Runaway characteristics of gantry cranes for container handling by wind gust

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Abstract
Gantry cranes used for container handling at container terminals (hereinafter referred to as cranes) have experienced unexpected runaways caused by wind. In the present paper, a method of dynamic simulation analysis for runaway of cranes caused by wind is described and the dynamic behavior of a standard-size crane while approaching runaway and after runaway as a result of a wind gust was clarified. In the dynamic simulation analysis, the crane was modeled by combining three-dimensional finite elements. The interaction forces generated between the wheels and the rails and the braking force of the rail clamps used to prevent crane runaway as a result of wind were modeled and calculated. The time change of wind velocity of a wind gust was modeled in terms of the increase ratio and the time change rate, and the wind load acting on the crane was calculated. Based on these models, a standard-size crane was analyzed herein. According to the analysis results, when the gust acted in the rail direction under the condition in which the friction coefficients between the pads of the rail clamps and the rails and those between the wheels and the rails decreased, the sea-side clamp pads and driving wheels entered the slip state, after which the land-side clamp pads and driving wheels entered the slip state. Then, the entire crane started to run away. Due to the time lag like this, the crane may run away even though the total wind load is smaller than the total resistance force acting to prevent runaway, which is simply calculated as the sum of the maximum static friction forces generated between the clamp pads and the rails and between the wheels and the rails. Moreover, although the increase ratio of the wind velocity is small, if a wind gust having a large time change rate blows and the fluctuation of the inertial force in the rail direction becomes large as a result of the swinging of the upper part of the crane, the probability of crane runaway increases when the inertial force increases in the runaway direction.

Key words : Materials handling equipment, Gantry crane, Dynamic behavior, Wind gust, Wind load, Inertial force, Friction, Runaway, Numerical simulation, FEM

1. Introduction

Container ships have become larger with the increasing use of marine container transportation, and therefore the number of containers being carried is increasing. For this reason, the machinery used for container handling at container terminals, which are the delivery base for containers, must become more efficient and safer. Cranes are the most widely used machinery for handling containers in container terminals. Since cranes have become larger and are installed on quays outdoors, they must withstand the natural environment, especially winds. Occasionally cranes experience unexpected runaway as a result of strong winds, resulting in serious accidents, partly because their weight is reduced for the purpose of energy savings (Accident investigation board of runaway container cranes at the Port of Omaezaki, 2010). In order to prevent such accidents, it is important to establish guidelines for wind-resistant design and operational management of cranes based on dynamic simulation analysis while sufficiently taking into account
wind conditions at container terminals and contact conditions between the pads of the rail clamps and the rails and those between the wheels and the rails, coupled with conventional measures and guidelines based on past experience.

In studies on wind loading of cranes (hereinafter referred to as wind loading), some researchers conducted wind tunnel tests and analyzed the wind load acting on scale models of a crane ([Yamanaka et al., 1997], (Ohmi et al., 1998), (Scarabino et al., 2005), (Lee Seong-Wook et al., 2006), (Kang and Lee Sang-Joon, 2008), and (Lee Sang-Joon and Kang, 2008)]. Han calculated the wind load acting on the crane through a numerical simulation and compared the numerical results with the results of a wind tunnel test (Han, D. and Han, G., 2011). In a study on the behavior of a crane under wind loading, Konig measured the body vibration of a portal crane, which carries heavy loads similar to those carried by cranes for container handling, which are considered herein (Konig et al., 1980). One of the present authors conducted a travel simulation analysis for a tire-type portal crane used as container handling machinery in container terminals (Takehara et al., 2000). They analyzed the influence of wind load on traveling performance. As described above, a number of researchers have investigated the wind load acting on cranes, the behavior of a portal crane under wind loading, and the influence of the wind load on the traveling performance of a tire-type portal crane. However, there has been no study on the simulation analysis of crane runaway as a result of wind loading.

The present study attempts to establish guidelines for the prevention of runaway cranes and for wind-resistant designs as a safety measure. As such, we conducted a dynamic simulation analysis of a crane installed in a windy environment at a container terminal and investigated the dynamic behavior of the crane approaching runaway and after runaway. In a previous study on crane runaway due to wind, we conducted dynamic simulation analysis on a crane subjected to a steady horizontal wind in the direction of the rail, and investigated the dynamic behavior of the crane during runaway and the interaction force generated between wheels and rails [(Abe et al., 2011a, 2011b) and (Takahashi et al., 2013a)]. In addition, we also conducted an analysis on a crane that received a steady horizontal wind in an oblique direction with respect to the rail direction (Takahashi et al., 2013b). However, unsteady winds such as gusts, in which the wind velocity increases rapidly, may blow in actual container terminals. Depending on how the wind velocity increases, a crane may start to run away. The purpose of the present paper is to describe a method of dynamic simulation analysis based on a study by one of present authors (Takehara et al., 2000), and to clarify the dynamic behavior of a crane approaching runaway and after runaway by wind gust.

