Abstract:
In this research, for understanding vibration specifications of a multi-degree of freedom 3-axle rigid truck, the ride quality of the truck is investigated. The modeled truck is a Benz 2624 model. The analysis is considered for an off-road duty. This research produces a modeling procedure and simulation of the truck. A linear model of the system contains the seat and cab suspensions, rigid live axles, and also suspension geometries is used, the Lagrange’s equation is utilized to obtain the motion equations and system matrices. Then, a numerical central difference method is used to obtain system responses subject to the sinusoidal road excitations. Because of lack of knowledge in vehicle’s physical parameters, the dynamic properties of its components have been achieved by modeling the truck in SOLIDWORKS environment. Finally, a MATLAB based code is developed to calculate system time responses under different cases, when the truck is moving at low speed. With some modifications, the developed model can also be used for newer trucks, for that, it’s necessary to have accurate information for input data to be able to change the current model.

Keywords: Vibration analysis, Lagrange equations, Road excitations, System response, Multi-axles truck.

I. INTRODUCTION
The modeling procedure is an important part of engineering. There are two types of the modeling: Numerical and physical. Both types are widely used in engineering [1-12]. One of the application of modeling and analyzing in engineering is vibration modeling. For example, when the frequency is close to bridge’s natural vibrations, vibration has an important role in design and maintenance of bridge structures specially [13]. In timber pile bridges in rural areas as well as long span bridges made by steel and concrete material dangerous vibrations due to traffic, wind and earthquake loads could result to collapse [14-15]. In building industry, many construction elements such as floor slabs would face vibration problem which interrupt the comfort ability of the residents [16]. In some building system (e.g. timber slabs [17]) this issue would become a major concern and several solutions are proposed to mitigate this issue by using structural solutions [18-19]. Vibration analysis contributes to improving in many fields and products, e.g., aerospace [20-22], automobile, transportation, and so on. The most common goal is to improve product quality by identification and suppression of unwanted vibrations. Multi-axle truck is as a real-world example, which needs its vibration breakdown. The interaction between vehicles wheels and road surface causes a dynamic excitation. The elevation of the road surface uneveness and the vehicle speed causes different vibration levels [25-26]. Heavy vehicles produce the most perceptible vibrations. For describing the dynamic behavior of the vehicle with a vehicle model, the model is consisted of discrete masses, springs, friction elements and dampers[24-29]. When a linear model of vehicle is obtained, the calculation of axle loads is facilitated by utilizing Frequency Response Functions [24,26]. A deterministic function is used to express local road uneveness that shows the deviation of the travelled surface from a true planar surface. By the use of a Power Spectral Density the global road uneveness can also be expressed in a stochastic way [30-35]. Many models such as quarter, bicycle, half and full models of vehicle with different numbers of DoF have been investigated in vehicle dynamics [36-39]. Eight-DoF model for vehicles is one of the most famous models, including forward, lateral, yaw and rolling motion plus, four degrees of freedom for travel of each wheel[40, 41]. Multimode system dynamic models of vehicles have also been proposed in the literature. For example, Rahmani Hanzaki et al. proposed a methodology for dynamic analysis of a multimode system with spherical joints. They have considered a suspension system of a vehicle as an example for that [42]. Applying this methodology on a three-axle truck creates the complicated computation. Hence, other people also employed discrete model for the truck. For instance, Tabatabaei developed a 16-DoF nonlinear model an articulated vehicle, which is validated experimentally [43]. By acquisition of developed model by optimizing several components in the truck it is also possible to reduce air pollutants specially CO [44]. This paper presents a survey on the equations of motion, by utilizing Lagrange equations, to determine system responses subject to sinusoidal road excitations for a complete 3-D rigid three-axle truck model, i.e., Benz 2624 model. This analysis is useful to better understand the coupled motions of the wheels. The validation of our equations has been verified with ADAMS in the previous paper. The developed 19 DoF model can also be applied on other trucks by applying changes in material properties and adding estimations. The previous 19 DoF model which was used by Zeidi et al. [45, 46], was very useful and is also improved in the current study. Mesh topology which was used in the model that was elaborately discussed in Zeidi et al. [47-50], is used for
ADAMS modeling which is used for verification with MATLAB results. In those papers, mostly structured mesh where used in which unstructured mesh is also included in the cases that was hard to simulate with structured mesh.

