Effects of the Bogie Mechanism on the Dynamic Behavior of Crawler-type Construction Machines in Traveling on Firm Grounds with Continuous Bumps*

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Abstract
In crawler-type construction machines, flexible suspension system like bogie mechanism type suspension is used to reduce and disperse the load acting on the running gear and to absorb the vibration from uneven ground. However, the quantitative effects of the bogie are not so clear. In this paper, by using a modeling and simulation method for crawler-type construction machine with bogie mechanism, which is developed by the authors, in traveling on firm grounds with continuous bumps, representative simulation cases to analyze the dynamic behavior were conducted. Then K-type and X-type bogie as flexible suspension and locked K-type and X-type bogie as rigid suspension were selected. In two bump pitch cases, effects of the bogie mechanism introduction and the bogie type on the dynamic behavior of the machine, such as the lower rollers load and the bouncing and pitching behavior were shown quantitatively and discussed.

Key words: Construction Machinery, Modeling, Comfortability in Riding, Crawler, Track, Bogie Mechanism, Traveling Simulation, Continuous Bumps, Dynamic Suspension Load, Bouncing, Pitching

1. Introduction
Crawler-type vehicles or tracked vehicles demonstrate high mobility and stability on uneven grounds. For this reason, it is widely utilized from civil uses, such as construction and agriculture, to military. It is general that as the suspension system of crawler type vehicles, while rigid (fixed) type is used in construction machines for the high stability at the working, flexible (movable) type like trailing arm type etc. used in military vehicles for the superior impact-absorbing performance at the high speed traveling. In recent years, also in construction machines, flexible type suspension system realized by bogie mechanism called K-type or X-type from the form as shown in Fig.1(a) and (b) has become to be used. The aim of the bogie mechanism introduction is to improve the durability of the traveling equipment and the driver comfort related to the vibration by the relief and distribution of the impact load from uneven ground. However, there are many unknown points in the load and the behavior when flexible suspension system is introduced into construction machines and in the influence on the dynamic characteristics of the whole machine. The quantitative effects of the introduction on those are not so clear.

In construction machines, flexible suspension system which can realize various demands,
those are vibration comfort as well as workability, mobility, stability and durability, with sufficient balance is called for. Although many studies on crawler-type vehicles with flexible suspension system have been done, military vehicles are set as the main object\(^{1(3)}\), and there is almost nothing for construction machines. Studies on construction machines equipped with flexible suspension system is desired because the mechanism, structure and requirements for construction machines differ from those of military vehicles. Authors have built a numerical simulation analysis method utilized for the rational design development of crawler-type construction machines with some flexible suspension system\(^{4(5)}\). To obtain useful knowledge for the optimization of the running equipment, by using the above method, the dynamic behavior was calculated in the case of machines having some typical bogie mechanism traveling over a large single bump when excessive class load acts on the machine. Such kind of case study is meaningful for the actual design. The calculation results were compared with the experimental ones and the validity of the calculation method has been confirmed.

In this paper, based on the simulation analysis method by the authors, some cases in which a construction machine travel on grounds with hard and continuous bumps are analyzed. Under these conditions, the machine receives medium or small class load repeatedly and steadily. The analysis under the conditions are fundamentally important for the estimation on long term durability of the running equipment and the comfort of the driver to steady state vibration. In the simulation, crawler-type construction machines having flexible suspension system by two kinds of bogie mechanisms shown in Fig.1 are investigated. The dynamic load and behavior of the suspension system and the body are calculated. From these results, the effects of the introduction of the bogie mechanisms and the difference between the effects by the bogie mechanism type are clarified quantitatively and discussed.

2. Simulation analysis method

2.1. Assumptions

In the analysis, main assumptions are as follows.

(1) Construction machine is the left-right symmetry for the traveling direction central axis and the motion is two-dimensional in the plane containing the horizontal direction and the vertical direction.

(2) Air resistance in traveling can be neglected.

(3) The machine travels on firm work ground where the running gear is exposed under burden conditions.

(4) The grouser tip of the crawler shoes contacts with the ground, because the machine travels on firm ground.

