Simulation of Unstable Strip Running on Hot Run-Out-Table

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In the hot-rolling process, defects resulting from running instability when a thin strip runs on the Run-Out-Table (ROT) at high speed in the state of non-tension (head or tail of the strip) pose a chronic problem. As a measure to prevent unstable strip running, the maximum line speed has been limited. Consequently, this has also become an aggravating factor that impedes higher productivity. Although various methods to solve and improve this problem have been proposed from an early date, it has not been eradicated completely.

In this paper, we propose a numerical simulation method to simulate the strip running phenomenon on the ROT. It was found that the numerical results correspond to the theoretical and experimental results. Use of the proposed method has reduced both the cost and time necessary to improve the unstable strip running phenomenon.

KEY WORDS: hot strip rolling process; strip; run-out-table; ROT; unstable; buckling; multi-body dynamics; plastic bending; numerical simulation.

1. Introduction

In the hot strip rolling process, a running hot strip is rolled to the target thickness by the plural finishing mills. It is then cooled with water to the target temperature on a Run-Out-Table (ROT) having a length of about 100 m to 150 m and is finally coiled by the coiler. The ROT is a kind of conveyer with plural table rollers (TR), apron guides (AP) and side guides (SG). In conventional batch-process hot strip rolling, strips run at high speed on the ROT in a non-tensioned condition at the head and tail ends. The running velocity of comparatively thin strips is about 11 m/s. Various defects, termed folded defect in head (**Fig. 1**) and folded defect in middle (**Fig. 2**), are caused by threading instability¹⁾ in this process and have become a chronic problem. In addition, when the running strip moves in the lateral direction before passing the entrance of the SG, the head of the strip may collide with the entrance of the SG, causing the problem called trip trouble (trip on side guide). When a folded defect in the head end or middle of a strip is folded under by the coiler pinch roller, the affected part must be cut off and discarded in the next process, which reduces product yield. If threading becomes extremely unstable, the head end sometimes fails to reach the coiler, resulting in serious trouble and long downtime. To avoid these prob-

lems, on the operational side, the maximum threading speed is constrained to a stable speed based on past experience, but this impedes higher productivity. In addition, the heating temperature is set high to compensate for the temperature drop at the head end due to the threading speed restriction, and this is a problem from the viewpoint of energy unit consumption.

Fig. 1. Folded defect in head.

Fig. 2. Folded defect in middle.

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Various methods have been proposed to solve the problem of unstable strip running and improve the process, but it has not been eradicated completely. Verification and optimization of countermeasures for unstable threading phenomena are difficult, as these phenomena occur under high speed and unsteady conditions in actual equipment. Therefore, it has been expected that a practical numerical simulation technique makes it possible to simulate these phenomena on a computer.

Various types of plate conveying simulations have been studied in the past. A plate conveying dynamic simulation technique using a discrete elasticity beam model²⁾ has been proposed for conveying rollers and guides for 2D problems. Simulation time can be shortened with this model, as region integrals are not used. For the target phenomena of this paper, the strip model is an elasto-plastic beam with large deflection, the strip collides with plural TRs and AGs, and braking and driving force caused by friction and fluid-dynamics forces (air and water) act on the strip. For practical simulation of these complex phenomena, it is important to achieve robustness and efficiency in the simulation.

For 3D problems that consider lateral strip movement, the strip lateral walking simulation on conveying an endless belt to evaluate the effect of conveying roller alignments by normal FEM and a large deflection elastic plate model have been proposed.³⁾ The large deflection discrete strip model has been proposed to improve the robustness of calculations and shorten calculation time.⁴⁾ This model is an approximation of the large deflection plate model and is an extension of the discrete beam model. Practical modeling for numerical simulation and simulation of strip running on the ROT have not been described by the discrete strip model, while a formal representation of the discrete strip model has been proposed.