II. MODELING THE THREE-AXLE TRUCK

Most reasonable techniques to obtain mass properties of the components of a manufactured vehicle is to use experimental methods, but these techniques are very costly. Therefore, in this work, to model a three-axle truck and to find masses, centers of mass, moments of inertia etc. the Solid works software is employed. These physical properties are highly necessary for dynamic simulation of the truck. Figures 1 and 2 show two views of the assembled model of the truck, and some of the truck’s components, respectively. In this part, the weighty components of the truck are modeled precisely, such as chassis, tires, differentials, cabin, springs etc. Non-homogeneous material is assigned to this model since differential consists of several material and precision of properties, which are obtained from this model are more acceptable.

III. OBTAINING THE EQUATIONS

For determining dynamic behavior of the mentioned three-axle truck, the Lagrange method is utilized. The truck is considered a 19-DoF mathematical model. As shown in Figure 3. M₁, M₂ and M₃ are the axles of the truck. Blue springs are considered on behalf of tires, red springs as leaf springs of the suspensions systems. Green springs are counted for connecting cabin to the frame and finally, purple spring is used to suspend driver’s seat with respect to the cabin. As the rests, W,θ, and φ illustrate displacement, roll, and pitch of the truck in this dynamic analysis. Hence, the 19 DoFs are as follow:

- Driver seat bounce, one degree: w₁₀⁶;
- Cab bounce, pitch and roll, three degrees; orderly w₁₀⁴, θ₁₀⁴, φ₁₀⁴;
- Chassis bounce (sprung mass), pitch and roll, three degrees; w₁₀₀, θ₁₀₀, φ₁₀₀, respectively:
- Front axle, its bounce and roll, two degrees; orderly w₁₀¹, θ₁₀¹;
- Intermediate axle, bounce and roll, two degrees; orderly w₁₀², θ₁₀²;
- Rear axle, bounce and roll; two degrees; orderly w₁₀³, θ₁₀³;
- 6 bounce motion of the 6 wheels; w₁, w₂, w₃, w₄, w₅, w₆; where w₁ and ; w₅are the bounce of left and right steer wheels, respectively; w₃ and w₄ are the bounce of left and right wheels of the middle axle, correspondingly; w₅ and w₆ are the bounce of left and right wheels of rear axle, respectively.

The vector of coordinates for the vehicle is written as:
\[
W_{19} = \begin{bmatrix}
    w_{106} & w_{104} & w_{100} & \varphi_{104} & \varphi_{100} & \theta_{104} & \theta_{100} & w_{101} & \theta_{101} & w_{102} & \theta_{102} & w_{103} \\
    \theta_{103} & w_{1} & w_{2} & w_{3} & w_{4} & w_{5} & w_{6}
\end{bmatrix}^T
\]

(1)
Figure 3. The scheme of the 19-DoF model for the truck

Figure 4(a) shows truck model in X-Z plane and distances between different important points. In addition, Figure 4(b) indicates the model in Y-Z plane and the related parameters.

Figure 4. CGs (Center of gravity) and other essential parameters of the model

Equations of Motion
The Lagrange equation is well-known in the following form for this system:

$$\frac{d}{dt}\left(\frac{dT}{dW_{19}}\right) - \left(\frac{dT}{dW_{19}}\right) + \left(\frac{dP}{dW_{19}}\right) + \left(\frac{dR}{dW_{19}}\right) = 0$$

(2)

where $T$, $P$ and $R$ are the kinematic, potential and dissipation energies of the system, respectively.