(5) The traveling condition is non-work straight traveling.

2.2. Features of the simulation model

Simulation models with the following features are constructed.

(1) Construction machine is treated as a multi-body system considering the generality.

(2) Bogie mechanism which affects the characteristics of loads acting on the suspension and machine body vibration is modeled as detailed as possible.

(3) Crawler belt is treated as a continuous coupling of individual track link and the restriction of relative rotation angle (angle of bend) between adjacent track links is taken into
account.
(4) Upper and lower rollers radius affecting the crooked state of crawler belt is considered.
(5) Supporting equipment of the driver cabin is taken into consideration.
(6) Mechanical properties of the work ground are modeled by viscoelastic plastic model in the shear direction and viscoelastic model in the normal direction considering the generality.
(7) Surface shape of the work ground is set by connecting the segments which connect some representative points as shown in Fig. 2(d).
(8) Mechanical properties of the work ground can be set for each ground element.

2.3. Numerical analysis method of the dynamic behavior
Using the model described in the preceding section, appropriate kinematic constraints and external forces acting on each component of the body are set(4)(5) and a mixed system of differential algebraic equations(6) is derived as follows.

$$\begin{align*}
\begin{bmatrix}
M & \Phi_q^T \\
\Phi_q & 0
\end{bmatrix}
\begin{bmatrix}
\ddot{q} \\
\lambda
\end{bmatrix} =
\begin{bmatrix}
Q^A \\
\gamma
\end{bmatrix}
\end{align*}
$$

where, $M$ is the mass matrix, $q$ is the generalized coordinate vector, $Q^A$ is the generalized external force vector, $\Phi_q$ is the Jacobian obtained from differentiating the constraint equation left side with $q$, $\lambda$ is the Lagrange multiplier vector, $\gamma$ is the right side term of the acceleration equation. The constraint conditions of each component of the machine needed to obtain $\Phi_q$ and $\gamma$ are described in the next section. Moreover, the forces acting on each part of the machine body that become each component of $Q^A$ are described in §2.5. Numerical analysis of the dynamic behavior of crawler-type construction machines are conducted by solving the Eq.(1) numerically using the Gauss elimination method and the Newmark $\beta$ method.

2.4. Constraint conditions on the machine components
The constraint conditions of the machine components constituting the simulation model shown in Fig. 2 are as follows.

(1) The track frame is combined rigidly with the body and the idler cushion.
(2) The rotation center of the driving wheel moves with sharing the specified node on the machine body and that of the driven wheel (idler) moves with sharing the node on the idler cushion.

(3) The swing center of the bogie and the rotation center of the upper wheels move with sharing the specified node on the track frame, respectively.

(4) The rotation center of the lower wheels move with sharing the node on the bogie.

(5) Adjacent track links to each other share a track pin and move.

2.5. Forces acting on the machine components

The following forces act on each component in the simulation model shown in Fig.2.

(1) Gravity

(2) Buffer torque and swing angle restriction torque of bogie mechanism

(3) Track pin frictional force

(4) Supporting force of driver’s cabin

(5) Crooked angle restriction torque of track links

(6) Contact reaction force between track links and wheels (driven wheel (idler), upper and lower wheels)

(7) Contact reaction force between track links and driving wheel

(8) Running driving force acting on track links

(9) Reaction force acting on track shoes from the traveling ground

The details of above-mentioned (7)∼(9) fundamental for the analysis of traveling on uneven grounds are explained in the following sections.

2.6. Contact reaction force between track links and driving wheel

Figure 3 shows a contact model of track links and driving wheel. Contact reaction force calculated from the following formula is made to act on the driving wheel center and track pins when $r_i^F$ and $r_i^R$ are the distance from the driving wheel center to the front track pin of the i-th track link and the rear track pin, respectively and $r_D$ is the pitch circle radius of the driving wheel.