First, we compared the results of the dynamic simulation analysis under the assumption that crane runaway was caused by wind and the results of an image analysis of the runaway of an actual crane due to wind. The fundamental reliability of the dynamic simulation analysis method was confirmed from the standpoint of motion characteristics. We then modeled the time change of the wind velocity based on the increase ratio and the time change rate and conducted a dynamic simulation analysis under the assumption that the crane receives a wind gust in the rail direction. From the obtained results, we initially discussed the outline of the runaway characteristics of the crane. In addition, the behavior of the system consisting of the wind, the crane, and the rail was shown with the distributions of the wind load that the crane received, and the swinging of the crane and the slipping of the clamp pads and the wheels were considered. Finally, we showed the relationship between the characteristics of the swinging of the crane and the driving force of the runaway consisting of the wind load and the generated inertial force. Moreover, we clarified the influence of the time change rate of the wind velocity on the runaway characteristics of the crane.

2. Description of the crane and classification of its states

Figures 1(a) and 1(b) show schematic diagrams of the crane and traveling unit considered in the present study. The crane travels on rails set on the quay by traveling units with wheels. A trolley traverses a girder and a boom, together with an operation cabin, and a spreader loads and unloads containers. As shown in Fig. 1(a), T.C.G. indicates the overall center of gravity of the crane.

The states of a crane can be classified into the operation state and the stop state according to the standards of the Japan Crane Association (JCAS, 2009a). The operation (in-service) state is the state in which the crane is operating, including the state in which the crane is on standby while its power is on because workers are waiting for instructions, or for other reasons. The stop (out-of-service) state is the state in which the power to the crane is turned off, the operator is away from the crane, and the proper measures are taken to prevent runaway of the crane at anchoring position. In the present paper, the operation state is further classified into the cargo-handling state, in which the crane is handling containers, and the cargo-handling standby state, in which cargo handling is stopped because workers are
waiting for instructions or for other reasons.

3. Method of dynamic simulation analysis
3.1 General

We assumed the case in which a crane is subjected to wind in the rail direction and conducted a dynamic simulation analysis in which the crane was modeled by three-dimensional beam elements using FEM. The following motion equation was solved using the Newmark β method (Newmark, 1959) to calculate the behavior of various parts of the crane:

\[
[M]\ddot{u} + [C]\dot{u} + [K]u = F - F_c
\]  

(1)

For Eq. (1), the mass matrix \([M]\) was found using the consistent mass method, and the damping matrix \([C]\) was found based on proportional damping. Since dynamic simulation analysis of the runaway crane involved a large displacement, a rotation matrix (Maeda and Hayashi, 1976) for the displacement increment per unit time was used to divide the displacement into rigid displacement and elastic displacement. Since the accumulated elastic force, \(F'_e\), was handled in the incremental system, it was used to consider the elastic force resulting from elastic displacement before the increment. Here, \(F'_e\) was calculated by multiplying the element stiffness by the elastic displacement, which is the accumulation of the elastic displacement increment per unit time, \(\Delta \epsilon\). The dead weight, the interaction force between the wheels and the rails, the braking force of the rail clamps, and the wind load were substituted into the external force terms in Eq. (1). The methods used to model the following items are described in the following section: the crane, the interaction between the wheels and the rails, the rail clamp braking force, the wind environment around the crane, and the wind load.

3.2 Crane model

Figure 2(a) shows a schematic diagram of the crane model. In order to analyze the deformation behavior of the crane, the crane was modeled using three-dimensional beam elements, as shown in Fig. 2(a). In the figure, X, Y, and Z are the longitudinal direction of the rails, the lateral directions of the rails, and the vertical direction, respectively. The
following assumptions were used in modeling.

1. Joints between elements are assumed to be rigid joints or pin joints based on the types of joints used in actual cranes.
2. The machinery housing is modeled by the three-dimensional beam elements, and the mass and the surface area that is exposed to wind are considered.
3. The operation cabin, the trolley, and the spreader are assumed to be an integral structure that is modeled in three-dimensional beam elements, and the mass and the surface area that is exposed to wind are considered.
4. Annexed structures, such as stairs and elevators, are not modeled.

Figure 2(b) shows a schematic diagram of the traveling units and rail clamps of the crane model. The nodes at the lower-end of the crane model were assumed to be the center of the contact surface between the wheels and the rails and the center of the contact surface between the pad of the rail clamps and the rails, and the corresponding constraint conditions are described in Sections 3.3 and 3.4.

3.3 Model of contact interaction between wheels and rails

3.3.1 General

In order for the crane to travel on the rails, it is equipped with traveling units under the lower-end of each leg, as shown in Fig. 1(b). Each traveling unit is equipped with driving wheels and driven wheels. A DC motor was assumed to be connected to the driving wheels through a reduction gear, and the DC motor was equipped with an electromagnetic disk brake. Under these conditions, the contact interaction force in each direction between the wheels and the rails was modeled as follows.