The kinetic energy of the system is as follow:

$$T = \frac{1}{2} M_1 (W'106)^2 + \frac{1}{2} M_6 (W'104)^2 + \frac{1}{2} M_8 (W'100)^2 + \frac{1}{2} M_3 (W'101)^2 + \frac{1}{2} M_2 (W'102)^2 + \frac{1}{2} M_3 (W'103)^2$$
\[
\frac{1}{2} l_{x} (\theta')^2 + \frac{1}{2} l_{b} (\theta')^2 + \frac{1}{2} l_{x} (\theta')^2 + \frac{1}{2} l_{1x} (\theta')^2 + \frac{1}{2} l_{x} (\theta')^2 + \frac{1}{2} l_{x} (\theta')^2 + \frac{1}{2} l_{cy} (\theta')^2 + \frac{1}{2} l_{by} (\theta')^2
\]

(3)

Moreover, the potential energy of the system is obtained as:

\[
P = \frac{1}{2} K_c (W_{106} - W_{105})^2 + \frac{1}{2} K_c (W_{45} - W_{31})^2 + \frac{1}{2} K_c (W_{46} - W_{32})^2 + \frac{1}{2} K_c (W_{46} - W_{32})^2 + \frac{1}{2} K_c (W_{47} - W_{35})^2 + \frac{1}{2} K_c (W_{48} - W_{36})^2 + 12K (W_{33} - W_{13})^2 + 12K (W_{34} - W_{12})^2 + 12K (W_{25} - W_{25})^2 + 12K (W_{26} - W_{26})^2 + 12K (W_{71} - W_{21})^2 + 12K (W_{72} - W_{22})^2 + 12K (W_{23} - W_{23})^2 + 12K (W_{24} - W_{24})^2 + 12K (W_{25} - W_{25})^2 + 12K (W_{26} - W_{26})^2 + 12K (W_{46} - W_{36})^2
\]

(4)

And the dissipation energy of the system is:

\[
R = \frac{1}{2} c_c (W_{106} - W_{105})^2 + \frac{1}{2} c_c (W_{45} - W_{31})^2 + \frac{1}{2} c_c (W_{46} - W_{32})^2 + \frac{1}{2} c_c (W_{47} - W_{35})^2 + \frac{1}{2} c_c (W_{48} - W_{36})^2 + 12c I (W_{33} - W_{13})^2 + 12c I (W_{34} - W_{12})^2 + 12c e\sigma(W_{17} - W_{17})^2 + 12c e\sigma(W_{18} - W_{18})^2 + 12c e\sigma(W_{25} - W_{25})^2 + 12c e\sigma(W_{26} - W_{26})^2 + 12c e\sigma(W_{36} - W_{36})^2
\]

(5)

By differentiating of T, P and R with respect to the coordinates and time according to eq. (2), equations of motion can be organized as:

\[
M \ddot{W} + C \dot{W} + K \dot{W} = 0
\]

(6)

In which M_{19}, K_{19} and C_{19} are orderless mass matrix, stiffness matrix and damping matrix of the 19 DoF of the truck-poster system model. In this equation, \( W_{19} \), \( W_{19} \) and \( W_{19} \) are acceleration vector, velocity vector and displacement vector of the 19 DoF truck-poster system model. In addition, system mass matrix, M_{19} which is a diagonal matrix, is calculated as follows:

\[
M_{19} = \text{diag} \left[ M_{1x}, M_{1y}, M_{1z}, M_{o1}, M_{o2}, M_{o3}, M_{o4}, M_{o5}, M_{o6} \right]
\]

