Suspension Fault Diagnostics Using Vehicle Pitch and Roll Models

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ABSTRACT

The vehicle suspension system, including springs, dampers and stabilizer bars are critical to vehicle riding and handling experience. Automatic fault detection, isolation and failure prognosis of the suspension system will greatly improve vehicle perceived quality, serviceability and customer experience. In our previous work, a static diagnostic approach using a ramp with the known slope is proposed. Even though the method can effectively isolate the suspension system faults to each vehicle corner, it requires additional setups at dealerships. In this work, a passive approach using the vehicle pitch and roll models is presented, which can accurately isolate broken springs, leaking dampers, and broken stabilizer bars. Some enabling conditions are proposed to improve the overall algorithm robustness. The proposed solution is verified using the data collected from a test vehicle.

1. INTRODUCTION

Suspension systems, including springs, dampers and stabilizer bars, are critical to the vehicle ride and handling experience. For most passenger vehicles, the main type of springs is the coil spring. The coil spring is a compression spring, which transfers the kinetic energy to the potential energy. The stabilizer bar, also called sway bar or anti-roll bar, is a type of torsion springs, which reduces the vehicle roll angle during cornering. The vehicle damper, also named shock absorber, is a hydraulic device to absorb external shock impulses. Common vehicle damper contains check valves and orifices that control the internal oil flow with an internal piston. Due to the oil viscosity and the piston resistance, when the damper is compressed, the kinetic energy is transferred to heat, which is used to reduce the shock.

The suspension components degrade overtime. Most suspension components are diagnosed by the technicians' inspection during the maintenance, which associates with labor costs and maintenance time. A real-time and accurate diagnostic solution to detect and isolate the spring, damper and stabilizer bar faults, allows early detection before failures occur. As a result, it can greatly enhance customer satisfaction and perceived quality, vehicle serviceability and availability, which in turn can reduce the warranty/repair cost and fleet maintenance cost. The common failure mode for vehicle springs is the broken spring. It may be due to corrosion, fatigue or overloading. Several diagnostic approaches for the faulty springs are proposed in (Yin & Huang, 2015), (Pravin & Sivakumar, 2019) (Muhammad & Douglas, 2005) (Zhao & Wang, 2004), (Nozaki & Inagaki, 1998), (Börner, Straky, Weispfenning, & Isermann, 2000), and (Luo, Pattipati, Qiao, & Chigusa, 2008). The main methodologies include the data driven based vehicle vibration frequency analysis and physics-model based fault characterization. The readers are referred to our previous paper for more details of spring fault diagnostics (Du, Mai, & Sadjadi, Fault Diagnostics and Prognostics for Vehicle Springs and Stablizer Bar, 2020).

