1. Introduction

In the mass production of thick steel plate in the steel industry, a roller system is used to transfer the steel plate. During the transfer, the steel plate deflects due to the gravitational force, and this causes impact between the roller and plate. This impact causes a very loud noise—more than 120 dBA—and workers are exposed to this noise, with potential for degradation of their hearing ability over time. Generally, the noise-induced permanent threshold shift (NIPTS) happens when workers are exposed to noise level of more than 85 dBA. Thus, most workers use, for their convenience, foam-type earplugs, which show good performance especially for high frequencies over 1,000 Hz. This type of earplug can reduce the noise level by about 30–45 dBA, but its effectiveness varies with the wearing method. Although it is very effective against very loud noise, it is not sufficient to prevent noise-induced hearing loss.

Rubber-covered rollers are used and reduce the noise level to 80 dBA, but their application is limited because of the high temperature of the steel plate. The effect on noise reduction decreases very much as time goes by because of strain hardening of the rubber.

Research studies on impact noise are conducted mainly in the field of architectural engineering. Here, engineers focus on the floor impact noise and analyze structural-borne noise coupled with vibration. Besides floor impact noise, there are some other research studies, such as impact noise generation due to a wheel passing over rail joints, numerical analysis of impulsive noise generation and propagation, etc. However, there have been few research studies related to impact force. Impact force is difficult to deal with, because the impact phenomenon involves a complex interrelationship of energy and momentum transfer, energy dissipation, deformation, impact velocity, geometry and coefficient of restitution. Although in the classical theory the coefficient of restitution was considered as a constant for a given material, modern investigations show that the coefficient is highly dependent on the geometry and the impact velocity, as well as on the material.

Lately, two basic approaches, which differ in their treatment of colliding bodies, have been developed. The first approach simplifies the mathematical formulation by considering the colliding bodies as rigid. The second approach is more sophisticated, and the colliding bodies are allowed to undergo local and global deformations. The impact force, however, can be changed by so many system factors such as mass, velocity, material properties, shape, restitution coefficient, etc., that an accurate numerical formula can hardly be obtained. Also, the experimental approach is difficult because sensing the impact force is very difficult when the system is moving. Moreover, the impact force cannot be measured precisely because of time delay in the force sensor. Thus, finite element method is widely used to analyze the impact phenomenon.

In this study, we firstly introduce the noise index to quantify the relation between impact force and noise. Secondly, we analyze the impact force using the finite element method under various conditions. To reduce the impact noise, we selected the deflection as the controllable factor among various factors that affect the impact force. Finally, we carried out experiments to validate the FEM analyses.

2. Noise Index and Finite Element Formulation

Generally, two facts are known for an impact phenomenon. First, the impact noise increases with the impact force.
3. Model Definition and Dynamic Behavior Analysis

In the analysis, the plate is placed on rollers with the condition of contact and the rollers are rotating with a constant angular velocity of 6.4 rad/s. The gravitational force is acting on the plate, and the friction force is applied between the plate and the rollers.

Figure 2 represents the simplified model for the system. The roller size and the distance between rollers are taken from the field conditions. To reduce the impact noise, it is important to find the controllable factor. The mass, material and shape of rollers are fixed, and the velocity of plate transfer is maintained because of the production rate, but the deflection of the plate can be reduced by installing auxiliary rollers in-between the main rollers, so we selected deflection as the controllable factor for reduction of the impact noise. The meshed model for the current system is shown in Fig. 3.

The element for the plate was thin shell 163, and Belytschko-Tsay was used for the element formulation.11,12 On this shell element, two Gauss points are defined in thickness-directional integration.13 The roller is meshed by 3D Solid 164, and element formulation used for this element is fully integrated selectively-reduced solid.14

For the current system, dynamic behavior analyses for thickness values of 10, 12 and 20 mm were conducted, and the results are shown in Fig. 4. Definition of ‘x displacement’ and ‘deflection’ in Fig. 4 is shown in Fig. 5. To reduce the deflection, auxiliary rollers were positioned between the main rollers. Location of auxiliary rollers between the main rollers was designed to minimize the deflection of the steel plates considering install space in experiment. For this purpose, 12 cases of position were modeled and we selected the best one. In this model, the auxiliary roller was 150 mm in diameter and 300 mm in length. Top height of the auxiliary rollers is equal to that of the main rollers. Fig. 6 represents the location of the auxiliary rollers in the meshed model.

Results of dynamic behavior analysis for this model are shown in Fig. 7, and the deflections with and without the auxiliary rollers are compared in Table 1 for the case when the steel plate collides with the roller. From Table 1, we know that the auxiliary rollers greatly reduce the deflection which we select as the controllable factor for reduction of impact noise.