In this paper, we propose using a numerical simulation method based on multi-body dynamics (MBD) to simulate the strip running phenomena on the ROT. First, the modeling for simulation of 2D strip running on the ROT by using the discrete elasto-plasticity beam model is described. Next, the modeling for simulation of 3D strip running on the ROT by using the large deflection discrete strip model extended to the lateral direction is described. The proposed 3D strip running simulation is applied to the problem of trip trouble at the entrance of the SG. Finally, the simulation results are compared with the theoretical and experimental results, clarifying the fact that the proposed simulation method is appropriate for simulation of strip running on the ROT.

2. Modeling of 2D Strip Running Simulation on ROT

MBD is used to calculate the simulation of strip running on the ROT and is compatible with the discrete beam model. MBD is a kind of numerical simulation method to solve a dynamic system composed of plural rigid bodies, constraint elements among those bodies, force elements acting on those bodies and abstract elements, which are the differential equation system and controllers. ADAMS is used as the simulation platform. It is provided by MSC Corporation and is a kind of general mechanism analysis software.

2.1. Discrete Beam Model of Strip

The discrete beam model of a strip is shown in Fig. 3. The strip deflection is represented by plural rigid bodies and hinge constraint elements between those bodies. The strip bending moments are represented by torsion spring elements, which are set on the hinge constraint elements. Considering the torsion spring elements as a mechanical system, the system input is the angle between the bodies, and the system output is the bending torque. In the case of an elastic beam, the bending torque M is defined as

$$M = \frac{EI}{\Delta L} \Delta \theta \quad \dots \dots \dots (1)$$

where, E is Young's modulus, I is the second moment of area, ΔL is the length of a rigid body and $\Delta \theta$ is the angle between rigid bodies.

Large deflection of the strip in the vertical direction provokes local plastic bending deformation when the running strip behavior becomes unstable. The hot strip is considered to be an elasto-perfect plastic beam. Considering a static case, if the angle $\Delta \theta$ is smaller than the angle $\Delta \theta_Y$, which is the elastic limit angle between bodies, the bending torque is represented by Eq. (1). On the other hand, if the angle $\Delta \theta$ is greater than the angle $\Delta \theta_Y$, the bending torque is defined as

$$M = \frac{EI \Delta \theta_Y}{\Delta L} \quad \dots \dots \dots (2)$$

The angle $\Delta \theta_Y$ is defined as

$$\Delta \theta_Y = \frac{3\sigma_Y \Delta L}{Eh} \quad \dots \dots \dots (3)$$

where, σ_Y is the yield stress of the strip and h is the thickness of the strip. Considering the hysteresis effect for a dynamic case, the bending torque is represented by

$$M = \frac{EI}{\Delta L} (\Delta \theta - \Delta \theta_P) + C \frac{d}{dt} (\Delta \theta - \Delta \theta_P) \quad \dots \dots \dots (4)$$

$$\frac{d}{dt} \Delta \theta_P = \frac{EI}{C_P \Delta L} H(|\Delta \theta - \Delta \theta_P| - \Delta \theta_Y) (\Delta \theta - \Delta \theta_P) \quad \dots \dots (5)$$

where, C is the viscosity damping coefficient, $\Delta \theta_P$ is the plastic angle between bodies, C_P is the viscous-plasticity damping coefficient and H is the Heaviside function. Qualitatively, the robustness of the calculation increases as C becomes larger, and strip bending becomes more brittle as C_P becomes smaller. The Heaviside function H , which is a discontinuous function, makes the calculation become

Fig. 3. Discrete beam model.

unstable. Thereby, an approximated Heaviside function with a smoothed step edge is used to stabilize the calculation. The bending model using elasto-perfect plastic law of Eqs. (4) and (5) provides a shorter calculation time and better robustness than that with hardening law.⁵⁾

Elastic extension in the longitudinal direction can be appended to the discrete beam model by using spring elements between the bodies. However, irregular collisions between the non-tensioned running strip and the TRs cause high frequency vibration in the longitudinal direction with introduction of the elastic extension. Time increment must be enough small so as to prevent the high frequency vibration from making the calculation unstable. Therefore, the elastic extension is not important and should not be used in a simulation of a threading strip running on the ROT.