Where, “diag” illustrates that the \( M_{19} \) is a diagonal matrix and \( M_{1} \) to \( M_{o6} \) are located on the main diagonal of the matrix. In this relation, \( M_{1} \) and \( M_{2} \) are masses of the seat and the driver, and the cab, respectively; \( I_{x} \) and \( I_{y} \) are inertia of the cab about \( X \) and \( Y \) axes, respectively. Correspondingly, in the following, \( M_{o6} \), \( I_{bx} \) and \( I_{by} \) point to sprung mass, inertia of the sprung mass about \( X \) and \( Y \) axes, respectively; Also, 1, 2, and 3 as the indexes in order point to the front axle, middle axle, and the rear axle of the truck. Similarly, \( M_{19} \) to \( M_{o6} \) indicate the masses of the front left to rear right wheels, as well. The 19 nonzero values have been obtained from the truck model in Solid works software utilizing mass properties. Now, the system damping matrix and stiffness matrix can be written in the following form:

\[
C_{19} = \begin{bmatrix}
C_{1,1} & C_{1,2} & \cdots & C_{1,19} \\
C_{2,1} & C_{2,2} & \cdots & C_{2,19} \\
\vdots & \vdots & \ddots & \vdots \\
C_{19,1} & C_{19,2} & \cdots & C_{19,19}
\end{bmatrix}
\]

\[
K_{19} = \begin{bmatrix}
K_{1,1} & K_{1,2} & \cdots & K_{1,19} \\
K_{2,1} & K_{2,2} & \cdots & K_{2,19} \\
\vdots & \vdots & \ddots & \vdots \\
K_{19,1} & K_{19,2} & \cdots & K_{19,19}
\end{bmatrix}
\]

The non-zero components of \( C_{19} \) and \( K_{19} \) are as follows:

\[
C_{1,1} = C_{v}, C_{1,2} = C_{2,1} = -C_{v}, C_{3,1} = C_{v}, e_{1}, C_{1,4} = C_{v}, d_{1};
\]

\[
C_{2,2} = C_{v} + C_{c1} + C_{c2} + C_{c3} + C_{c4}, \quad C_{2,3} = C_{3,2} = -C_{v}, e_{1} +
\]

\[
C_{c1, e_{2}} = -C_{v}, d_{1} - C_{c1}, d_{2} - C_{c2}, d_{2} + C_{c3}, d_{3} + C_{c4}, d_{3}, C_{2,5} = C_{v} - C_{c1} - C_{c2} - C_{c3} - C_{c4},
\]

\[
C_{c2, e_{2}} = C_{c2, e_{2}} - C_{c1} - C_{c2} - C_{c3} - C_{c4}, e_{2} + C_{c3, e_{2}} = C_{c2, e_{2}} - C_{c1} - C_{c2} - C_{c3} - C_{c4}, e_{2} + C_{c3, e_{2}} = C_{c2, e_{2}} - C_{c1} - C_{c2} - C_{c3} - C_{c4}, e_{2} + C_{c3, e_{2}} = C_{c2, e_{2}} - C_{c1} - C_{c2} - C_{c3} - C_{c4}, e_{2}
\]

\[
C_{2,6} = C_{c2, e_{2}} = C_{c2, e_{2}} - C_{c1} - C_{c2} - C_{c3} - C_{c4}, e_{2} + C_{c3, e_{2}} = C_{c2, e_{2}} - C_{c1} - C_{c2} - C_{c3} - C_{c4}, e_{2}
\]

\[
C_{2,7} = C_{c2, e_{2}} = C_{c2, e_{2}} - C_{c1} - C_{c2} - C_{c3} - C_{c4}, e_{2} + C_{c3, e_{2}} = C_{c2, e_{2}} - C_{c1} - C_{c2} - C_{c3} - C_{c4}, e_{2} + C_{c4, e_{2}} = C_{c2, e_{2}} - C_{c1} - C_{c2} - C_{c3} - C_{c4}, e_{2}
\]