3.3.2 Vertical direction

Figure 3 shows the model of the contact interaction between the wheels and the rails in the vertical direction. This model consists of a spring, a damper, and a joint that is separated when a tensile force acts on the joint, in the vertical direction. The vertical force (hereinafter referred to as the wheel load, \( W_{ZcR} \)) that the wheel tread receives acts on the node assumed to be a contact surface between the tread of the wheel and the top of the rail (hereinafter referred to as the wheel node), and is calculated using following equation:

\[
W_{ZcR} = \begin{cases} 
-K_{Zc}d_{Zc} - C_{Zc}d_{Zc} & (R_{Zc} \geq 0) \\
0 & (R_{Zc} < 0)
\end{cases}
\]

where \( K_{Zc} \) and \( C_{Zc} \) are the spring constant and the damping coefficient in the vertical direction between the wheel tread and the top of the rail, respectively, and \( d_{Zc} \) and \( d_{Zc} \) are the displacement and the velocity in the vertical direction at the wheel node, respectively. In the present paper, the spring constants are defined based on the relationship between the load and the elastic deformation amount of the wheel tread and the rail as calculated by the contact analysis (ANSYS Mechanical). For suppressing the unnecessary vibration in the vertical direction, the value of \( C_{Zc} \) was decided without the divergence of solution in the first step of the dynamic simulation analysis in which the force of gravity acts on the crane.

3.3.3 Longitudinal direction

A static friction force is generated in the rail direction at the contact surface between the wheel tread and the rail when the crane resists runaway caused by a driving force consisting of the wind load and the inertial force generated by the swinging of the crane (hereinafter referred to as the runaway driving force). When a crane runs away, the driving wheel is locked by the electromagnetic disk brake and a dynamic friction force by slipping acts between the driving wheels and the rails. Moreover, a rolling friction force is generated at the contact surface between the driven wheels and the rails.

Figure 4 shows the model of the contact interaction between the wheel and the rail in the rail direction. This model consists of a spring, a damper, and a slider for setting the limit of the static friction force, in the rail direction, as shown in Fig. 4. The friction force acting on the wheel node in the rail direction that is generated between the wheel tread and the rail, \( R_{xy} \), was calculated by the following equation:
where $K_{WX}$ and $C_{WX}$ are the spring constant and the damping coefficient in the rail direction between the wheel tread and the top of the rail, respectively, $d_{WX}$ and $\dot{d}_{WX}$ are the displacement and velocity in the rail direction at the wheel node, $\mu_{WX}$ and $\mu_{WX}$ are the dynamic friction coefficient between the driving wheel tread and the rail and the rolling friction coefficient between the driven wheel tread and the rail, and $R_{WX}$ is the maximum static friction force in the rail direction. When the absolute value of $R_{WX}$ is smaller than the value of $R_{WX}$, the static friction force acts on the wheel node. Moreover, when the absolute value of $R_{WX}$ is larger than the value of $R_{WX}$, the dynamic friction force by slipping acts on the driving wheel node and the friction force by rolling acts on the driven wheel node.

After crane runaway, when the runaway driving force becomes smaller than the resistance force preventing runaway, the crane decelerates and stops. After the crane stops, the friction force between the wheel tread and the rail was again calculated by Eq. (3). Entering the static state from the motion state was judged based on the following equations (4) and (5):

$$\dot{d}_{WX}(t) < \dot{d}_{WX}(t)$$

$$\dot{d}_{WX}(t-\Delta t) \dot{d}_{WX}(t) < 0$$

where $\dot{d}_{WX}(t)$ is the velocity of the motion limit at the wheel node in the rail direction for switching to the static state from the motion state at the present time, $t$, $\dot{d}_{WX}(t-\Delta t)$ is the velocity of the wheel node at the time before one unit time step, $t-\Delta t$, in the dynamic simulation analysis. If $\dot{d}_{WX}(t)$ at the present time, $t$, is smaller than $\dot{d}_{WX}(t)$ and the direction of $\dot{d}_{WX}(t)$ is not the same as the direction of the velocity $\dot{d}_{WX}(t-\Delta t)$ at the time before one unit time step, $t-\Delta t$, it was assumed that the wheel node sticks to the rail. The change of the wheel nodes from the motion state to the static state described in the next section was also considered using the same method.

### 3.3.4 Lateral direction

A static friction force and a dynamic friction force in the lateral direction of the rail are generated between the tread of the wheel and the top of the rail by the body behavior of the crane. These friction forces are treated using the method described in Section 3.3.3. If the displacement of the wheel node in the lateral direction is larger than the distance between the wheel flange and the side face of the rail in the first step of the simulation analysis in which the center of the wheel flange coincides with the center of the rail in the width direction, the wheel flange is assumed to be in contact with the side face of the rail and the contact reaction force is calculated. If the wheel flange contacts the side face of the rail, the friction force in the rail direction at the contact surface, which acts on the wheel nodes, is calculated as the resistance force.

### 3.4 Rail clamp braking force model

Cranes have rail clamps to prevent runaway caused by wind while in the cargo-handling standby state. The rail clamp is operated by hydraulic pressure, and clamp pads clamp both sides of the rail. The rail clamps are installed in the center of the sill beams of the land-side and the sea-side, as shown in Fig. 1. In the state in which the crane is resisting the wind, the static force in the rail direction is generated at the contact surface between the clamp pad and the side of the rail. In the state in which the crane is running away, the dynamic force in the rail direction is generated at the

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Fig.3 Contact interaction model between wheel and rail in the vertical direction

Fig.4 Contact interaction model between wheel and rail in the rail direction
contact surface. Figure 5 shows the rail clamp braking force model. In the model, \( R_{RF} \) is the friction force in the rail direction at the contact surface between the clamp pad and the rail. This force was treated using the method described in Section 3.3.3. The force acted on the node assumed to be the contact surface between the clamp pad and the side face of the rail (hereinafter referred to as the rail clamp node).