The most common fault of the vehicle damper is the oil leakage, due to the accidents or the material fatigue. When it occurs, the functionality of dampers is affected, which results in a bouncier driving experience, and the driver may feel vibration of the steering wheel. There exist some papers on vehicle damper diagnostics. One common method is the vibration-based frequency analysis. A diagnostic solution, which employs the Fast Fourier Transform (FFT) and the cross-spectrum graphs, is described in (BIALKOWSKI & KRĘZEL, 2017). Accelerometers are mounted on the vehicle, and the vibration response is analyzed. The vibration amplitudes of each test are calculated using the spectrum analysis. The coefficient is then calculated using the maximum amplitude of each test divided by the band power. For the faulty damper, the coefficient is significantly higher than the one generated from the healthy damper, which...
indicates that it is a good fault signature to detect the damper faults. Another method by monitoring the magnitude of the system vibration frequency is introduced in (Alcantara, Brooks, Lopez, Menendez, & Mendoza, 2013). The semi-active suspension scheme is built using the magneto-rheological damper. The damper faults can be identified from the information of specific frequency ranges and a road profile amplitude compensation. As a baseline, the road profile amplitude in frequency domain can be established using acceleration data with healthy dampers. The faulty damper causes the magnitude of the suspension system resonance peaks to increase, which can also be identified from the bode diagram. In another word, the suspension system with faulty dampers shows the higher magnitude of steady state frequency response when passing the same road profile, which is used for damper faults detection. Recently, machine learning techniques are also applied in damper diagnostics. One typical approach using the Convolutional Neural Networks is shown in (Zehelein, Pottmann, & Lienkamp, 2020). The vehicle driving data, such as longitudinal acceleration and wheel speeds, and the vehicle damper health states are collected and processed after trend removal. FFT is applied for the vibration frequency analysis. Frequency features, such as peak spectrum amplitudes and corresponding frequencies, of every unique vehicle configuration with different damper effects are classified. The actual classification is performed in the fully connected layer using the averaged frequency analysis data points. The experiment results show a high accuracy for damper fault detection. Another solution proposed in (Koláček & Dostál, 2013) employs the tire force information for damper diagnostics. The test vehicle is placed on the vibration platform with the known vibration frequency. The contact forces on the tire are measured to characterize the efficiency of external shock compensation by dampers. The experiments with the faulty damper show longer tire force oscillation time and higher frequency of the tire force variation. However, this solution requires a specific test platform and technicians to perform the measurement and analysis, which may be costly for a large fleet of vehicles.

For the stabilizer bar fault diagnostics, to the best of our knowledge, there is no paper published.

In summary, there is no established method that is both fast and accurate. To bridge the gap, we proposed a static ramp test scheme (Du, Mai, & Sadjadi, Fault Diagnostics and Prognostics for Vehicle Springs and Stabilizer Bar, 2020) in 2020 to quickly detect and isolate spring and stabilizer bar faults. Even though the diagnostic results are accurate (97.6% accuracy and 0% false positive rate on 82 test data sets collected from a test vehicle), this solution requires a ramp setup at each service location. In this work, we propose a passive monitoring solution to detect and isolate suspension faults on the go. The solution is based on the vehicle pitch and roll models on selected acceleration maneuvers or lane change/cornering maneuvers. For the rest of this paper, the details of the proposed solution are presented in Section 2, and the validation with real vehicle data are presented in Section 3. Section 4 concludes the paper.

2. SUSPENSION PASSIVE DIAGNOSTICS ALGORITHM

In this section, the suspension passive diagnostics algorithm is introduced, including the theoretical analysis using the pitch model and roll models, fault signature extraction, and the isolation algorithm.

The vehicle pitch model (Turco, Borodani, & Klaarenbeek, 1997) (Jin, Yin, & Chen, 2019) is used to represent the chassis dynamic behavior, when the vehicle is under an acceleration maneuver. In this scenario, the pitch angle is increasing. Shown in Figure 1, \( M, C_g \) and \( h \) are the total vehicle mass, the central gravity (CG) and the height of CG of the vehicle, respectively. \( F_1 \) and \( F_2 \) are the traction force for the rear tires and the front tires, respectively. \( a_x \) is the longitudinal acceleration. \( p \) is the pitch angle of the vehicle. \( K_p \) and \( C_p \) are the pitch stiffness constant and the pitch damping constant, respectively.

According to the Newton’s second law for rotation, the dynamic equation of vehicle chassis during acceleration maneuver can be represented as follow:

\[
I_p \ddot{p} + C_p \dot{p} + K_p p = M a_x
\]

(1)

where, \( p, \dot{p} \) and \( \ddot{p} \) are the pitch angle, pitch rate and pitch acceleration, respectively. \( I_p \) is the moment of inertia around the CG (lateral direction). \( K_p \) and \( C_p \) are the pitch stiffness constant and the pitch damping constant of the vehicle, respectively. As shown in the above equation, \( I_p, M \) and \( h \) are the intrinsic properties of vehicle. The value of \( \dot{p}, \ddot{p} \) and \( a_x \) can be measured by the inertial measurement unit (IMU). The \( p \) is calculated from the integration of \( \dot{p} \) in a given time period after bias removal. During the steady state, where the value of \( \dot{p} \) and \( \ddot{p} \) are very small, the values of \( I_p, \dot{p} \) and \( C_p \) are small and can be neglected. Then the pitch model Eq. (1) can be simplified as follows:

![Figure 1. Simplified schematics for a vehicle during acceleration (side view)](image)
For the roll model (Turco, Borodani, & Klaarenbeek, 1997) (Jin, Yin, & Chen, 2019), $M_s$ is the sprung mass of the vehicle, which is the vehicle mass portion supported by the suspension components and the tires. $h_s$ is the distance of CG height above the roll axis of the vehicle. $a_y$ is the lateral acceleration. $r$ is the roll angle of the vehicle. $K_r$ and $C_r$ are the roll stiffness constant and the roll damping constant of vehicle, respectively. According to the Newton’ second law for rotation, the dynamic equation during the cornering maneuver can be represented as follows:

\[
(I_x + M_s h_s)^2 \ddot{r} + C_r \dot{r} + K_r r = M_s h_s a_y \]

where, $r$, $\dot{r}$ and $\ddot{r}$ are the roll angle, roll rate and roll acceleration, respectively. $I_x$ is the moment of inertia around the CG (longitudinal direction). $K_r$ and $C_r$ are the roll stiffness constant and roll damping constant of vehicle, respectively. $I_x$, $M_s$ and $h_s$ are the intrinsic properties of the vehicle. The value of $\ddot{r}$, $\dot{r}$ and $a_y$ can be measured by the IMU. And $r$ is calculated from the integration of $\dot{r}$ after bias removal. During the steady state of the lane change or cornering maneuver, when the value of $\dot{r}$ and $\ddot{r}$ are very small, the values of $(I_x + M_s h_s)^2 \ddot{r}$ and $C_r \dot{r}$ are small and can be neglected. Then the roll model can be simplified as follows,

\[
K_r r = M_s h_s a_y
\]

Faulty stabilizer bars normally refer to the loose or the disconnection of the end link from the joint. When the stabilizer bar is disconnected, the stabilizer bar cannot provide necessary torsion force to balance the momentum, when vehicle is under the cornering maneuver. It will cause a bigger roll angle and may even lead to a vehicle rollover. Therefore $K_r$ is a good fault signature for stabilizer bar fault detection.

To achieve the real-time signal processing and generate the suspension diagnostics results, a diagnostic and isolation algorithm is developed, which includes two main parts, the data processing and fault diagnostics.

When the vehicle starts, the data processing is running continuously, which includes the data buffering, bias removal, maneuver detection and parameters calculation, as shown in Figure 3. The buffer is firstly reset, and the vehicle signals are continuously processed, including the $a_x$, $a_y$, $\dot{p}$, $\ddot{p}$, $\dot{r}$, $\ddot{r}$ and yaw rate ($\dot{\psi}$). The frequency of the data acquisition in this work is 100 Hz. The above signals are then automatically saved in a moving buffer that can store the latest 13 seconds of data. The pitch model and the roll model, discussed in the previous section, are then employed to analyze the data.

The data during the acceleration maneuver are recorded for the pitch model. The vehicle signals used in the pitch model are $a_x$ and $\dot{p}$. The enabling condition to determine an acceleration maneuver is shown as follows:
Where $T_1$ is a threshold to determine whether an acceleration maneuver occurs. All thresholds in the following paragraphs are presented using symbol $T$. Due to measurement bias, the vehicle signals should be centered each time. The signal centering (bias removal) is performed, using vehicle’s steady state values before the identified acceleration maneuver. The data point in the first 2 seconds, where 2 is a calibration parameter, are processed with the following enabling condition,

$$\max(|\dot{a}|) < T_2$$  
$$\max(|a_x|) < T_3$$  
$$\text{mean}(|a_x|) < T_4$$

If the above enabling conditions are satisfied, the mean value of each signal is considered to be the signal bias. In the steady state, the bias should be small, i.e.

$$|\dot{p}_{\text{bias}}| < T_5$$  
$$|a_{x,\text{bias}}| < T_6$$

If the calculated bias is larger than the above thresholds, the calculated bias may not be accurate. Therefore, the diagnostic algorithm will not be activated. If the above criteria are satisfied, all the data in the buffer are subtracted by the $\dot{a}_{x,\text{bias}}$ for the further roll stiffness estimation.