\[
C_{2,8} = C_{c2, e_{2}} = C_{c2, e_{2}} - C_{c1} - C_{c2} - C_{c3} - C_{c4}, e_{2} + C_{c4, e_{2}} = C_{c2, e_{2}} - C_{c1} - C_{c2} - C_{c3} - C_{c4}, e_{2}
\]
\[ K_{2.1} = -K_s, \quad K_{2.2} = 4K_c + K_s, \quad K_{2.3} = -K_s, e_1, \quad K_{2.4} = K_s(2d_3 - 2d_2) - K_s, d_1, \quad K_{2.5} = -4K_c, \quad K_{2.6} = K_c(2b_5 + 2b_4); \]
\[ K_{3.1} = K_e, e_1, \quad K_{3.2} = -K_e, e_1, \]
\[ K_{3.3} = 4K_c, e_2^2 + K_s, e_1^2, \quad K_{3.4} = K_e, e_1, d_1; \]
\[ K_{4.1} = K_s, d_1, \quad K_{4.2} = -K_s, d_1 - 2K_s, d_2 + 2K_{3.3}, d_1. \]
\[ K_{4.3} = K_s, d_1, e_1, \quad K_{4.4} = K_s, d_1^2 + 2K_c(d_2 + d_3^2), \quad K_{4.5} = 2K_c(d_2 - d_3); \]
\[ K_{5.1} = -2K_c, d_2, e_2, \quad K_{5.2} = -2K_c, d_3, b_3, b_2 + 2K_c, d_3, b_3; \]
\[ K_{5.3} = -4K_c, K_{5.4} = -2K_c, d_3 + K_s, d_2 + 2K_c + K_t, + 4K_c, \quad K_{5.5} = 2(K_t + K_c, + K_s), a_1 + 2K_c, e_2; \]
\[ K_{5.6} = -2K_t, b_1 + 2K_t, b_2 - 2K_t, b_1, K_{5.7} = 2K_t, b_1, b_2 - 2K_t, b_2, \quad K_{5.10} = -2K_t, \quad K_{5.11} = 2K_t, a_3, a_4 + 2K_c, e_2, \]
\[ K_{6.1} = -2K_t, a_1, b_1 - 2K_t, b_1, a_2 + 2K_t, a_1, b_3 - 2K_t, e_1, b_4, \quad K_{9.1} = 4K_c, b_3, d_1, K_{9.2} = -2K_t, b_1, d_2 + 2K_c, b_1, d_3 + 2K_c, b_2 - 2K_c, b_2, \quad K_{9.3} = -2K_t, b_1 + 2K_t, b_2 + 2K_c, b_2 - 2K_c, b_2 - 2K_c, b_2, \quad K_{9.4} = -2K_t, b_1 + 2K_t, b_2 + 2K_c, b_2 - 2K_c, b_2, \quad K_{9.5} = -2K_t, b_1 + 2K_t, b_2 + 2K_c, b_2 - 2K_c, b_2; \]
\[ K_{9.6} = -2K_t, a_7, K_{9.7} = -2K_t, b_1, K_{9.8} = -2K_t, b_1, a_2; \]
\[ K_{9.9} = 2K_c, b_1, a_2 + 2K_c, a_1, b_3, K_{9.14} = -K_c, a_3, K_{9.15} = K_c, a_2, K_{9.16} = -K_t, b_1, b_2, K_{9.17} = -2K_t, b_1, b_2, \]
\[ K_{9.18} = -2K_t, a_1, a_2, a_3, a_4 + 2K_c, a_2^2, K_{9.19} = -2K_t, b_1, b_2; \]
\[ K_{10.1} = -2K_c, K_{10.2} = 2K_c, K_{10.3} = -2K_t, a_3, a_4 + 2K_c, a_2 + 2K_c, a_2, K_{10.4} = -2K_t, b_1, b_2, \quad K_{10.5} = -2K_t, b_1, b_2; \]
\[ K_{10.6} = -2K_t, b_1, a_2, a_3, a_4 + 2K_c, a_2 + 2K_c, a_2, \quad K_{10.7} = -2K_t, b_1, b_2, \quad K_{10.8} = -2K_t, b_1, b_2, \quad K_{10.9} = -2K_t, b_1, b_2; \]
\[ K_{11.1} = 2K_c, a_3, a_4 + 2K_c, a_2, K_{11.10} = -2K_t, a_3, a_4; \]
\[ K_{11.11} = 2K_c, a_3, a_4 + 2K_c, a_2, K_{11.12} = -2K_t, a_3, a_4; \]
\[ K_{11.13} = 2K_c, a_3, a_4 + 2K_c, a_2, K_{11.14} = -2K_t, a_3, a_4; \]
\[ K_{11.15} = 2K_c, a_3, a_4 + 2K_c, a_2, \quad K_{11.16} = -2K_t, a_3, a_4; \]
\[ \omega_d = A_i \sin(\omega_d, t + \phi_i) \]
\[ i = 1, 2, \ldots, 6 \]
\[ \omega_d = 2\pi(\frac{\psi}{\psi})^2 \]