### 3.5 Wind environment model

In the dynamic simulation analysis of the present paper, the horizontal wind, having a vertical wind velocity distribution that follows a power law, was assumed to blow in the rail direction, as shown in Fig. 6. The wind velocity at an arbitrary height, \( V_{W}(h_{W}) \), was calculated using the following equation:

\[
V_{W}(h_{W}) = V_{W}(h_{0}) \left( \frac{h_{W}}{h_{0}} \right)^{m}
\]

where \( m \) is the power law index corresponding to the roughness of the surface, and \( h_{0} \) is the standard height. For a given height, the wind velocity was assumed to be uniform in the lateral direction of the rail.

### 3.6 Wind load model

The wind load acting on the crane was calculated using the following procedure based on the international standard that defines the method of wind load calculation (ISO4302, 1981). The definitions of the symbols in the equation are shown in Fig. 6.

1. The wind velocity at the height of each beam element of the crane model, \( V_{W}(h_{W}) \), was calculated using Eq. (6).
   - When the longitudinal direction of the element is horizontal, the wind velocity at the height of the node used to construct the element was calculated. When the longitudinal direction of the element is tilted with respect to the horizontal plane, the wind velocity at the average height of each region of the element that is equally divided in the height direction was calculated.

2. The wind pressure, \( p \), at the height of the node or the average height of each divided region was calculated using the following equation:

\[
p = \frac{1}{2} \rho \left( V_{W}(h_{W}) \right)^{2}
\]

3. The wind load, \( F_{W} \), that acts on the element or the divided region was calculated using the following equation, as specified by the international standard (ISO4302, 1981):

\[
F_{W} = ApC
\]

where the effective frontal area, \( A \), is the area of the crane surface exposed to the wind projected onto a surface perpendicular to the wind direction. If two or more surfaces overlap each other, the effective frontal area was multiplied by the reduction ratio for effective frontal area specified by the international standard. For the wind force coefficient, \( C \), for each element, the value specified by the same standard is used. When the longitudinal direction of the element is tilted with respect to the horizontal plane, the total wind load acting on each divided region was calculated.

4. The wind loads distributed on the element were converted into equivalent nodal loads on both end nodes of the element, and these loads acted on both end nodes.
4. Fundamental reliability of the dynamic simulation analysis method

4.1 General

The fundamental reliability of the dynamic simulation analysis conducted in the present paper was confirmed. One of the present authors has shown that the dynamic behavior of a tire-type crane receiving wind load can be analyzed when the constraint conditions in the contact part of the traveling units of the crane are modeled appropriately (Takehara et al., 1999a, 1999b, 1999c, 2000). In this section, we focused on the crane approaching runaway and after runaway and confirmed the fundamental reliability of the proposed method of dynamic simulation analysis from the standpoint of the motion characteristics of the crane by comparing the simulation results with the results of an analysis of picture image data obtained when an actual crane runaway caused by wind.

4.2 Analysis conditions

4.2.1 General

Table 1 shows the fundamental analysis conditions. In the present paper, a crane of standard size having a load rating in the 40-ton class that is widely used in Japan was chosen for the simulation analysis. The crane has 16 driving wheels and 16 driven wheels, and the arrangement of these wheels is shown in Fig. 2(b).

We assumed that a strong wind started to act on the crane based on the runaway of an actual crane caused by wind. The crane was assumed to be in the cargo-handling standby state and the trolley was in the rest position, as described in Section 2. In addition, the rail clamps and electromagnetic disk brakes in the DC motor were operated in order to maintain the cargo-handling standby state of the crane. Since the friction force generated between the clamp pads of the rail clamps and the rails, and that generated between the wheels and the rails constitute the majority of the resistance force preventing runaway, these friction coefficients have to be determined appropriately, as described in the following section.

4.2.2 Determination of the friction coefficient

Under the condition in which the crane is likely to experience runaway caused by wind, the rails were assumed to be submerged due to rainwater collected in the ditch in which the rails were laid, and the friction coefficients between the clamp pads and the rails and between the wheels and the rails were assumed to have decreased due to deterioration of the components with age. Although the materials of the wheels and the rails of the crane differ from those used in a
study on the abrasion of mild steel in pure water by Doi et al. (1975), the friction coefficient between stainless steel and chromed mild steel in pure water was approximately half that under the dry condition. Therefore, the static friction coefficients between the clamp pads and the rails and between the driving wheels and the rails were set to be approximately half the static friction coefficients under the dry condition of JCAS, as defined in the guidelines for the device for preventing crane runaway (JCAS 1201, 2009b). Based on the relationship between static friction coefficients and dynamic friction coefficients of metal materials (Hashimoto, 2006), the dynamic friction coefficients between the clamp pads and the rails and between the wheels and the rails were set to be approximately 0.7 times the static friction coefficients.

4.2.3 Setting of wind velocity and direction

Figure 7 shows the time change of the wind velocity \( V_w(h_{0h}) \) in the dynamic simulation analysis. Based on the wind velocity and wind direction when the actual crane ran away, the time change of the wind velocity after time \( t = 0 \) s was defined based on the 10-minute average wind velocity and moment wind velocity at the anemometer in the container terminal read at 1-minute intervals. The anemometer was installed at a height of \( h_{0h} = 22.9 \) m from the ground, which is the height of the roof of a building near the actual crane that ran away. Therefore, we used the wind velocity at \( h_{0h} = 22.9 \) m in this analysis. In addition, the wind direction was assumed to be constant and in the rail direction.