After all signals are centered, the $a_x$ and $\dot{p}$ signals from the 3rd second will be loaded to the pitch model. All centered $a_x$ and $\dot{p}$ signals are processed using a low-pass filter.

To calculate $\ddot{p}$, the derivative of $\dot{p}$ is calculated with the time interval ($dt$) of 0.1 second. To calculate $p$, $\dot{p}$ is integrated with $dt$ of 0.1 second, using Eq. (14).

$$\ddot{p}(t) = \frac{\dot{p}(t)-\dot{p}(t-1)}{dt}$$  
$$p(t) = \int_0^t \dot{p} \cdot dt$$

For the lane change or cornering maneuvers, the enabling condition for the bias removal is defined as follows,

$$\max(|\dot{r}|) < T_7$$  
$$\max(|\dot{a}_y|) < T_8$$

After the bias of $\dot{r}$ is calculated, all $\dot{r}_{\text{bias}}$ data are subtracted by the $r_{\text{bias}}$ for the further roll stiffness estimation.

During the acceleration maneuver, the pitch stiffness constant is firstly estimated. As stated in the previous paragraphs, the estimation is achieved during the steady state, where the values of $\dot{p}$ and $\ddot{p}$ are significantly small and can be neglected. A moving window is set to monitor the signals and determine the steady state. In this work, the size of the moving window is set to be 80, which is 0.8 seconds. In order to determine the steady state, the enabling conditions are set as below:

$$|\text{mean}(\dot{p})| < T_9$$  
$$|\text{mean}(\ddot{p})| < T_{10}$$  
$$\frac{|\max(p)-\min(p)|}{\bar{p}} < T_{11}$$
If the enabling conditions above are satisfied, the PitchIndex is used to label the first data point. However, the steady state does not ensure to lead to the good estimation of the pitch stiffness constant. There exist some corner cases causing the inaccurate estimation of the pitch stiffness constant.

Figure 4 shows one corner case – spring hysteresis. The maximum \( a_x \) value appears at around 2 seconds, but the steady state is detected at around 4 seconds. There is a significant decrease of \( a_x \) before the steady state. If \( a_x \) decreases, it reflects the fact that the spring may change from the compression condition to the extension condition, which leads to the spring hysteresis.

In this scenario, the pitch stiffness estimation may vary a lot. The following constraint is added to avoid the pitch stiffness estimation. When the maximum value of \( a_x \) signal is in front of the PitchIndex, the difference between the maximum \( a_x \) value and the \( a_x \) value at the PitchIndex point, \( a_{x,\text{diff}} \), should satisfy:

**Constraint 1:** \( a_{x,\text{diff}} < T_{14} \)

Similarly, there also exist corner cases that the pitch angle decreases before the steady state, which may also lead to the inaccurate estimation of the pitch stiffness constant. Another constraint is added to avoid such a phenomenon. If the maximum value of \( p \) is in front of the PitchIndex, the difference between the maximum \( p \) value and the \( p \) value at the PitchIndex point, \( p_{\text{diff}} \), should satisfy:

**Constraint 2:** \( p_{\text{diff}} < T_{15} \)

Based on the pitch model, \( a_x \) should be directly proportional to \( p \) during the acceleration. However, the opposite trend is found in some datasets, which is shown as Figure 5.