$$F_{id}(t) = \begin{cases} -k_D \Delta L(t) - c_D \Delta \dot{L}(t) & , r_i^j < r_D \\ 0 & , r_i^j \geq r_D \end{cases} \quad (2)$$

$$F_{Di}(t) = -F_{id}(t) \quad (3)$$

where $k_D$ and $c_D$ are the spring constant and the damping coefficient of the contact part of a track link and driving wheel, respectively. Moreover,

$$\Delta L(t) = r_i^j(t) - r_D \quad (i: \text{Track link No.}, j: F,R) \quad (4)$$

2.7. Running driving force acting on track links

As shown in Fig.3, running driving force $F_{Di}$ expressed by the following formula is made to act on the center-of-gravity position of the track links in contact with the driving wheel.

$$F_{Di}(t) = \frac{T_D(t)}{N(t)r_i(t)} \quad (5)$$

where, $N(t)$ is the number of the track links in contact with the driving wheel, $r_i(t)$ is the distance between the driving wheel center and a track link center of gravity and $T_D(t)$ is the driving wheel output torque. $T_D(t)$ is determined for every time from driving-wheel rotation angle speed $w_D(t)$ and $T_D$-$w_D$ characteristic curve. In addition, a track link in contact with the driving wheel shall be filled with the following formula.

$$r_i^F < r_D^r \quad \text{and} \quad r_i^R < r_D^r \quad (i: \text{Track link No.}) \quad (6)$$

where $r_D^r$ is the driving wheel addendum-circle radius.
2.8. Reaction force of track shoe receiving from the ground

Figure 4 shows the contact model between a track shoe grouser connected with a track link and the ground. In order to take grouser thickness $t_G$ into consideration to the contact judgement of a grouser and the ground, the tip form of a grouser is approximated with the circle of diameter $t_G$. In Fig. 4, Q and R are intersections with the perpendicular line from the center P of the grouser tip approximation circle to the ground surface and the grouser tip approximation circumference, respectively. $R'$ expresses the point R in time $t - \Delta t$. The normal force $F_{ni}$ acting on the grouser tip of a track link $i$ at time $t$ is calculated from the following formula.

$$F_{ni} = -K_n z_{Gi} - C_n D_z / \Delta t$$

$$z_{Gi} = \frac{\| QR \|}{\| D_{x1} D_{z1} \|} = R'R$$

(7)

where $K_n$ and $C_n$ are the spring constant and the damping coefficient in the normal direction of the ground, respectively. $z_{Gi}$ is the sinkage amount of the grouser tip. $D_{x1}, D_{z1}$ are the movement amount of the grouser tip in the $x_{Gr1}'$ and the $z_{Gr1}'$ direction during $\Delta t$, respectively. The tangential force $F_{ti}$ acting on the grouser tip of a track link $i$ at time $t$ is determined from the following formula.

$$F_{ti} = \begin{cases} 
F_{ti}', & |F_{ti}'| < \mu_s |F_{ni}| \\
-\text{sign}(D_{x1}) \mu_d |F_{ni}|, & |F_{ti}'| \geq \mu_s |F_{ni}|
\end{cases}$$

(8)

where $K_t$ and $C_n$ are the spring constant and the damping coefficient of the ground in the tangential direction, respectively. $F_{ti}'$ is the static friction force. $\mu_s$ and $\mu_d$ are the static and the dynamic friction coefficients between a grouser and the ground, respectively.

3. Traveling simulation on firm ground with continuous bumps

3.1. General

When machines travel on firm ground with continuous bumps, the effects of the bogie mechanism on lower roller shafts dynamic load as the suspension load and the behavior of the track frame as the result of vibration transmission from the ground to the machine body through the bogie mechanism are analyzed by simulation. It is investigated together with the correlation with lower rollers behavior expressing the bogie behavior. The change of the effects of the bogie mechanism by the bump pitch is also examined.

3.2. Analysis conditions

K-type bogie mechanism and X-type bogie one are analyzed as representative bogie mechanism type. Those are called AK and AX-type concisely after this. In addition, it is
analyzed when K-type and X-type bogie mechanism are locked in order to examine the effect of the bogie mechanism existence. Those are called concisely LK and LX-type. Former two types are called A-type in which the bogie is activate and latter two types L-type in which the bogie is locked.

