.

A. *Fail-Hard*

In the fail hard mode, one of the actuators is assumed to be blocked and can be simulated as a spring with very high stiffness between the bogie frame and the axle. The bogie stability at different speed for the selected active control schemes are compared in Figs.5a and 5b, where the fail hard occurs in the leading and trailing wheelset respectively. In Figs.5a and 5b, the critical speed is reduced to 62m/s for the active sky-hook control scheme with the malfunction of the leading actuator and around 75m/s if the trailing actuator fails-hard respectively. A fault in the trailing actuator would result in an even lower critical speed for all three active control schemes. However the fail-hard condition poses a more problem for the curving performance of the bogie.

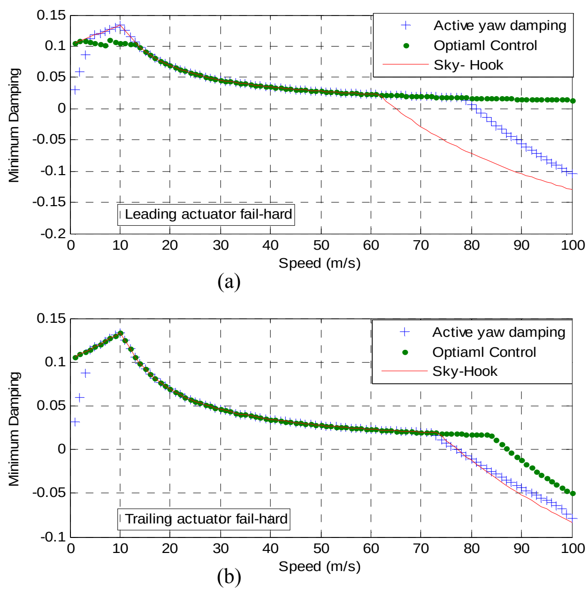

Figure 5: Stability of different control schemes with leading/trailing fault

The simulation result in Fig.6 and Fig.7 indicate clearly that when fail hard occurs in the leading and trailing control input respectively, the original controller will not be able to provide the 'right' control effort as the contact creep forces at the both leading and trailing wheelset increase significantly on the curved track, delivering a poor steering condition. The objective of the fault tolerance in the fail-hard case is therefore is to try and minimize the adverse impact of the actuator failure on the curving performance. It can be seen that the curving performance is more under risk when fail-hard occurs for the leading actuator. The simulation results in Fig. 6 and 7 suggest that the contact forces with the original controller are even worse than that with the passive suspension (Fig.2). However, retuning of the control gains for the remaining actuator does seem to provide a solution to improve curving performance of the bogie in the event of fail-hard in the leading or trailing actuator as evidenced by results in Fig.8 and Fig.9 respectively. The simulation results in Fig.8 and 9 in comparison with Fig.6 and 7 of the original controller indicate that the re-tuned optimal controller for the remaining actuator can reduce the contact forces and maintain curving performance close to that of the passive suspension. In this approach optimal controllers are tuned manually by choosing different values for weighting factor. Although the stability is guaranteed with the optimal control design, the re-design of the other active control schemes is less straight-forward and the research is ongoing to ensure such designs will meet both stability and performance requirements, e.g. by applying optimization technique to search for the best control structures and control gains.