4.3 Motion characteristics of the crane

Figure 8 shows time history of the rail direction (X direction) velocity of the crane with wind velocity \( V_w(h_{0h}) \). In the figure, \( V_{RCXL} \) and \( V_{RCXS} \) are the velocities of the land-side and sea-side rail clamps, respectively, as shown in Fig. 2(a). However, the solid line indicating \( V_{RCXL} \) and the dashed line indicating \( V_{RCXS} \) in Fig. 8 are in agreement. Time \( t_R \) is the time at which the crane starts to run away and was determined based on the slippage of the land-side and sea-side rail clamps and all driving wheels according to the results of the dynamic simulation analysis. The white circles are the velocity in the rail direction that was obtained by image analysis of an actual runaway crane. The velocity obtained by picture image analysis is that at the joint part of the horizontal brace and the leg part on the windward land-side. The results for the actual crane were plotted so that the time at which the crane starts to run away is in agreement with time \( t_R \).

The acceleration and the velocity of the actual crane were approximately equal to those obtained by the dynamic simulation analysis until approximately \( t = 85 \) s, as shown in Fig. 8. For the actual crane, the acceleration increased somewhat after \( t = 85 \) s. The reason for this increase is considered to be that the braking forces decreased after failure of the braking device. A comparison of the results of the dynamic simulation analysis and the results of the image analysis for the actual crane reveals that the motion characteristics when the crane runs away as a result of horizontal wind in the rail direction can be determined using the proposed dynamic simulation analysis method.

5. Runaway characteristics of the crane caused by wind gust

5.1 General

In actual container terminals, unsteady wind with a rapid velocity increase, as in the case of a gust, may blow. The possibility that the crane will run away as the result of a wind gust is considered. In this section, we modeled the time change of wind velocity of the wind gust based on the increase ratio and the time change rate of the wind velocity and conducted the dynamic simulation analysis under the assumptions that the crane was in the cargo-handling standby

![Fig. 7 Time change of the wind velocity reference from the runaway of an actual crane](image)

![Fig. 8 Time history of the runaway velocity of the crane obtained through dynamic simulation analysis and image analysis](image)
state upon receiving a wind gust. Based on the analysis results, the runaway characteristics of the crane caused by the wind gust were quantitatively clarified. In addition, the influence of the time change rate of the wind velocity on the runaway characteristics was also discussed.

5.2 Analysis conditions
We assumed that the gust starts to blow against the crane, as shown in Table 1, in the cargo-handling standby state. The values of the friction coefficient between the clamp pads and the rails and between the wheels and the rails were the same as those listed in Table 1.

Figure 9 shows the time change model of wind velocity. Here, $V_{WZ}(h_0)$ and $V_{WH}(h_0)$ are the standard wind velocity at standard height, $h_0$, and the maximum wind velocity at that height, respectively. Moreover, $t = 0$ s is the time when the wind velocity starts to increase. In addition, $\Delta t_{mh}$ is the holding time of the maximum wind velocity, and $\Delta t_g$ is the length of time from the initiation of the wind velocity increase until the end of the wind velocity decrease. The wind conditions are shown in Table 2. For the increase ratio of the wind velocity, $V_{WH}(h_0)/V_{WZ}(h_0)$, and the time change rate, $dV_{W}(h_0)/dt$, we referred to an actual measurement study on wind gusts (Tomokiyo et al., 2010). In the study by Tomokiyo, the increase ratio in the wind velocity was from approximately 1.8 to 6.5 and the time change rate was from approximately 3.8 to 8.4 m/s². Under the condition of a large increase ratio, e.g., exceeding 2.0, in actual container terminals, cranes travel to the anchoring positions and proper measures for preventing runaway are taken. Therefore, the crane was assumed to have received a wind gust for which the increase ratio of the wind velocity is not large but the time change rate is large. Moreover, in the above-described study by Tomokiyo, the time for the wind velocity to be the maximum is approximately 1 second, and the wind direction changes very slightly. For these reasons, $\Delta t_{mh}$ is 1 second, and the wind direction does not change with time in the present study.

5.3 Characteristics of dynamic behavior of a crane approaching runaway and after runaway caused by wind gust

5.3.1 Total wind load and velocity of the crane in the rail direction
Figures 10(a) and 10(b) show the time history of the total wind load, $F_{TW}$, acting on the crane under the conditions of Table 2. In the figures, $R_{TSC}$ and $R_{TDC}$ are the total resistance forces of the runaway of the crane in the static state and in the runaway state, respectively, as obtained by simple calculation. Here, $R_{TSC}$ is the total maximum value of the static friction force, which is simply calculated in terms of the static friction coefficient, as shown in Table 1, and the pushing force of the clamp pads or the average wheel load obtained by dividing the crane weight by the number of wheels. Moreover, $R_{TDC}$ is the total friction force, which is simply calculated in terms of the dynamic or rolling friction coefficients, as shown in Table 1, and the pushing force or the average wheel load.