2.2. Algorithm of High Speed Collision Detections and Model of Collision Force

Next, collision between a strip and the bodies of the TR and AG is considered. ADAMS has a function of the collision element among rigid bodies. When a discrete strip has n rigid bodies and the bodies of the TR and AG have m rigid bodies, the number of collision elements is $n \times m$. Since the calculation time becomes longer and practical calculation becomes difficult as the number of bodies n and m increases, methods to reduce the calculation load of collisions will be considered.

As shown in Fig. 3, the collision points are set in the right lower and upper ends of a strip rigid body. The collision force between a strip and the upper side collision bodies with the strip acts at the upper collision point. The collision force between the strip and the lower side collision bodies with the strip acts at the lower collision point. But, collisions among strip rigid bodies are ignored. It is considered that the profiles of the collision body with a strip are only an arc (TR) and line segment (AG). **Figure 4** shows the positional relationship between a collision point on a strip rigid body and a collision body with the strip. The distance D between the collision point and the collision body is calculated by

$$\mathbf{R} = \mathbf{R}_0 + \mathbf{s} \cdot (\mathbf{P} - \mathbf{R}_0) \mathbf{s} \quad \text{..... (6)}$$

$$D = \sqrt{(\mathbf{P} - \mathbf{R}) \cdot (\mathbf{P} - \mathbf{R})} \quad \text{..... (7)}$$

where, $\mathbf{P}$ is the position vector based on the origin O of the collision point on a strip rigid body, $\mathbf{R}_0$ is the position vector based on the origin O of the collision body with the strip and $\mathbf{s}$ is the control vector of the profile of the collision body

Fig. 4. Distance from contact point to object.

with the strip. When the profile of a collision body with a strip forms an arc, the vector $\mathbf{R}_0$ is the center position vector of the arc and the vector $\mathbf{s}$ is the zero vector. When the profile of the collision body with a strip is a line segment, the vector $\mathbf{R}_0$ is the position vector on the reference line of the line segment and the vector $\mathbf{s}$ is the unit tangent vector of the segment line. The vector $\mathbf{R}$ is the nearest position vector on the profile of a collision body with a strip to the collision point $\mathbf{P}$. If the distance D is smaller than the distance R_C , a collision occurs between a strip and a collision body with the strip. The collision bodies TR and AG are set along the pass line so that the strip runs linearly on the ROT. As shown in **Fig. 5**, every divided region has its unique collision body. When the collision point $\mathbf{P}$ passes through the boundary and moves to position $\mathbf{P}'$, the collision body with the strip moves from position $\mathbf{R}_0$ to position $\mathbf{R}'_0$ along the trajectory without colliding with the collision point $\mathbf{P}$ at a much higher speed than the velocity of the collision point $\mathbf{P}$, forestalling collision point $\mathbf{P}$. Moreover, various characteristics such as the control vector, rotational velocity and coefficient of friction of the collision body on movement are changed continuously by using a spline function. These calculation processes make it possible to reduce the number of collision bodies to only one (Algorithm of High Speed Collision Detections).

As for the collision forces, a collision point is considered to collide against a collision body with infinite mass. The collision force F_{IMP} is given by

$$F_{IMP} = K_{IMP}(R_C - D) - C_{IMP} \frac{d}{dt} D \quad \text{..... (8)}$$

where, K_{IMP} is the collision elastic coefficient and C_{IMP} is the collision damping coefficient. Equation (8) becomes effective when $D \leq R_C$ is satisfied. The direction of the collision force is the normal direction to the profile of a collision body. The driving (or braking) force, which originates from a friction force represented by multiplication of a collision force and a friction coefficient, acts at a collision point in the tangential direction of the profile of the collision body.

2.3. Model of External Forces Acting on Strip

As for the external forces acting on a strip, excluding the collision friction forces with TR and AG, these forces include the force of gravity, aerodynamic forces and col-

Fig. 5. Algorithm of high speed collision detections.

lision friction forces with the cooling water. The external forces act at the center of gravity against the strip rigid body.