In addition, to check the efficiency of the auxiliary rollers, the dynamic behavior analysis for the 12 mm thick plate is carried out with the condition that the height of the auxiliary roller is 0.8 mm lower than the main roller. The

Second, the impact noise is greater if a higher frequency sound is greatly generated. The noise index is introduced using these two factors.11

The noise index is calculated as follows:

\[ R_N = \frac{R_{\text{max}}}{T_{\text{max}}} \] .................................\( (1) \)

where, \( R_{\text{max}} \) is the maximum impact force and \( T_{\text{max}} \) the time elapsed to reach the maximum impact force after collision. Figure 1 shows graphical interpretation of \( R_{\text{max}} \) and \( T_{\text{max}} \). Because a higher frequency sound will be greatly generated with a smaller \( T_{\text{max}} \), Eq. (1) can satisfy previous two facts.

To determine the noise index, two parameters, \( R_{\text{max}} \) and \( T_{\text{max}} \), have to be calculated. We calculate these values using the roller-plate impact simulation. An FEM code for simulation has been developed based on LS-Dyna, which is a general-purpose FEM code for analyzing the dynamic response of structures. The solution methodology is based on the explicit time integration. A Lagrangian formulation is used to describe the deformation. A weak form of the equilibrium equations is expressed by the principle of virtual work as follows:

\[
\delta \pi = \int_{V} \rho \delta \ddot{x} \, dv + \int_{\Gamma} \sigma_{ij} \delta \dot{x}_{i} \, ds \]
\[ - \int_{\Gamma} \rho \ddot{x} \, ds - \sum_{i=1}^{n} t \delta \xi_{i} ds = 0 \] .................................\( (2) \)

where \( x \) is the displacement, \( \ddot{x} \) the acceleration, \( \sigma_{ij} \) the Cauchy stress tensor, \( \rho \) current density, \( f \) the body force density, \( t \) the traction boundary conditions satisfying \( \sigma_{ij} n_{i} = t_{j}(t) \), respectively.

When the mesh of finite elements is superimposed and summed over the \( n \) elements, \( \delta \pi \) is approximated by:

\[
\delta \pi = \sum_{m=1}^{n} \delta \pi_{m} = \sum_{m=1}^{n} \left[ \int_{V_{m}} \rho \ddot{x} \Phi_{m}^{n} \, dv \right. \\
\left. + \int_{V_{m}} \sigma_{ij} \Phi_{ij}^{n} \, dv \right] - \int_{\Gamma_{m}} t \Phi_{i}^{n} ds = 0 \] .................................\( (3) \)

where \( \Phi_{i} \)'s are the shape functions of the parametric coordinates.

Equation (3) is solved numerically using the contact-impact algorithm based on the augmented Lagrange method.

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3. Model Definition and Dynamic Behavior Analysis

In the analysis, the plate is placed on rollers with the
result is shown in Fig. 8, where the deflection is 1.64 mm and increases by 1.06 mm compared with the case when the heights of the auxiliary and the main rollers are the same.

4. Impact Analysis

Impact analyses for the various conditions were carried out based on the explicit time integration. Considering the mesh and data size, moving velocity of the plate and rotat-

| Thickness of plates | 10 mm | 12 mm | 20 mm |
|---------------------|-------|-------|-------|
| w/o aux. rollers    | 9.64  | 7.02  | 2.61  |
| w/ aux. rollers     | 0.84  | 0.58  | 0.34  |

Fig. 8. Deflection when the height of the auxiliary rollers is 0.8 mm lower than the main rollers.
The ing velocity of the roller, we selected 5 μs as the time interval of data acquisition.

Results of impact analysis with and without the auxiliary rollers are shown in Fig. 9 and Fig. 10. $R_{\text{max}}$ and $T_{\text{max}}$ explained in Fig. 1 are obtained from the numerical analysis results and Table 2 shows $R_{\text{max}}$ and $T_{\text{max}}$ for various conditions. These results are used to calculate the noise index later. On the results, $R_{\text{max}}$ decreases as the deflection decreases for the same plate thickness but the values of $T_{\text{max}}$ are almost same for various conditions.

To check the effect of the deflection on the impact force, impact analysis for the 12 mm-thick plate and auxiliary rollers 0.8 mm lower in height than the main roller was carried out. The result is shown in Fig. 11.

The maximum impact force was 2 194 N. From the previous analysis results in Table 2, for the same plate thickness, the impact force was 3 947 N without the auxiliary rollers, and 1 100 N with the auxiliary rollers. So we can know that although the height of auxiliary rollers was lowered by only 0.8 mm, the efficiency of reduction of the impact force worsened by 38.4%.

5. Experiments

To validate the FE analysis, experiments were carried out. Figure 12 shows the experimental setup. In this study, we focus on the total sound pressure level due to impact force because impact force causes structural vibration by
which noise is generated. Also, the effect of structural mode is not considered because the modal response is magnified only according to the impact force.