Table 2 Wind conditions

<table>
<thead>
<tr>
<th>Case No.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard height, $h_0$ [m]</td>
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<td>15*</td>
<td>15*</td>
<td>15*</td>
<td>15*</td>
<td>15*</td>
</tr>
<tr>
<td>Standard wind velocity, $V_{WZ}(h_0)$ [m/s]</td>
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<td>16.0*</td>
<td>16.0*</td>
<td>16.0*</td>
<td>16.0*</td>
<td>16.0*</td>
</tr>
<tr>
<td>Increase ratio of wind velocity, $V_{WH}(h_0)/V_{WZ}(h_0)$ [-]</td>
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<td>1.8</td>
<td>1.8</td>
<td>1.8</td>
<td>1.8</td>
<td>1.8</td>
</tr>
<tr>
<td>Time change rate of wind velocity, $dV_{W}(h_0)/dt$ [m/s²]</td>
<td>1.0</td>
<td>5.0</td>
<td>10.0</td>
<td>1.0</td>
<td>5.0</td>
<td>10.0</td>
</tr>
<tr>
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<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
</tr>
<tr>
<td>Acting time of gust, $\Delta t_g$ [s]</td>
<td>26.6</td>
<td>6.1</td>
<td>3.6</td>
<td>20.2</td>
<td>4.8</td>
<td>2.9</td>
</tr>
<tr>
<td>Ratio of the holding time of maximum wind velocity to the acting time of gust, $\Delta t_{mh}/\Delta t_g$ [-]</td>
<td>0.04</td>
<td>0.16</td>
<td>0.28</td>
<td>0.05</td>
<td>0.21</td>
<td>0.34</td>
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<tr>
<td>Power law index, $m$ [-]</td>
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<td>0.125</td>
<td>0.125</td>
<td>0.125</td>
<td>0.125</td>
<td>0.125</td>
</tr>
<tr>
<td>Wind direction</td>
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<td>Horizontal in the direction of rails</td>
<td>Horizontal in the direction of rails</td>
<td>Horizontal in the direction of rails</td>
<td>Horizontal in the direction of rails</td>
<td>Horizontal in the direction of rails</td>
</tr>
</tbody>
</table>

* Used height and wind velocity when calculating the wind load acts on the structural parts of the crane in the cargo-handling standby state (Japan Crane Association, 1997).
Figures 11(a) and 11(b) show the time history of the velocity of the crane in the rail direction under the conditions of Table 2. As shown in Fig. 8, the velocities in the rail direction of the land-side and sea-side rail clamp, \( V_{RCL} \) and \( V_{RCS} \), shown in Fig. 2(a) are shown. Although the details are described in Section 5.3.2, the time \( t_s \) is the time at which the land-side rail clamp and all of the land-side driving wheels enter the slip state, and time \( t_l \) is the time at which the sea-side rail clamp and all of the sea-side driving wheels enter the slip state. These times were judged based on Eq. (3).

In all cases listed in Table 2, when the wind velocity is maximum, \( F_{W1} \) is smaller than \( R_{TDC} \), as shown in Figs. 10(a) and 10(b). However, as shown in Fig. 11(a), the entire crane started to run away in all cases shown in Table 2 at time \( t_R \) when the wind gust with \( V_{W}(h_0) = 1.8 \) occurred. As shown in Fig. 11(b), the entire crane started to run away under the condition of \( dV_{W}(h_0)/dt = 10.0 \text{ m/s}^2 \) when the gust with \( V_{W}(h_0) = 1.6 \) occurred, although \( F_{W1} \) is smaller than \( R_{TDC} \). The reasons for these results are considered in Sections 5.3.2 and 5.3.3.

### 5.3.2 Dynamic behavior of the wind-crane-rail system

Figure 12 shows the dynamic behavior of the wind-crane-rail system under the condition of case 2 in Table 2. The red solid line shows the state of the crane without the deformation. The black solid line shows the state of the crane without the deformation. The reason for these results are considered in Sections 5.3.2 and 5.3.3.
As shown in Fig. 12(a), when the crane was struck by a wind having $V_{W}(h_0) = 16$ m/s, the part above the traveling units (hereinafter referred to as the upper part of the crane), mainly the legs, the brace beams, the booms, and the girders, deformed in the rail direction with the swinging that is clarified by the acceleration and the velocity shown in the middle and lower parts of Figs. 13 in Section 5.3.3. At this time, the rail clamps and the driving wheels of land-side and sea-side were adhering to the rails. When the wind velocity increased from $V_{W}(h_0) = 16$ m/s, the above-mentioned parts deformed more with swing in the rail direction as shown in Fig. 12(b). The rail clamp and the driving wheels of sea-side entered slip state in the rail direction. The rail clamp and the driving wheels of land-side then entered the slip state in the rail direction, as shown in Fig. 12(c). At this time, the entire crane started to run away. The rail clamp and the driving wheels of the sea-side entered the slip state earlier than those of the land-side. This is because the moment around the vertical axis passing through the T.C.G. caused by wind load acts in the clockwise direction, as viewed from the top of the crane. If the rail clamp and the driving wheels of the sea-side enter the slip state, the friction force between the rail clamp and the rails and that between the driving wheels and the rails of the sea-side becomes a dynamic friction force. Therefore, the crane starts to run away, although, based on a simple calculation, the total wind load, $F_{TW}$, that the crane receives is smaller than the total resistance force in the static state, $R_{TSC}$, as described in Section 5.3.1.