As for the aerodynamic force acting on a strip rigid body, the aerodynamic force vector $\mathbf{F}_{\text{AIR}}$, referring to the formula of Rayleigh,⁷⁾ is represented by

$$\mathbf{F}_{\text{AIR}} = F_{\text{AIR}} \mathbf{n}_{\text{CM}} \quad \dots\dots\dots (9)$$

$$\mathbf{n}_{\text{CM}} = (-\sin \theta, \cos \theta) \quad \dots\dots\dots (10)$$

$$F_{\text{AIR}} = \frac{1}{2} \rho_a b \Delta L \left(v^x |v^x| c_F^x + v^y |v^y| c_F^y \right) \quad \dots\dots\dots (11)$$

$$c_F^x = \frac{4C_L \sin \theta}{4 + \pi |\sin \theta|} \quad \dots\dots\dots (12)$$

$$c_F^y = \frac{-4C_L \cos \theta}{4 + \pi |\cos \theta|} \quad \dots\dots\dots (13)$$

where, $\mathbf{n}_{\text{CM}}$ is the unit normal vector of a strip rigid body, θ is the angle of attack of the strip rigid body, b is the strip width, ρ_a is the air density, v^x is the velocity along the x axis (longitudinal direction) of the strip rigid body, v^y is the velocity along the y axis (vertical direction) of the strip rigid body and C_L is the lift coefficient of the strip rigid body.

As the fluid-dynamic force of the cooling water, water collision forces act in the direction of water injection and water viscosity forces act in the direction of strip movement.

3. Modeling of 3D Strip Running Simulation on ROT with Large Deflection Discrete Strip Model

The large deflection discrete strip model⁸⁾ is applied to the strip model to solve the dynamic lateral movement problems, including camber problems, lateral walking, *etc.* that occur on the ROT. Both large deflection in the vertical direction and torsion cause large displacement in the lateral direction. SGs are usually installed in an actual ROT to constrain the lateral movement of strips.

As shown in **Fig. 6**, as the basic design, the large deflection discrete strip model comprises two discrete beams. The distance c between the discrete beam and the center line in the width direction is $\sqrt{3}b/6$. Considering the strip to be

elastic, the functional of the large deflection discrete strip model is defined as

$$\begin{aligned} \Pi = & \sum_{i=1}^n \left\{ \frac{1}{2} K_B (\Delta \theta_i^{(1)})^2 + T^{(1)} (1 - \cos \theta_i^{(1)}) \Delta L + \frac{1}{2} K_T (\Delta \omega_i^{(1)})^2 + \frac{1}{2} K_G (\Delta \varepsilon_i^{(1)})^2 \right\} + \\ & \sum_{i=1}^n \left\{ \frac{1}{2} K_B (\Delta \theta_i^{(2)})^2 + T^{(2)} (1 - \cos \theta_i^{(2)}) \Delta L + \frac{1}{2} K_T (\Delta \omega_i^{(2)})^2 + \frac{1}{2} K_G (\Delta \varepsilon_i^{(2)})^2 \right\} \end{aligned} \quad \dots\dots\dots (14)$$

and the geometric constraint conditions are given by

$$(\theta_i^{(1)} - \theta_i^{(2)}) \Delta L = \frac{\sqrt{3}b}{6} (\Delta \omega_i^{(1)} + \Delta \omega_i^{(2)}) \quad \dots\dots\dots (15)$$

$$\Delta \omega_i^{(1)} = \Delta \omega_i^{(2)} \quad \dots\dots\dots (16)$$

$$\Delta \varepsilon_i^{(1)} \Delta L = \frac{\sqrt{3}b}{6} (1 - \cos \Delta \omega_i^{(1)}) + \frac{\sqrt{3}b}{6} (1 - \cos \Delta \omega_{i+1}^{(1)}) \quad \dots\dots\dots (17)$$

$$\Delta \varepsilon_i^{(2)} \Delta L = \frac{\sqrt{3}b}{6} (1 - \cos \Delta \omega_i^{(2)}) + \frac{\sqrt{3}b}{6} (1 - \cos \Delta \omega_{i+1}^{(2)}) \quad \dots\dots\dots (18)$$