In Figure 5, \( a_{x,1} \) and \( a_{x,2} \) are the maximum \( a_x \) value and the \( a_x \) value where the PitchIndex locates, respectively. \( p_1 \) is the \( p \) value corresponding to the \( a_{x,1} \) value, which is not necessarily referred to maximum \( p \) value. \( p_2 \) is the \( p \) value, where the PitchIndex locates. The \( a_{x,\text{diff}} \) is positive, while the \( p_{\text{diff}} \) is negative, which indicates that \( a_x \) increases while \( p \) decreases. In this situation, the final estimation of the pitch stiffness constant is significantly small. To address this issue, we define \( \text{Abs}_\text{sum}1 \) as:

\[
\text{Abs}_\text{sum}1 = |a_{x,\text{diff}}| + |p_{\text{diff}}| \quad (25)
\]

If the \( \text{Abs}_\text{sum}1 \) is small, it indicates that variations of \( a_x \) and \( p \) are small and acceptable. Otherwise, the change trend of the acceleration and the pitch angle should be the same. The overall constraint is shown as below:

**Constraint 3:**

\[
a_{x,\text{diff}} \times p_{\text{diff}} > T_{16} \quad \text{or} \quad \text{Abs}_\text{sum}1 < T_{17} \quad (26)
\]

Constraint 3 is based on the maximum \( a_x \) value. Similarly, another constraint should be in place based on the maximum \( p \) value. \( p_3 \) is the maximum \( p \) value before the steady state. \( a_{x,3} \) is the \( a_x \) value corresponding to the \( p_3 \) value. \( a_{x,\text{diff2}} \) and \( p_{\text{diff2}} \) are calculated using following equations:

\[
a_{x,\text{diff2}} = a_{x,3} - a_{x,2} \quad (27)
\]

\[
p_{\text{diff2}} = p_3 - p_2 \quad (28)
\]

where \( a_{x,2} \) and \( p_2 \) are the corresponding values at PitchIndex point. We also define \( \text{Abs}_\text{sum2} \) as follows:

![Figure 4. Acceleration maneuver for the corner case 1 (Decrease of \( a_x \))](image1)

![Figure 5. Acceleration maneuver for the corner case 3 (Opposite trend of \( a_x \) and \( p \))](image2)
\[ \text{Abs. sum2} = |a_{x \text{diff}}| + |p_{\text{diff}}| \]  

(29)

And the constraint 4 is shown as follows:

**Constraint 4:**

\[ a_{x \text{diff}} \times p_{\text{diff}} > T_{18} \text{ or } \text{Abs. sum2} < T_{19} \]  

(30)

In some corner cases, \( a_x \) and \( p \) show large variation before the steady state, which causes challenges for us to determine the actual steady state. One example of such a corner case is shown in Figure 6. In the figure, index1 refers to the location where \( p \) value is 90\% of the maximum \( p \) value. The corresponding \( a_x \) value is labeled as \( a_x \) (index1). In this situation, \( a_x \) changes significantly before the steady state, while \( p \) slightly changes. The variation of \( a_x \) is larger than the variation of \( p \). Therefore, the pitch stiffness estimation will change significantly. In order to avoid such corner cases, the ratio 1 is defined as follows:

\[ \text{ratio} 1 = \frac{\max (a_x (\text{index1:PitchIndex})) - \min (a_x (\text{index1:PitchIndex}))}{\max (p (\text{index1:PitchIndex})) - \min (p (\text{index1:PitchIndex}))} \]  

(31)

The constraint 5 based on the ratio 1 is shown as below:

**Constraint 5:** \( T_{20} < \text{ratio} 1 < T_{21} \)  

(32)

Please note that the thresholds of above 5 constraints are determined after the data analysis from the large number of test vehicle data. If an acceleration maneuver satisfies these constraints, the pitch model will be activated to estimate the pitch stiffness constant \( K_p \). Otherwise, the pitch model will not be activated, and no estimation will be performed. To estimate the value of \( K_p \), the averages of \( a_x \) and \( p \) during steady state (0.8 seconds) are calculated. Then the value of \( K_p \) is estimated using the following equation based on Eq. (2).