Where \( A_i \) is road roughness magnitude in meter, \( \omega_d \) is drive frequency in rad/s, \( v \) is truck forward speed in m/s. \( L \) is Road surface wave length in meter and \( \phi_i \) is the phase angle of the n-th wheel in rad.

And the equation of motion is as below:
\[ M \ddot{W} + C \dot{W} + K \cdot W = f(t) \]

(8)

Where \( M \) is system mass matrix, \( C \) is damping matrix, \( K \) is stiffness matrix, \( W \) is acceleration vector, \( \dot{W} \) is velocity vector, \( \ddot{W} \) is displacement vector, all of the 13 DoF truck model and \( f(t) \) is road excitation vector, which is in the form of:

\[
\begin{bmatrix}
0 \\
0 \\
\vdots \\
0 \\
\end{bmatrix} = \begin{bmatrix} k_{w1}A_1 \sin(\omega_d, t + \phi_1) + k_{w2}A_1 \sin(\omega_d, t + \phi_2) \\
k_{w1}A_2 \sin(\omega_d, t + \phi_1) - k_{w2}A_2 \sin(\omega_d, t + \phi_2) \\
k_{w3}A_1 \sin(\omega_d, t + \phi_3) + k_{w4}A_1 \sin(\omega_d, t + \phi_4) \\
k_{w3}A_2 \sin(\omega_d, t + \phi_3) - k_{w4}A_2 \sin(\omega_d, t + \phi_4) \\
k_{w5}A_1 \sin(\omega_d, t + \phi_5) + k_{w6}A_1 \sin(\omega_d, t + \phi_6) \\
k_{w5}A_2 \sin(\omega_d, t + \phi_5) - k_{w6}A_2 \sin(\omega_d, t + \phi_6)
\end{bmatrix} \quad (13 \times 1)
\]

As mentioned system matrices in Equation (8) is 13×13 and the vector \( G \) (gravity vector) does not appear here, because equilibrium positions are chosen for the systems initial position.

For numerical solution and dynamic simulation, the central difference method has been used, it’s equation is as below:
\[ \ddot{\psi} = \frac{w_{t+\Delta t} - 2w_t + w_{t-\Delta t}}{2\Delta t} \]

(9)

\[ \dot{\psi} = \frac{w_{t+\Delta t} - 2w_t + w_{t-\Delta t}}{2\Delta t} \]

(10)

As depicted in the following figure, the road profile that is used in the simulation has a continuous sinusoidal variation:

**Figure 5. Road profile**

In this paper, for simulating a typical road condition, one drive frequency have been used. A low drive frequency is chosen which has a magnitude of 2 Hz. Two cases have been chosen with a various phase angle (ϕ). In the first case the left steer wheel and the right steer wheel have no phase difference which means \( \phi_{12} \) is equal to 0 and in the second case we have 90 degrees’ phase lag for the right steer wheel compared to the left one, which similarly means that \( \phi_{12} \) is equal to \( \pi/2 \). For comparison, the two cases with various phase angles are depicted in Figure 6.
Subject to the depicted phase angles, the phase angle of the nth wheel $\phi_n$ is evaluated as follow:

$$
\phi_1 = 0, \quad \phi_2 = \phi_1 - \phi_{12}, \quad \phi_3 = \phi_1 - \phi_{13}, \quad \phi_4 = \phi_2 - \phi_{24}, \quad \phi_5 = \phi_1 - \phi_{15}, \quad \phi_6 = \phi_2 - \phi_{26}
$$