where, n is the number of rigid bodies of a discrete beam, $T^{(1)}$ and $T^{(2)}$ are the tensions acting in discrete beams 1 and 2, $\Delta \theta_i^{(1)}$ and $\Delta \theta_i^{(2)}$ are the deflection angles between beam rigid bodies i and $i-1$ for discrete beams 1 and 2, $\Delta \omega_i^{(1)}$ and $\Delta \omega_i^{(2)}$ are the torsion angles between beam rigid bodies i and $i-1$ for discrete beams 1 and 2, $\Delta \varepsilon_i^{(1)}$ and $\Delta \varepsilon_i^{(2)}$ are the angles around the vertical direction between beam rigid bodies i and $i-1$ for discrete beam 1 and 2, K_B is the bending stiffness of a discrete beam, K_T is the torsion stiffness of a discrete beam and K_G is the membrane stiffness. K_B , K_T and K_G are represented respectively by

$$K_B = \frac{EI}{2(1-\nu^2)\Delta L} \quad \dots\dots\dots (19)$$

$$K_T = \frac{GJ}{2\Delta L} \quad \dots\dots\dots (20)$$

$$K_G = \frac{Ehb^3}{120\Delta L} \quad \dots\dots\dots (21)$$

where, ν is the Poisson ratio of a strip, $G(=E/2(1+\nu))$ is the transverse stiffness of the strip and $J(=h^3b/3)$ is the Saint Venant's torsional constant. The first term of the right side of Eq. (14) represents the potential functional of discrete beam 1, and the second term represents the potential functional of discrete beam 2. In the discrete strip model, which is derived from the infinitesimal deflection plate model, the membrane stiffness K_G becomes zero. However, if one assumes K_G to be zero, the strip torsion becomes much larger than the reality and thus fails to reproduce the actual strip phenomenon. Therefore, the proper K_G should be defined for a precise simulation. The membrane stiffness K_G , which represents the effect of geometric nonlinearity, is very important for simulating a strip running on a ROT. **Figure 7** shows the mechanism diagram of the large deflection discrete strip model. It is considered that there are many

Fig. 6. Discrete strip model.

Fig. 7. Mechanism diagram of discrete strip model.

mechanisms embodying the specifications shown in Fig. 6, such as normal mechanical designs. The mechanism in Fig. 7 is the result of numerous improvements to stabilize MBD calculations and is one practical mechanism with an extensive record of use, while it is not proven that the mechanism shown in Fig. 7 is the best design.

The equation of the torque vector used as the torque vector element in Fig. 7 is given by

$$\begin{Bmatrix} M_i^{T(1)} \\ M_i^{B(1)} \\ M_i^{G(1)} \end{Bmatrix} = \begin{Bmatrix} K_T \Delta \omega_i^{(1)} \\ K_B \Delta \theta_i^{(1)} \\ K_G \Delta \varepsilon_i^{(1)} \end{Bmatrix}, \begin{Bmatrix} M_i^{T(2)} \\ M_i^{B(2)} \\ M_i^{G(2)} \end{Bmatrix} = \begin{Bmatrix} K_T \Delta \omega_i^{(2)} \\ K_B \Delta \theta_i^{(2)} \\ K_G \Delta \varepsilon_i^{(2)} \end{Bmatrix} \dots (22)$$

where, $M_i^{T(1)}$ and $M_i^{T(2)}$ are the components of the torsion torque in discrete beams 1 and 2, $M_i^{B(1)}$ and $M_i^{B(2)}$ are the components of the bending torque in discrete beams 1 and 2 and $M_i^{G(1)}$ and $M_i^{G(2)}$ are the components of the torque around the out-plane direction in discrete beams 1 and 2. Equation (22) corresponds to Eq. (1).

A local coordinate system is defined on every beam rigid body. Let $C_i^{(1)}$ be the coordinate system on beam rigid body i in discrete beam 1. If discrete beam 1 is constrained in a global plane, the rotation axis of the beam rigid body referring to local coordinate system $C_i^{(1)}$ and the rotation axis of the beam rigid body referring to local coordinate system $C_{i+1}^{(1)}$ are the same and are equal to the normal axis of the plane. On the other hand, if discrete beam 1 is not constrained in a global plane, the rotation axis of the beam rigid body referring to local coordinate system $C_i^{(1)}$ is different from the rotation axis of the beam rigid body referring to local coordinate system $C_{i+1}^{(1)}$. Therefore, it is considered that the rotation axis of the beam rigid body refers to local

coordinate system $C_{i+1}^{(1)}$ in this paper.