\[ K_p = \frac{m_{\text{max}} a_x}{p} \]  

(33)

After the pitch stiffness constant \( K_p \) is estimated, the process of the pitch damping constant \( C_p \) estimation will be triggered. In this work, the estimation of \( C_p \) is performed using the linear regression. To perform the regression, the Eq. (4) and (5) are employed, where \( X \) is \( p \) and the \( Y \) is the terms on the left side of Eq. (4). The only unknown parameter is the \( C_p \). In this work, the data selected for the curve fitting are before the steady state during an acceleration maneuver. This is also called the ramp-up state. An example curve fitting interval is shown in Figure 7. In the previous step, the steady state is determined, and the average of \( p_{\text{avg}} \) in the steady state is calculated. The start point of the ramp-up state is the location where the value of \( p \) is around 10\% of \( p_{\text{avg}} \). Similarly, the end point of the ramp-up state is the location where \( p \) is around 80\% of \( p_{\text{avg}} \). Then all signals, including \( p \), \( \dot{p} \) and \( \ddot{p} \) between these two points, are selected for further analysis.

For the linear regression in this work, the length of the data is 80, which is 0.8 seconds data. After the ramp-up state is determined, another moving window is used to select the qualified data for curve fitting. The criteria are shown as following:

\[ \min (\ddot{p}) > T_{22} \]  

(34)

\[ a_x (\text{end}) > T_{23} \]  

(35)

\[ \frac{a_x (\text{end}) - a_x (\text{first})}{DLD} \times 100 > T_{24} \]  

(36)

where DLD is data length. In a moving window for the qualified data, the value of last \( a_x \) data point in the window should be larger than 2 m/s\(^2\), while the minimum \( \dot{p} \) is larger than 0.01 deg/s. And the calculated coefficient should be larger than 1.5 m/s\(^2\). If the above criteria are satisfied, all signals in this window are selected for curve fitting. The result is the estimated value of \( C_p \).
Different from the pitch model analysis, the roll model is activated, when the vehicle is running with certain speeds and under lane change or cornering maneuvers. An example of the lane change maneuver is shown in Figure 8. To detect the lane change or cornering maneuver, the yaw rate signal \( \dot{\psi} \) is used as an enabling condition. If \( \dot{\psi} \) at any timestamp is larger than 1 deg/s, it indicates that vehicle steering wheel is turned, and the lane change maneuver may occur. The vehicle signals within the first 2 seconds are loaded for bias removal, while the signals corresponding to the following 11 seconds are loaded for the roll model analysis.

As stated in the Eq. (7), the estimation of the roll stiffness constant is based on the simplified vehicle roll model, which can be applied during the steady state of a lane change or cornering maneuver. In order to determine the steady state, the \( a_y, \dot{r} \) and \( \ddot{r} \) signals are loaded and processed using the low pass filters. The \( r \) data are obtained from the previous step. In the steady state, the variations of above signals are comparatively small, and the Eq. (7) can be used to calculate the roll stiffness constant. A moving window with the size of 20 is used to look for the steady state. If, within a moving window, the selected vehicle data are all satisfied with the following criteria, this interval is considered as the steady state.

\[
\begin{align*}
|\text{mean}(\dot{r})| &< T_{25} \\
|\text{mean}(r)| &< T_{26} \\
\frac{\text{max}(r) - \text{min}(r)}{\text{mean}(r)} &< T_{27} \\
\frac{\text{max}(a_y) - \text{min}(a_y)}{\text{mean}(a_y)} &< T_{28} \\
|\text{mean}(a_y)| &> T_{29}
\end{align*}
\]  

(37) (38) (39) (40) (41)

After the steady state is determined, the \( a_y \) and \( r \) signals in this interval can be used to calculate the roll stiffness constant \( K_r \), using the following equation.

\[
K_r = \frac{M \dot{a}_y}{r}
\]  

(42)

The calculated \( K_r \) will be then saved in the buffer. The estimated the health indicators \( K_p, C_p \) and \( K_r \), are then saved in the buffer. Each calculated parameter is considered as one available result. In another word, there are 3 buffers to separately save the \( K_p, C_p \) and \( K_r \) results. The diagnostics algorithm continuously monitors the results in each buffer. If any buffer has at least 6 available results, the diagnostics algorithm will be activated, and the corresponding parameters will be loaded.