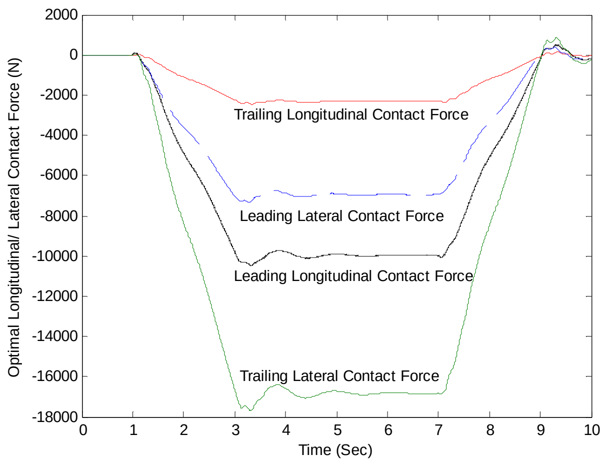

Figure 6: Original controller optimal controller /Leading Fail Hard

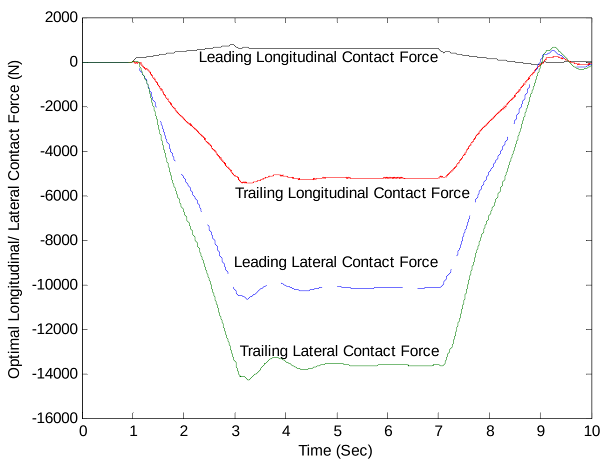

Figure 7: original optimal controller/Trailing Fail Hard

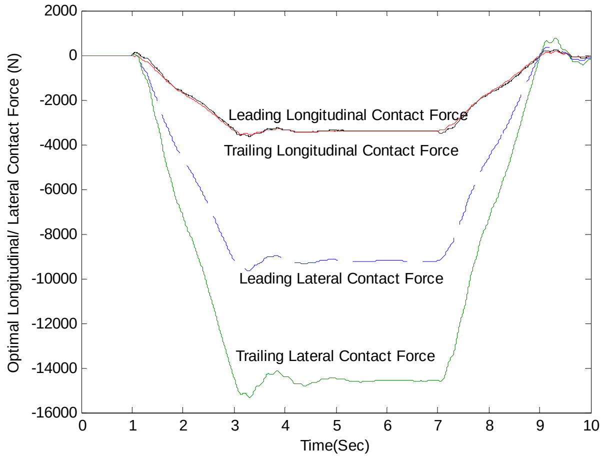

Figure 8: Retune manually optimal controller/ Leading fail Hard

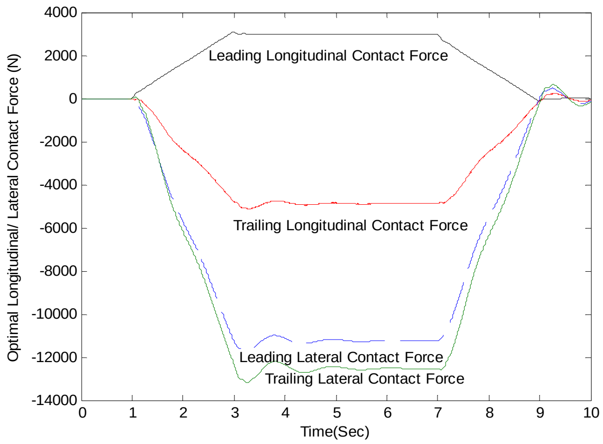

Figure 9: Retune manually Optimal Controller/ Trailing Fail Hard

B. *Fail-Soft*

The second type of the actuator malfunction is known as fail-soft, which is when one of the actuator is unconstraint from its control input. Control torque for the failed actuator in this scenario is zero and therefore not able to stabilize the kinematic mode of the wheelset. Figs.10a and 10b compares with the stability of the bogie with different active control schemes when one of the actuators fail-soft [3]. In Figs.10a and 10b, the critical speed is reduced to approximately 10m/s for all three active control schemes in the event of an actuator fault. Clearly, in the event of fail-soft condition the active controller would not able to stabilize the system and the operation speed of the system will have to be reduced very quickly to a very low level to avoid potentially dangerous situation if no other corrective actions are taken [3].

Therefore, in this scenario, the priority of fault tolerance for the active control systems is to preserve stability control in the presence of actuator fail-soft, with the curving performance a secondary design issue.

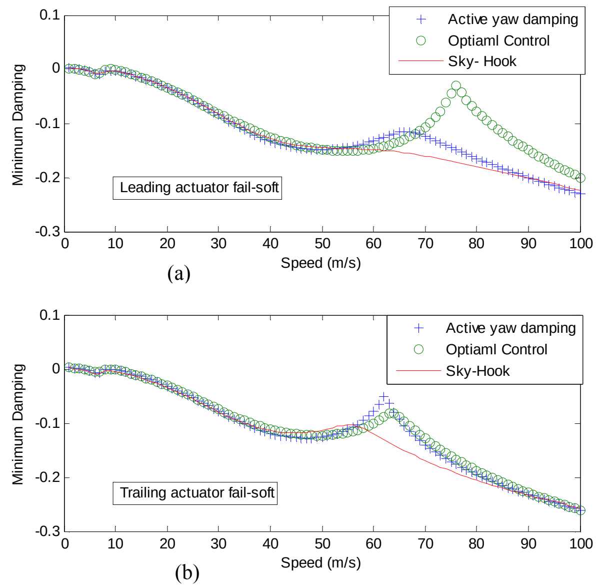

Figure 10: Stability of the different controller with leading/trailing actuator

In the event of fail-soft, the bogie stability is guaranteed if the number of control input is reduced from two actuators to one, and therefore more feedbacks are available for more sophisticated control design to provide desired control torque through the remaining actuator [3]. Fig.11 clearly reveals the bogie stability across a wide range of speed through re-designing of the optimal controller with one control input. It is also necessary to assess performance of redesign controller around the curve. Fig.12 gives the lateral and longitudinal contact forces at the wheel-rail contact points for the leading and trailing wheelset in the event of fail soft at the trailing actuator. The simulation result indicates clearly when the controller is re-designed with one control input (representing the remaining actuator), the perfect steering condition is achieved on the curved track, where the longitudinal contact forces at both leading and trailing wheelset are zero and the lateral contact forces of the two wheelsets are equal [13].