If the angle between beam rigid bodies is very small, Eq. (22) is sufficient. In case the angle between beam rigid bodies is not small and finite due to large deflection caused by an unstable running strip on the ROT, Eq. (22) makes some calculations become unstable, as described later. Dividing a beam into a finite number of rigid bodies breaks the linearity of very small rotations. Considering that $\Delta \varepsilon_i^{(1)}$, $\Delta \theta_i^{(1)}$ and $\Delta \omega_i^{(1)}$ are respectively the yaw, pitch and roll angle between beam rigid bodies, the coordinate transformation from coordinate system $C_{i+1}^{(1)}$ to coordinate system $C_i^{(1)}$ is given by

$$C_i^{(1)} = A_i^{R(1)} A_i^{P(1)} A_i^{Y(1)} C_{i+1}^{(1)} \dots (23)$$

where, the rotation matrix $A_i^{Y(1)}$, $A_i^{P(1)}$ and $A_i^{R(1)}$ are respectively

$$A_i^{Y(1)} = \begin{bmatrix} \cos \Delta \varepsilon_i^{(1)} & -\sin \Delta \varepsilon_i^{(1)} & 0 \\ \sin \Delta \varepsilon_i^{(1)} & \cos \Delta \varepsilon_i^{(1)} & 0 \\ 0 & 0 & 1 \end{bmatrix} \dots (24)$$

$$A_i^{P(1)} = \begin{bmatrix} \cos \Delta \theta_i^{(1)} & 0 & \sin \Delta \theta_i^{(1)} \\ 0 & 1 & 0 \\ -\sin \Delta \theta_i^{(1)} & 0 & \cos \Delta \theta_i^{(1)} \end{bmatrix} \dots (25)$$

$$A_i^{R(1)} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos \Delta \omega_i^{(1)} & -\sin \Delta \omega_i^{(1)} \\ 0 & \sin \Delta \omega_i^{(1)} & \cos \Delta \omega_i^{(1)} \end{bmatrix} \dots (26)$$

Considering the order of rotational transforms, using the torque vector

$$\begin{Bmatrix} M_i^{T(1)} \\ M_i^{B(1)} \\ M_i^{G(1)} \end{Bmatrix} = A_i^{Y(1)} \begin{Bmatrix} 0 \\ 0 \\ K_G \Delta \varepsilon_i^{(1)} \end{Bmatrix} + A_i^{P(1)} A_i^{Y(1)} \begin{Bmatrix} 0 \\ K_B \Delta \theta_i^{(1)} \\ 0 \end{Bmatrix} + A_i^{R(1)} A_i^{P(1)} A_i^{Y(1)} \begin{Bmatrix} K_T \Delta \omega_i^{(1)} \\ 0 \\ 0 \end{Bmatrix} \quad \dots (27)$$

acting in beam rigid body i referring to local coordinate system $C_{i+1}^{(1)}$ instead of Eq. (22) enables stable calculation in the case of large deflections. In the same ways, the above formulation is applied to discrete beam 2.

The various modeling techniques used for the discrete beam model can be applied to the discrete strip model evolved from the discrete beam model. Applying elasto-plastic bending to the discrete strip model, elasto-plastic bending torque like that in Eqs. (6) and (7) is applied instead of using the elastic bending torque $K_B \Delta \theta_i^{(1)}$ of the second term on the right side of Eq. (27). Applying a camber shape to the discrete strip model, the difference between the length of the 1st beam rigid body and the length of the 2nd one is set, and the yaw angle $\varepsilon_i^{(1)}$ and $\varepsilon_i^{(2)}$ are corrected according to the camber. Some general MBD analysis programs enable coupled numerical analyses of the mechanisms and the elastic plates solved by FEM. However, it is difficult to customize the embedded elastic plate model, but on the other hand, it is relatively easy to customize the discrete strip model.