Please note that only the buffer with at least 6 available results is processed. If a buffer with less than 6 available results, the data in that buffer will not be processed with the diagnostics algorithm. After the results are loaded, the average of these results is calculated, which are labeled as \( K_{p,a}, C_{p,a} \) and \( K_{r,a} \), respectively.

If the \( K_{p,a} \) is smaller than the threshold \( T_{Kp} \), it indicates that the vehicle spring is healthy. Otherwise, it indicates that the spring is faulty, and the diagnostics result will be reported. If the \( K_{r,a} \) is available, which is generated from the previous 6 \( K_r \) results, \( K_{r,a} \) is compared with the threshold \( T_{Kr} \). If \( K_{r,a} \) is larger than \( T_{Kr} \), it indicates that the stabilizer bar is healthy.

If \( C_{p,a} \) is larger than the threshold \( T_{Cp} \), it indicates that the vehicle damper is healthy. Otherwise, the vehicle damper is considered as faulty and the diagnostics results will be reported. All these thresholds should be calibrated for different suspensions systems.

3. ALGORITHM VALIDATION

To verify the proposed diagnostics algorithm, the test data during 1,721 acceleration or lane change maneuvers are collected from a test vehicle with springs, dampers and stabilizer bars in different health conditions. Among these test data, 210 are under the condition of one faulty spring, 248 are under the condition of the disconnected stabilizer bar, 222 are under the condition of one leaking damper, and 558 are under the healthy vehicle condition. In this paper, only the single point failure is considered. Here the leaking damper means 50% or 75% loss of damper oil, and the faulty spring means the bottom coil of the spring is removed (front springs), or two/three coils of springs are clamped together (rear springs). For each acceleration maneuver, the longitudinal acceleration is between 0.2g and 0.4g. For each lane change maneuver, the lateral acceleration is between 0.1g and 0.3g.

With regard to spring diagnostics, 411 test data with healthy components, 80 test data from disconnected stabilizer bar and 222 test data from leaking damper are labeled as healthy.
spring data. 210 test data from faulty springs are labeled as faulty spring data. Our diagnostic results for the spring diagnostics are shown in Figure 9. In these three figures, the “ground truth” is the actual test vehicle condition, and the “prediction” is the diagnostic result from the algorithm.

With regard to stabilizer bar diagnostics, 558 test data with healthy components, 214 test data from faulty springs and 226 test data from leaking damper are labeled as healthy stabilizer bar data. 82 test data from disconnected stabilizer bar are labeled as faulty stabilizer bar data. Our diagnostic results for the stabilizer bar are shown in Figure 10.

With regard to damper diagnostics, 411 test data with healthy components, 210 test data from faulty springs and 248 data from disconnected stabilizer are labeled as healthy damper data. 222 test data from leaking damper bar are labeled as faulty damper data. Our diagnostic results are shown in Figure 11.

Due to the enabling conditions and constraints for the maneuvers, some test results are not qualified, and no diagnostic result is generated. For example, 83 test data are filtered out due to the large calculated bias before the acceleration maneuver, which fails to satisfy the enabling condition. All qualified results are processed using the moving average. Please note that each diagnostic decision is based on 6 qualified results. The algorithm generates 77 decisions for spring diagnostics, 25 decisions for damper diagnostics and 112 decisions for stabilizer bar diagnostics.