The main specifications of analyzed crawler-type construction machine are shown in Table1. In K-type bogie, the vertical stroke of a pair lower roller installed in the minor bogie is different as to the position relative to the major bogie. In the Table1, the stroke of the lower roller near the major bogie shaft on the track frame is described in the parenthesis. Figure 5 shows the wheel interval of the analyzed machine. Table2 shows main analysis conditions on traveling and ground. In Table2, average rotational angle velocity $\bar{\omega}_D$ of the driving wheel and average traveling speed $\bar{V}_T$ in steady traveling on continuous bumps are also shown. The values were calculated from the simulation. Each part dimension of the continuous bump ground model was determined as follows.

1. Bump pitch $L_B$ is made to be the double of track link pitch $L_{T1}$.
2. Bump height $h_B$ is set for each crooked part of the crawler belt that is crooked continuously in a zigzag form under the maximum limit angle not to contact the concavity between the continuous bumps. By the above setting, repetitive loads act steadily on the machine body from the traveling ground. For comparison, 3 times of $L_{T1}$ was also chosen as the bump pitch $L_B$. In this condition, the crawler belt falls into the concavity between the continuous bumps ground and the bogie mechanism work more easily. After this, the case of $L_B = 2L_{T1}$ is called case1 (Short pitch) and the case of $L_B = 3L_{T1}$ case2 (Long pitch). In Fig.6, ground model with continuous bumps is shown with machine model. The reduction gear in traveling was made to be the lowest gear and the accelerator to be the full throttle condition considering that the machine travels on the ground with large motion resistance.

<table>
<thead>
<tr>
<th>Table 1</th>
<th>Main specifications of crawler-type construction machine</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Item</strong></td>
<td><strong>Value</strong></td>
</tr>
<tr>
<td>Overall length</td>
<td>5.6 m</td>
</tr>
<tr>
<td>Overall width</td>
<td>3.48 m</td>
</tr>
<tr>
<td>Tractor mass</td>
<td>75.2 $\times 10^3$ kg</td>
</tr>
<tr>
<td>Working mass</td>
<td>100 $\times 10^3$ kg</td>
</tr>
<tr>
<td>Engine rated output</td>
<td>0.641 $\times 10^3$ kW</td>
</tr>
<tr>
<td>Ground contact area</td>
<td>6.198 m²</td>
</tr>
<tr>
<td>Mean ground pressure</td>
<td>0.158 MPa</td>
</tr>
<tr>
<td>Shoe width</td>
<td>0.71 m</td>
</tr>
<tr>
<td>Groover height</td>
<td>0.105 m</td>
</tr>
<tr>
<td>(Single groove)</td>
<td></td>
</tr>
<tr>
<td>Track link pitch</td>
<td>0.318 m</td>
</tr>
<tr>
<td>Crawler base</td>
<td>4.365 m</td>
</tr>
<tr>
<td>Track gauge</td>
<td>2.77 m</td>
</tr>
<tr>
<td>Lower roller radius</td>
<td>0.15 m</td>
</tr>
<tr>
<td>Lower roller (Upper K-type)</td>
<td>0.055 m</td>
</tr>
<tr>
<td>Lower roller (X-type)</td>
<td>0.028 m</td>
</tr>
<tr>
<td>Vertical stroke (Lower K-type)</td>
<td>0.211 m</td>
</tr>
<tr>
<td>Vertical stroke (X-type)</td>
<td>0.016 m</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 2</th>
<th>Main analysis conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Case</strong></td>
<td><strong>Ground conditions</strong></td>
</tr>
<tr>
<td>1</td>
<td>63 m</td>
</tr>
<tr>
<td>2</td>
<td>63 m</td>
</tr>
</tbody>
</table>
3.3. Dynamic load of lower rollers

3.3.1. General One of the main objectives of the introduction of the bogie mechanism for crawler-type construction machine is, as mentioned in the introduction chapter, to reduce and disperse the load acting on the running gear. As the load of the running gear, the dynamic load of the lower roller shaft (hereafter, call it just lower roller) is taken into notice and analyzed. The results are discussed from the viewpoint of the effect of the existence of the bogie mechanism and the effect of the bogie mechanism type. The simulation model of the machine in this paper has 7 lower rollers as shown in Fig.2 in the previous section. In the backward or forward of the front and rear end lower roller, there exists driving wheel or idler. For this, the dynamic load and the behavior of the lower rollers differ from those of the other rollers a little. However, it was confirmed that every lower rollers has fundamentally similar tendency of the dynamic load and behavior. So, in this section, analysis results on the 4th and 5th lower rollers are mainly discussed. Those rollers are a pair of roller coupled with a bogie and locate near the machine body center.