4. Verification of Strip Running Simulation on ROT

Figure 8 shows the run-out-table simulator (ROS).⁶⁾ ROS is an experimental device with a scale of one-tenth that of the actual ROT and was used to verify the proposed strip running simulation on the ROT by comparing the simulation results with the experimental results. It is difficult to verify the strip running simulation on the ROT by using an actual ROT due to the large number of unknown errors caused by the equipment and operation and the difficulty of tuning the detailed experimental conditions and measuring the running behaviors. Plastic strips are discharged toward the right from the pinch roller, shown on the left side of Fig. 8, which corresponds to the last finishing mill. The plastic strips then run on the table rollers. All table rollers are driven at almost the same rotational velocity. As for a driving mechanism, a rotating shaft under the table rollers drives part of the table rollers by way of gum belts, and the table rollers are also connected with each other by gum belts. Considering the similarity law,⁶⁾ the material of the plastic strips was nylon. Young's modulus of the plastic strips was measured by a

Fig. 8. Run-out-table simulator.

vibration method. The temperature of the plastic strips was carefully controlled so as to keep the Young's modulus of the strips at a constant value. The deflections of the plastic strips were measured by laser displacement sensors, which were set on the ceiling of the safety covers at specific intervals, in order to observe the strip running behavior.

Table 1 shows the specifications of the ROS and the simulation conditions. The bending of the nylon strips is not plastic, but rather is elastic. Therefore, the bending torque shown in Eq. (1) was applied to the simulation for the ROS.

The lift coefficient C_L was decided to be π considering the behaviors of the folded defects in head in the ROS and in the actual ROT. The lift coefficient for a finite length of plate from the Kutta-Joukowski theorem becomes 2π .⁸⁾ On the other hand, the lift coefficient for an infinite length of plate from Rayleigh's flow becomes $\pi/2$.⁷⁾ In the simulation of strip running on the ROT with the lift coefficient of 2π , the length of the folded defect in head was longer than the actual length, indicating that the lift coefficient of 2π is too large. On the other hand, in the simulation of strip running on the ROT with the lift coefficient of $\pi/2$, the lift coefficient of $\pi/2$ was too small for the folded defect in head to occur. These theoretical lift coefficient values were used as reference, although it is natural that the lift coefficient of the strip in the head end, which is neither in a finite plate nor in an infinite plate, is different from these theoretical lift coefficient values.

Figure 9 shows a snapshot of the simulated strip running on the ROS. **Figure 10** shows the simulation results. The horizontal axis in Fig. 10 shows the strip velocity, and the

Table 1. Simulation conditions.

Symbols	Specifications	Values
	Table Roller Diameter	ϕ 40 mm
	Table Roller Pitch	43 mm
ρ	Strip Density	1 261 kg/m ³
E	Strip Young's Modulus	3.3E+8 Pa
h	Strip Thickness	0.21 mm
b	Strip Width	70 mm
μ	Slip Friction Ratio	0.47
U_c	Strip Critical Velocity	2.27 m/s
C_L	Lift Factor	π
ΔL	Plate Element Length	10 mm

Fig. 9. Simulated strip running on ROS.

vertical axis shows the maximum loop height. When the strip velocity was within 2 m/s, loops did not occur and the strip ran on the ROS in a stable condition. However, when the strip velocity was greater than 2 m/s, loops occurred and strip running on the ROS became unstable. The longitudinal compression load $\rho A v^2$, where ρ is the density of a strip, A is the cross-sectional area of the strip and v is the strip velocity, is caused by the centrifugal force acting on the strip and is called the inertia compression force. When the compression load $\rho A v^2$ is greater than the buckling load $EI(\pi/L)^2$, where L is the length between the table rollers, the running strip on the ROT is in a buckling condition and becomes unstable.¹⁾ As the strip velocity increases, the inertia compression force $\rho A v^2$ and the maximum loop height also become larger. Figure 9 shows the loops that occurred in unstable strip running on the ROS with the strip velocity of 4 m/s.

The theoretical value of 2.27 m/s for the critical velocity¹⁾ in Table 1 was in good agreement with the simulation result. The maximum loop height also agreed with the experimental result.³⁾ Using the simulation of strip running on the ROT, it is possible to simulate the critical condition and the strip post-buckling behavior in the unstable condition.