After the crane started to run away, the windward wheels of the sea-side and land-side entered the slip state in the lateral direction of the rail (+Y direction), as shown in Fig. 12(d). The reason for this is that, in addition to the above-mentioned moment, the wheel load of the wheels on the windward side decreased and the maximum static friction force in the lateral direction of the rail between the wheels and the rails decreased. In the wheels that entered the slip state in the lateral direction, the most windward wheel flange of the land-side slipped until it contacted the side of the rail. When the gust started to calm down and the wind velocity decreased, the sea-side rail clamp reentered the stick state, as shown in Fig. 12(e). After that, the land-side rail clamp reentered the stick state and the entire crane stopped, as shown in Fig. 12(f). Next, for example, in case 2 of Table 2, the slipping distances in the rail direction of the rail clamp of the land-side and sea-side are approximately 0.42 m and 0.48 m, respectively. Since the slipping distances of the rail clamps or the wheels associated with the gust are related to the abrasion characteristics between the clamp pads and the rails and between the wheels and the rails, the dynamic simulation analysis presented in the present paper is useful for quantitatively clarifying these characteristics.

### 5.3.3 Influence of the time change rate of the wind velocity on the runaway characteristics of the crane

The inertial force generated by swinging in the runaway direction of the crane caused by the gust is considered to promote crane runaway. In this section, we present the relation between the characteristics of the swinging crane and the runaway driving force, which consists of the wind load and the inertial force, and discuss the influence of the time change rate of the wind velocity of the gust on the runaway characteristics of the crane.

The upper part of Figs. 13(a) through 13(d) shows the time histories of $F_{TW}$, $F_i$, $F_{RD}$, and $R_{TDA}$. Here, $F_i$ is the inertial force generated on the crane, which was simply calculated by applying the average accelerations, $A_{TX}$ and $A_{MX}$, of the nodes, as described below, to the mass of the crane above the T.C.G. Moreover, $F_{RD}$ is the runaway driving force, which is the sum of $F_{TW}$ and $F_i$, and $R_{TDA}$ is the total resistance force preventing crane runaway obtained by the dynamic simulation analysis that is defined by following equation:

$$R_{TDA} = \sum R_{RXf} + \sum R_{WXf}$$  \hspace{1cm} (9)

where $R_{RXf}$ is the friction force of the rail clamp of the land-side and sea-side, as described in Section 3.4, $R_{WXf}$ is the friction force of each wheel, as calculated by Eq. (3) in Section 3.3.3, and $R_{TSC}$ and $R_{TDC}$, which are shown by the dashed lines in the upper part of Figs. 13(a) through 13(d), are the total resistance force preventing crane runaway in the static state and the runaway state by simple calculation, the values of which are same as in Fig. 10. In order to compare the magnitude of $R_{TDA}$ with that of $F_{RD}$, the sign of $R_{TDA}$ is inverted in the upper part of Figs. 13(a) through 13(d).

In the middle and lower parts of Figs. 13(a) through 13(d), the accelerations, $A_{TX}$ and $A_{MX}$, and the velocities, $V_{TX}$ and $V_{MX}$, in the rail direction at the node of the upper part of the crane are shown in order to clarify the characteristics of the swinging of the crane. Here, $A_{TX}$ and $V_{TX}$ are the values at the node at the top of the crane, indicated by $A$ in Fig. 2(a). Values for this node are shown because the acceleration and the velocity are easy to increase at the top of the crane.
(a) $t = 0.0$ s (wind velocity starts to increase)

(b) $t = 2.10$ s (sea-side clamp pad and driving wheels slip in the rail direction)

(c) $t = 2.75$ s (land-side clamp pad and driving wheels slip in the rail direction)

(d) $t = 3.20$ s (windward wheels slip in the lateral direction of the rail)

(e) $t = 5.20$ s (sea-side clamp pad sticks again)

(f) $t = 5.40$ s (land-side clamp pad sticks again)

Fig. 12 Dynamic behavior of the wind-crane-rail system (Case 2)
crane. Moreover, $A_{MX}$ and $V_{MX}$ are the acceleration and the velocity in the rail direction at the node that is equivalent to the machinery housing, indicated by B in Fig. 2(a). The inertial force of the machinery housing increases easily because the machinery housing is installed at the top of the crane, where the acceleration and the velocity increase easily and the weight is large. For this reason, we also focused on the machinery housing. In order to consider only the characteristics of the swinging at the top part of the crane, $V_{TX}$ and $V_{MX}$ are given as relative velocities, $V_{RCSX}$ and $V_{RCXL}$, with respect to the velocities of the rail clamps of the land-side and the sea-side, as shown in the following equations:

$$V_{TX} = V_{TX} - V_{RCSX}$$
$$V_{MX} = V_{MX} - V_{RCXL}$$

(a) Case of $\frac{V_{WH}(h_0)}{V_{WL}(h_0)} = 1.8$

As shown in Fig. 13(a), when a gust with $\frac{dV_{W}(h_0)}{dt} = 1.0 \text{ m/s}^2$ occurred, $A_{TX}$ and $A_{MX}$ fluctuated slightly from $t = 0$