Figure7 shows the relationship of the 4th and 5th lower rollers dynamic load $F_{Lr}$ and those positions $z_{Lr}$ in the vertical direction with the rollers position $X$ in the horizontal direction obtained from simulation analysis. $F_{Lr}$ is made dimensionless by dividing the dynamic load by a half value of the machine weight load. The origin of the vertical direction position $z_{Lr}$ is $z_{Lr0}$ that is the position in the traveling start and it is divided by the bump height $h_B$ to be dimensionless. In the bottom part of the Fig.7, related ground condition is drawn. In the interval shown in Fig.7, the characteristics of those changes with time are similar as the machine travels in a steady state.

3.3.2. Effects of the bogie mechanism existence In Fig.7, the effects of the bogie mechanism existence on the lower roller dynamic load are shown by comparing the results of A-type bogie with those of L-type one.

![Graphs showing the relationship of $F_{Lr}$ and $(z_{Lr} - z_{Lr0})/h_B$ with $x$ for 4th and 5th lower rollers](image-url)
In the case 1
From Fig.7(a), the movement of the lower roller in the vertical direction is about \( h_B/3 \) for L-type. For A-type, it is about \( h_B \) for AK-type and about \( h_B/2 \) for AX-type. As for L-type, because the lower roller can not press effectively the crawler belt in the concavity between the continuous ground bumps, the dynamic load \( F^*_{Lr} \) is productive mainly only for the lower roller which passes on the top part of the bump. This leads to increase the upper peak value compared with that of A-type.

In the case 2
From Fig.7(b), the lower roller dynamic load \( F^*_{Lr} \) is very large on the bump for L-type but for A-type, it does not show clear extreme value on the bump. This means that the bogie works very effectively. The vertical motion \( z_{Lr} \) of A-type lower roller becomes bump height \( h_B \) and the roller can press the crawler belt sufficiently in the concavity between the continuous bumps.

Situation of the running gear when the dynamic load shows the maximum value
In Fig.7(a) and (b), the time when the fourth lower roller dynamic load \( F^*_{Lr} \) shows the maximum value was specified. At the time, the states of crawler belt, lower rollers and loads are shown in Fig.8 and Fig.9. The above mentioned situation of crawler belt pressed to the ground concavity by lower rollers is proven. In both figures, the vertical components of lower roller dynamic load and crawler shoe grouser tip load are shown by thick arrows. Then, the range of the values is wide. The arrow length is not simply proportional to the load value and modified

Fig. 8  States of crawler belt, lower rollers and loads when \( F^*_{Lr} \) of 4th roller reaches maximum value. (Case1)

Fig. 9  States of crawler belt, lower rollers and loads when \( F^*_{Lr} \) of 4th roller reaches maximum value. (Case2)
just to display in the limited writing space. To recognize the exact value, the numerical value is written together. It is demonstrated that the A-type can disperse sufficiently the dynamic loads along the area to contact compared with L-type.

(4) Effect to reduce the suspension load
From above-mentioned (1) and (2), it was shown that the maximum value of the 4th and 5th lower roller dynamic load as the suspension load can be reduced in the A-type in which the bogie mechanism work effectively. In order to confirm whether this can say in all lower rollers, the maximum value $F_{Lr,\text{max}}^*$ of the dynamic load in all lower rollers were investigated and is shown in Fig. 10. It can be said that the A-type can reduce the maximum value of the dynamic load compared to the L-type in most of the lower rollers.