The ability to calculate lateral buckling and post-buckling stably and with good precision is important for simulating

the unstable behavior in 3D strip running. The fact that the calculations by the proposed model were stable and had good precision was verified by calculating the lateral buckling problem shown in Fig. 11. The theoretical value²⁾ of the in-plane lateral buckling moment M_C is given by

$$M_C = \sqrt{\left(T + \frac{EI}{1-v^2} \left(\frac{2\pi}{L}\right)^2\right) \left(\frac{Tb^2}{12} + \frac{Eh^3b^3}{144(1-v^2)} \left(\frac{2\pi}{L}\right)^2 + GJ\right)} \quad (28)$$

where, L is the length of the strip and T is the tension acting on the strip. ABAQUS/STANDARD Version 6.5, which is a general FEM program, was used to solve the lateral buckling problem, shown in Fig. 11, of a large deflection plate. MSC.ADAMS 2007r1, which is a general MBD analysis program, was used to solve the lateral buckling problem with the large deflection discrete strip model. Table 2 shows the calculation conditions. Figure 12 shows the relationship between the lateral displacement at the point where the in-plane moment acted and the in-plane moment M . The dotted line and the solid line show the simulation results of the large deflection discrete strip model by the torque formulas in Eqs. (22) and (27), respectively. Figure 11 shows a snapshot of the discrete strip when the in-plane moment is 2 000 Nm. The group of white round points in Fig. 12 shows the calculation result of the large deflection plate model. In this case, calculation became impossible when the in-plane

Fig. 10. Simulation results.

Table 2. Lateral buckling simulation conditions.

Symbols	Specifications	Values
ν	Poisson's ratio	0.3
E	Young's Modulus	2.1×10^{11} N/m ²
h	Plate Thickness	0.0012 m
b	Plate Width	0.25 m
L	Plate Length	1.0 m
T	Tension	9 800 N
M_C	Lateral Buckling Moment	807.5 Nm

Fig. 11. Lateral buckling strip shape of discrete strip model.

Fig. 12. Results of lateral buckling simulation.

moment became greater than 1 500 Nm. The results of both the plate model and the discrete strip model were in good agreement within 1.5 times the lateral buckling moment. Since the application target of this simulation is to enable virtual experiments of strip running on the ROT in order to improve actual production lines, it is considered sufficient if the critical behaviors can be calculated efficiently. The behavior that lateral displacement vibrated irregularly when the torque formula in Eq. (22) was used occurred in lateral buckling, as shown in Fig. 12. This unstable behavior increased the calculation time or made it necessary to stop the simulation when the unstable behavior was particularly severe. This unstable behavior is not caused by a physical phenomenon but by calculation instability. As shown in Fig. 12, the torque formula in Eq. (27) secured calculation stability. Using the torque formula in Eq. (27) enables numerical stability of the 3D strip running simulation on the ROT.

5. Conclusions

In this paper, we considered the numerical analysis of the problems termed folded defect in head, folded defect in middle and trip on the side guide, which are caused by strip

running instability when a thin strip runs on a ROT at high speed in the state of non-tension at the head or tail of the strip in the hot-rolling process. The following conclusions were drawn from the numerical investigations carried out in this work:

- (1) A strip running simulation technique for the ROT based on MBD was developed.
- (2) A proposed discrete beam model with elasto-plastic bending made it possible to stabilize the calculation.
- (3) An algorithm for detection of high speed collisions between the beam rigid body and the bodies of the TR and AG was proposed.
- (4) A proposed torque vector made it possible to stabilize the calculation for the large deflection discrete strip model based on MBD.
- (5) The critical stable threading velocity determined from the simulation results was in good agreement with the theoretical velocity.

(6) Based on a comparison of the simulation results with the experimental results of the ROS, it was confirmed that the proposed strip running simulation on the ROT enabled simulation of unstable threading behavior.

It is practically possible to apply the proposed simulation to the running stability assessment for cooling water and leveling of the TRs and AGs on the ROT and coiler.

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