The overall accuracy of the algorithm is calculated as the total number of correct diagnostic results (the same as the ground truth) divided by the total number of vehicle tests. The accuracy is 84.4% for spring diagnostics, 92% for damper diagnostics and 100% for stabilizer bar diagnostics. The false positive rate (FPR) reflects the percentage of actual healthy cases that are incorrectly identified as faulty cases. The FPR is 30.43% for spring diagnostics, 10% for damper diagnostics and 0% for stabilizer bar diagnostics. The false negative rate (FNR) indicates the percentage of actual faulty cases that are incorrectly diagnosed as healthy cases. The FNR is 7.58% for spring diagnostics, 5.88% for damper diagnostics and 0% for stabilizer bar diagnostics.

To further explore the noise factors that may affect the performance of the proposed algorithm, some robustness tests are conducted. Main noise factors include acceleration on the ramp, low tire pressure, driving on the rough road and different vehicle weights.

In the previous tests, the acceleration maneuvers normally occur on the flat ground. However, if the acceleration maneuver occurs on uphill or downhill, the vehicle pitch angle changes according, which may affect the estimation of $K_p$ and $C_p$. To validate algorithm robustness, 15 acceleration
tests are performed on a 7.2% grade ramp. No false positive alert is found.

Tire pressure is another main factor that may affect the algorithm performance. Normally, the tire pressure of test vehicle is 35 psi. Low tire pressure may change the vehicle corner height and further affect the pitch angle. The first tests are performed using the healthy vehicle with the FL low tire pressure (22 psi). No false positive alert is found. The second tests are performed using the healthy vehicle with four tires low pressure (22 psi). No false positive alert is found. In summary, the low tire pressures (22 psi) do not affect the accuracy of the algorithm.

A rough road with uneven road surfaces is selected to perform the road tests, including acceleration and lane change maneuvers. No false positive alert is found. The last robustness test is to increase the vehicle weight. The normal weight of the test vehicle with a driver and a passenger is 1600 kg. In the robustness test, 320 lbs. (145 kg) sandbags are loaded to the rear seats of the vehicle, to simulate the situation of two more passengers on the vehicle. The total vehicle weight is now 1745 kg. Several road tests are performed, and no false positive alert is found.

4. CONCLUSION

A real-time and passive approach is developed in this work to detect and isolate broken springs, leaking dampers and disconnected stabilizer bars. We demonstrate that the spring faults can be characterized by the fault signature, vehicle pitch stiffness constant, which is estimated from the longitudinal acceleration and the pitch rate during the steady state of longitudinal acceleration. The damper faults can be characterized by the fault signature, vehicle damping constant, which is estimated from the longitudinal acceleration and the pitch rate during the dynamic state of longitudinal acceleration. The stabilizer bar faults can be characterized by the fault signature, vehicle roll stiffness constant, which is estimated from the lateral acceleration and the roll rate in the steady state of lateral acceleration. A set of unique enabling conditions is employed to improve algorithm robustness, which includes the stationary driving maneuver detection for sensor bias removal, stationary acceleration detection for stiffness calculation, dynamic acceleration detection for damping constant calculation and hysteresis scenario detection to improve robustness. A novel fault isolation process is proposed to generate accurate results. Compared with state-of-art approaches, our method is fast and accurate and is capable of isolating three different suspension failures.

Based on 214 diagnostics results generated from 1,721 test cases collected from a test vehicle, the accuracy of the suspension passive diagnostics algorithm is 84.4% for spring diagnostics, 92% for damper diagnostics and 100% for stabilizer bar diagnostics. The false positive rate is 30.43% for spring diagnostics, 10% for damper diagnostics and 0% for stabilizer bar diagnostics. In the future, more robustness tests will be performed. We will further study the algorithm performance variations among different vehicles and different programs. The final algorithm will be developed and integrated with other chassis diagnostics, e.g. the rotor diagnostics (Du, Mai, Kazemi, & Sadjadi, 2020) and the wheel bearing diagnostics (Feng, Du, & Salman, 2019).

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