(5) Effect to disperse the suspension load.
To investigate the quantitative effect to disperse the suspension load, time average $F_{Lr}^*$ of the variance, that is the degree of dispersion around the mean value, of the dynamic load in seven all lower rollers was calculated. The result is shown in Fig. 11. The variance of the dynamic load for A-type is about a half to a third time smaller than that for L-type. It is proven that the dispersion or distribution effect of the suspension load by the A-type is enough larger than that by the L-type.

3.3.3. Effect of the bogie mechanism type
In the previous section, it was shown that A-type has good reduction and dispersion effect of lower roller dynamic load $F_{Lr}^*$ compared to L-type. In this section, it is examined how the bogie mechanism type in the A-type influences the dynamic load.

(1) In the case 1
From Fig. 7(a) and Fig. 10(a), the maximum value of lower roller dynamic load $F_{Lr}^*$ of the AK-type is smaller than that of the AX-type. This is based on the merit of the following characteristic of lower roller of the AK-type to crawler belt in the case 1. From Fig. 7(a), the vertical motion quantity of the AK-type is about $h_B$ and that of the AX-type about $h_B/2$. The lower rollers of the AK-type can press more the crawler belt in the concavity between the continuous bumps in comparison with AX-type. In the case 1, the machine body almost travels basically on the upper surface level of the ground bumps. This is shown from the bouncing behavior of the track frame described in Fig. 12 of §3.4 mentioned later. Therefore, the lower stroke of the rollers in the bogie mechanism has an important role for the lower rollers to follow the crawler belt in the ground concavity. Lower stroke of lower roller of the AK-type is...
larger than that of the AX-type as shown in Table 1. The merit of the characteristic to follow the ground shape made by the large lower stroke is connected with the small value of the variance which demonstrates the good load dispersion shown in Fig. 11.

(2) In the case 2
In Fig. 7(b) and Fig. 10(b), comparing the maximum value of lower roller dynamic load $F_{Lr}$ of the AK-type with that of the AX-type, there is no great difference. In the case 2, the machine body almost travels basically on the concavity plane level of the ground. This is shown from the bouncing behavior of the track frame described in Fig. 12 of §3.4 mentioned later. For this reason, the upper stroke of the rollers in the bogie mechanism has an important role to travel on continuous bump ground. As shown in Table 1, both bogies of AK-type and AX-type have sufficient upper stroke for the bump height $h_B(=0.03 \text{m})$. From the vertical motion of the lower rollers in Fig. 7(b), in the case 2, it is shown that the lower rollers of both AK and AX-type sufficiently force the crawler belt in the concavity in the continuous bump ground. Additionally, it seems to be that the characteristic to follow the continuous bump ground of AX-type is a little better than that of the AK-type when for both type, there exists enough motion allowance of the stroke like in the case 2. This is because the pair lower rollers on the same bogie in AX-type can move individually to some extent, whereas those of AK-type move as a strict pair with less independence. This for AX-type appears as a small merit of load dispersion effect shown in Fig. 11.

3.4. Dynamic behavior of the track frame

3.4.1. General
In this section, vibration absorbing effect of the bogie mechanism is examined from the dynamic behavior of the track frame. Figure 12 shows the time history of the bouncing $z_F$ at the gravity center of the track frame and the pitching $q_{TF}$ about it respectively. $z_{F0}$ is the base position of $z_F$ and $q_{TF0}$ is the base angle of $q_{TF}$. Those are the values when the machine starts to travel. And the bouncing amount is made dimensionless by dividing it by traveling ground bump height $h_B$. Taking $q_{TF0}$ as the base angle level, the counterclockwise direction of the pitching is set as the positive direction. In Table 3, total amplitude of main vibration of the bouncing $z_F$ at the track frame center of the gravity and the pitching $q_{TF}$ about it in those time history is shown. It was estimated from Fig. 12. Table 4 shows r.m.s. value of the bouncing acceleration of the gravity center of the track frame.