Where $\phi_{12}, \phi_{13}, \ldots$ are the phase angle difference between wheel 1 and 2, wheel 1 and 3, etc. As following:

$$
\phi_{12} = 0 \text{ or } \frac{\pi}{2} \text{ (depend on case of study)};
\phi_{13} = \frac{2\pi(b_1 + b_2)}{L}, \quad \phi_{15} = \frac{2\pi(b_1 + b_3)}{L}, \quad \phi_{24} = \phi_{13}, \quad \phi_{26} = \phi_{15}
$$

$L$ is the road surface wave length in meter and $b_1, b_2, b_3$ are some geometric distance that have been showed in figure 4.

The drive frequency can be defined in two ways. For a particular road with steady wave length of $L$, by a higher drive frequency the truck runs in a higher speed, while by a lower drive frequency, the truck runs at a lower speed. analogously, if we consider that the truck speed is steady, by a higher drive frequency the road surface possesses short wavelength characteristics, while in contrary, by a lower drive frequency the road surface possesses long wavelength characteristics. Therefore, the definition of simulation results is strongly related to conditions and assumptions in the simulation.

In this paper, the road wave length has been considered to be fixed and the truck speed is set to allow value of 5m/s, and by considering two different $\phi_{12}$ settings this gives us two cases in total, for both cases we have a low drive frequency, $\omega_d = 2$Hz and$A_d = 0.05$m, for case 1, $\phi_{12}$ is equal to zero and for case 2, $\phi_{12}$ is equal to $\pi/2$. Finally, the simulation results are performed by programming the equations in MATLAB.

IV. RESULTS
The following figures show the system time responses under the two expressed cases.

**Case 1:**

![Case 1](image1)

![Case 1](image2)

![Case 1](image3)

Figure 7. System time response for seat and cab for $\phi_{12} = 0$

Figure 8. System time response for chassis for $\phi_{12} = 0$

Figure 9. System time response for axles for $\phi_{12} = 0$

In this case striking roll motions observed from the cab. Although the road excitations are symmetrical, the weight distribution is not symmetrical due to the offset of the driver-seat system. Thus, because of a slightly differ of the left side and the right side, the differential effect may be further magnified through the cab suspension system and then causing a significant roll motion in the cab. Albeit, the transient states for all DoF motions have a short duration.
Case 2:

In this case, all DoFs are showing short transient state excluding the chassis roll. The striking cab roll may be mostly because of the asymmetric excitations and also the possible effect of the asymmetrical weight distribution which discussed in Case 1.

V. CONCLUSION

The three-axle truck have been modeled in form of 19-DoF system. Although this model of truck is a linear model, it has some unique features including the cab suspension, the seat suspension and the suspension geometry, which are vital in ride modeling for heavy vehicles but are often disregarded. By a model in SOLIDWORKS software physical properties of the truck are evaluated. The equations of motion are derived by utilizing Lagrange equation. And to obtain the system responses subject to sinusoidal road excitations numerical central difference method is assisted. Finally, system’s time responses have been obtained under two cases for the truck in low speed motion, and this is useful to know the vibrating component of the truck. However, some of them may only be effective to this specific truck model, they do help for better understanding the traits of this kind of vehicles and also helps to understand how to develop a more realistic nonlinear model.

VI. REFERENCES


[14]. A Mohammadi, JH Gull, R Taghinezhad, A Azizinamin, 2014, Assessment and Evaluation of Timber Piles Used in Nebraska for Retrofit and Rating, Department of Civil and Environmental Engineering Florida International University Miami, Florida, NDOR Research Project