Fig. 13 Time history of the total wind load, the inertial force, the runaway driving force, the total resistance force, $X$ direction acceleration and velocity of the crane
s at which the wind velocity and the wind load began to increase until $t_s$ at which the rail clamp and the driving wheels of the sea-side entered the slip state. As shown in Fig. 13(c), when a gust with $dV_W(h_0)/dt = 10.0 \text{ m/s}^2$ occurred, $A_{TX}$ and $A_{MX}$ fluctuated from $t = 0$ s until $t_S$. The period of time corresponds to a half portion of the fluctuation cycle. The fluctuation amplitude of the latter large $dV_W(h_0)/dt$ case is larger than that of former small $dV_W(h_0)/dt$ case. In both cases shown in Figs. 13(a) and 13(c), the cycle of the fluctuations was approximately 2 seconds. This cycle corresponds to the natural vibration cycle of the deformation mode in the rail direction of the crane as determined by the modal analysis. The runaway driving force, $F_{RD}$, fluctuated with the inertial force generated by the swinging of the upper part of the crane. For the cases shown in Figs. 13(a) and 13(c), $F_i$ was approximately 0 at time $t_L (= t_H)$ when the crane started to run away. Under these conditions, the crane is thought to have started to run away as a result of the increasing runaway driving force caused by the increasing wind load.

(b) Case of $V_{WH}(h_0)/V_{WH}(h_0)=1.6$

As shown in Fig. 13(b), when a gust with $dV_W(h_0)/dt = 1.0 \text{ m/s}^2$ occurred, $A_{TX}$ and $A_{MX}$ fluctuated slightly the time $t = 0$ s until $t_S$, as in Fig. 13(a). Moreover, $F_i$ was generated and fluctuated corresponding to the fluctuation of these accelerations. The rail clamp and the driving wheels of the sea-side entered the slip state at time $t_S$, whereas those of the land-side did not. Therefore, the entire crane did not start to run away. Here, as shown in the figure, $F_{RD}$ did not exceed the total resistance force preventing crane runaway, $R_{TD}$, that was obtained from the dynamic simulation analysis.

As shown in Fig. 13(d), when a gust with $dV_W(h_0)/dt = 10.0 \text{ m/s}^2$ occurred, $A_{TX}$ and $A_{MX}$ fluctuated and also the values of $A_{TX}$ and $A_{MX}$ became negative in approximately 1 second. This time length corresponds to the natural vibration cycle (approximately 2 seconds) of the deformation in the rail direction of the upper part of the crane. The value of $F_i$ became positive and large which could not be neglected. Therefore, $F_{RD}$ was larger than $R_{TD}$ during time $t_L$ from time $t_S$ and the crane started to run away.

If a wind gust has a large acceleration, such as $10.0 \text{ m/s}^2$, even if the increase ratio is small, the fluctuation of the inertial force generated by the swinging of the upper part of the crane increases. Therefore, the probability of crane runaway increases when the inertial force increases in the runaway direction, and the influence of the inertial force on crane runaway becomes impossible to ignore.

6. Conclusion

We herein proposed a method of dynamic simulation analysis in order to clarify the dynamic behavior of a crane that is widely used in container terminals while approaching runaway and after runaway caused by a wind gust. In the present study, we assumed that the wind velocity in the rail direction increases rapidly over a short time period, as in the case of a wind gust, and we assumed that the crane was in the cargo-handling standby state during rainy weather, so that the friction coefficient between the pads of the rail clamps and the rails and that between the wheels and the rails decreased. The primary conclusions of the present study are as follows.

1. The clamp pad and the driving wheels of the sea-side entered the slip state, and after that the clamp pad and the driving wheels of the land-side then entered the slip state and the crane started to run away, because the moment generated by the wind load acted around the vertical axis, which passes through the center of gravity of the crane.
2. If there is a time lag between the time at which the clamp pad and the driving wheels of the sea-side enter the slip state and the time at which those of the land-side enter the slip state, then even when the total wind load that the crane receives is smaller than the total resistance force preventing runaway (as calculated simply as the sum of the total maximum static friction force that is generated between the clamp pads and the rails and that between the wheels and the rails), the crane may start to run away.
3. The part above the traveling units of the crane, which is primarily composed of the legs, the brace beams, the booms, and the girders, deformed in the rail direction with during swinging caused by a wind gust in the rail direction.
4. Even if the increase ratio, $V_{WH}(h_0)/V_{WH}(h_0)$, is small, when a gust has a large time change rate, $dV_W(h_0)/dt$, and the fluctuation of the inertial force in the rail direction due to the swinging of the upper part of the crane described in (3) becomes large, the probability of crane runaway increases as the inertial force in the runaway direction increases.
In the present paper, the friction coefficients between the clamp pads and the rails and between the wheels and the rails that become the resistance force preventing the runaway were constant. In the future, we are going to investigate the influence of the variation of these friction coefficients on the dynamic behavior of the crane while approaching runaway and after runaway. Moreover, we intend to propose guidelines for wind-resistant design and operational management of cranes in order to prevent runaway crane accidents and contribute to safety.

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