3.4.2. Effect of the bogie mechanism existence
(1) In the case 1
From Fig. 12(a) and Table 3, the total amplitude of main bouncing for A-type is a little larger

![Fig. 12](image-url)
than that for L-type. As for main pitching, the total amplitude for A-type is considerably larger than that for L-type. Although the reduction and dispersion of the suspension load are done by introducing bogie mechanism, it should be noticed that the flexibility of the bogie mechanism may induce especially the pitching vibration. By contrasting the bouncing behavior of A-type in the case with that in the case2, it can be understood that in the case2 the machine body almost travels on the level of ground bump upper plane. In this condition, the machine body is in an unstable state and the pitching vibration is easy to occur. Then, the frequency of the pitching vibration corresponds to that of the excitation due to the continuous bumps on the ground. It is similar in the case2 described next, too.

(2) In the case2
In Fig.12, it is shown that the machine body travels almost on the level of the ground concavity plane by contrasting the bouncing behavior of A-type in the case2 with that of A-type in the case1. Although the lower roller configuration of AK is the same as that of LK and the same for AX to LX, for both AK and AX, the machine travels with fending off the ground bumps using the sufficient upper stroke of lower rollers to the ground bump height. Therefore, as shown in Table3, the total amplitude of the pitching as well as the bouncing of AK decreases in comparison with that of LK. The situation is similar for AX to LX.

(3) Riding comfort for vibration
Table4 shows r.m.s. value of the bouncing acceleration at the gravity center of the track frame which is related to vibration riding comfort. It is proven that comparing A-type results with L-type ones, by introducing bogie mechanism, r.m.s. value of the bouncing acceleration can be reduced except for AX-type in the case1. As for AX-type in the case1, an additional note is described in the next section.

3.4.3. Effects of the bogie mechanism type
In this section, the effect of the difference of the bogie mechanism type on the track frame behavior is examined.

(1) In the case1
From Fig.12(a) and Table3, on both bouncing and pitching behavior, those total amplitude for AK-type, in which the lower stroke of lower rollers has more allowance in comparison with AX-type, is smaller than that for AX-type. From Table4, r.m.s. value of the bouncing acceleration at the track frame for AK-type is also considerably smaller than that for AX-type. In the case1, the r.m.s. value of the bouncing acceleration for AX-type is not so small. This is related to the effect of the impact load occurring by the collision of bogie and track frame, because there is no buffer material like rubber pad in the AK-type between bogie and track frame.

(2) In the case2
From Fig.12(b) and Table3, the total amplitude of the bouncing vibration for AK-type is smaller than that for AX-type, whereas that of the pitching vibration for AK-type is larger than that for AX-type. The r.m.s. value shown in Table4 of the bouncing acceleration for AK-type is a little larger than that for AX-type. In the case2, the upper stroke of the lower rollers is necessary for the vibration absorption from the reason mentioned previously. In the case2, for both AK-type and AX-type, the upper stroke of the lower rollers has enough allowance to the bump. Therefore, the superiority of AK-type for AX-type observed in the case1 decreases.
4. Conclusions

The dynamic behavior of crawler-type construction machine with representative bogie mechanisms in traveling on firm ground with continuous bumps was analyzed by simulation. The effect of the existence of the bogie mechanism and the bogie mechanism type on lower roller dynamic load and track frame behavior was examined. The effect to reduce and disperse the suspension load by the bogie mechanism and the effect to absorb the vibration input from the ground was quantitatively shown and discussed with the effect of the ground bump pitch. Due to the large upper and lower stroke of lower rollers, the K-type bogie mechanism has the characteristic for the lower rollers to follow the track link effectively. This characteristic produces superior effect to reduce and disperse the suspension load and to absorb the vibration. However, it should be taken into account for the optimum design that the machine body pitching behavior may be induced when the ground bump pitch is short like in the case1 in this paper.

In the future, to obtain the knowledge useful for the further optimization of traveling systems, by changing the dimension and properties condition of bogie, ground condition and working condition except for the bogie mechanism type, their effects on the dynamic behavior of construction machine would be clarified and discussed. And the consideration from the viewpoint of a balance with working performance would be also deepened. Finally, the authors express great appreciation to staff members in development division, Komatsu Ltd. for providing much valuable materials and advices and supporting this study.

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