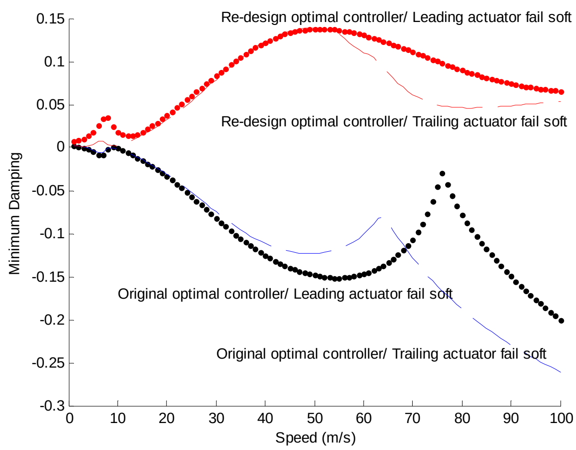

Figure 11: Stability of original / re-design optimal controller with leading/trailing fail soft in actuator

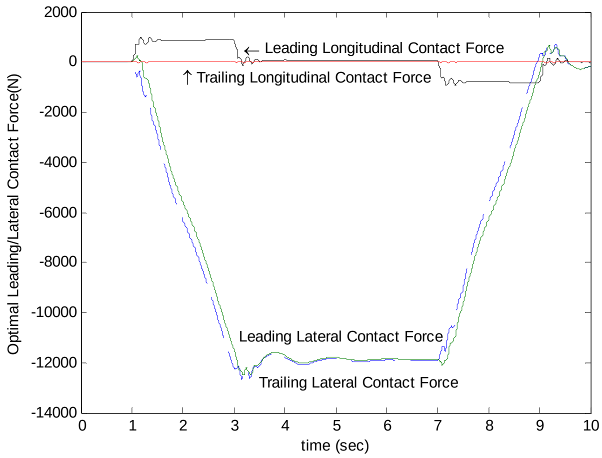

Figure 12: Curving performance of re-design optimal controller with trailing fail-soft

V. CONCLUTION AND FUTURE WORK

This paper has studied the fundamental fault tolerant control issues for actively controlled railway vehicles through analytical redundancy to guarantee controllability of the system in the event of actuator fault. A reconfiguration based strategy for managing both soft and hard faults has been investigated, focusing on solving instability and curving performance issues respectively. The design reconfiguration controller has been evaluated by their performance capability as evidenced in simulation results.

Research is ongoing to develop optimization technique to search for the best control gain and control structure in the event of fail-hard in such a way that ensure both stability and curving performance. However, there is clearly scope for extending the work other failure modes with different actuator configurations.

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APPENDIX A

Vehicle symbol and parameter in the simulation

Symbols	***Parameters***
$y_{w_1}, y_{w_2}, y_g, y_v$	Lateral diplacement of leading, trailing wheelset, bogie frame and vehicle body
$\psi_{w_1}, \psi_{w_2}, \psi_g$	yaw diplacemet of leading, trailing and bogie fram
V_s	Vehicle forward speed (50m/s)
m_w, I_w	wheelset mass (1250 kg) and yaw inertia (700 kgm^2), respectively
I_g, I_v	Half guage of wheelset(0.7 m), half spacing of axle(1.225 m)
r_0, λ	wheel radius(0.45 m), and conicity (0.2)
m_g, I_g	Bogie frame mass(6945 kg), and Yaw inertia (3153 kgm^2) respectively.
K_{sc}, C_{sc}	Secondary Lateral and longitudinal stiffness (511 kNm^{-1}), and damping(37 kNsm^{-1}) respectively.
K_s, C_s	primary Lateral stiffness (4750 kNm^{-1}), and damping (7705 N sm^{-1}) respectively.
m_v	Half vehicle mass (15000kg)
f_{11}, f_{22}	Longitudinal and lateral creepage coefficient (10MN).
R_1, R_2	Radius of the curved track at the leading and trailing Wheeslet(1250 m).
$\theta_{c_1}, \theta_{c_2}$	cant angle of the curved track at the leading and trailing wheelset (6°)
y_{t_1}, y_{t_2}	Track lateral diplacement for leading and trailing wheelsets, respectively
τ_{w_1}, τ_{w_2}	Controlled torque for leading and trailing weelset respectively.
g	Gravity (9.8 m/s^2)