Noise was measured using the B&K microphone, type 4189. Two B&K accelerometers, type 4393, were attached to a bearing block in the $x$ and $y$ directions. Here, the $x$ direction means the plate moving direction and the $y$ direction is the direction of gravitational force. Noise and acceleration data were acquired and analyzed using the FFT with the B&K PULSE software.

Noise by the surroundings was measured before the measurement of the impact noise to check the effect of the background noise. Generally, if the background noise is lower than the measurement noise by more than 10 dBA, it has little effect on the noise measurement. In this experiment the background noise level was 51.9 dBA. Since the impact noise was more than 90 dBA for all cases, it had no effect on the impact noise measurement.

The results of noise levels with and without the auxiliary rollers are summarized in Table 3. Table 3 shows that the noise level is reduced by more than 10 dBA when the auxiliary rollers are installed. The results indicate that the impact noise can be reduced by reducing the impact force.

The acceleration of bearing part in time domain was measured to comprehend the tendency of the impact force. In fact, the impact force cannot be obtained from the acceleration of bearing part but we can suppose that impact force is reduced if the acceleration decreases. The acceleration for the 10 mm-thick plate is shown in Fig. 13. In Fig. 13, the abscissa and ordinate denote time and acceleration. The upper graph of each figure represents acceleration in the $y$ direction and the lower graph in the $x$ direction. The accelerations in both directions are reduced when the auxiliary rollers are installed.

One can think that because the plate is moving in the $x$ direction, the acceleration in this direction will be greater. However, the results show that accelerations in both directions are almost the same. To compare the analytical results with the experiment, impact forces in the $x$ and $y$ directions are calculated for the 10 mm-thick plate without the auxiliary rollers. In this case, the impact forces were 2,574 and 2,660 N for the $x$ and $y$ direction, respectively, and the results agree with the experiment very well.

Figure 14 represents the noise spectrum for the 10 mm-thick plate. In Fig. 14, the abscissa and ordinate denote frequency and sound pressure level. Each figure represents the noise level in the frequency domain by performing the conversion to the 1/3 octave band. In this figure, the sound pressure level is reduced for all frequency range when the auxiliary rollers are installed.

The experiment for the auxiliary rollers with 0.8 mm lower height was conducted to validate the analytical result. The noise level was 107 dBA.

### Table 3. Results of noise level (dBA).

| Thickness of plates | 10 mm | 12 mm | 20 mm |
|---------------------|-------|-------|-------|
| w/o aux. rollers    | 112   | 110   | 107   |
| w/ aux. rollers     | 98.6  | 100   | 95    |

6. Results and Discussion

The noise index $P_N$ was calculated using Eq. (1) with $R_{\text{max}}$ and $T_{\text{max}}$ evaluated by numerical analysis. The values for various conditions are given in Table 4 with the impact forces and noise levels. The relationships among the noise index, the impact force and the noise level are shown in...
Figs. 15(a) and 15(b).

Figure 15 shows that noise level increases as the maximum impact force and noise index increase. However, in Figs. 15(a) and 15(b), the relationships are almost the same except for one point, because the $T_{\text{max}}$ values calculated are almost the same as 40 $\mu$s. Thus, the validity of Eq. (1) must be confirmed using different $T_{\text{max}}$ data. For this purpose, we carried out another impact analysis with the known noise data from the field. The model is composed of a plate and the rubber-covered roller shown in Fig. 16.

Impact analysis for the 3 mm-thick rubber and the 10 mm-thick plate without the auxiliary rollers is shown in Fig. 17.

At the results in Fig. 17, $R_{\text{max}}$ is about 620 N and is about 2 ms. Compared with $T_{\text{max}}$ (40 $\mu$s) of the previous analyses, the elapsed time to reach the maximum impact force increased about 50 times because of the rubber material. Figure 18 shows the relations among the impact noise, the impact force and the noise index including this result.

In Figs. 18(a) and 18(b), regression was performed to check the validity of Eq. (1). Linear and polynomial regressions are not suitable to express tendency, so nonlinear regression was performed and the R-square value was calculated. The R-square value is 0.7298 for Fig. 18(a) and 0.978 for Fig. 18(b). This means that the noise index reflects the tendency of the noise level better than the maximum impact force. It can be deduced that the noise level is affected not only by $R_{\text{max}}$, but also by $T_{\text{max}}$, simultaneously.

7. Conclusion

The current roller system used to transfer the steel plate makes a very loud impact noise. This work suggests a method of quantification and reduction of the impact noise that is generated by the roller and plate in the steel industry. The noise index was introduced to quantify the impact noise. FE analysis and an experiment were conducted to
validate it. The deflection was selected as the controllable factor to reduce the impact noise. The auxiliary roller was designed and installed to reduce the deflection of the plate.

The deflection of the plate was calculated using dynamic analysis with and without the auxiliary rollers. The impact between the roller and plate due to deflection of the plate was analyzed, and the impact force and the elapsed time required to determine the noise index were calculated. Finally, noise measurement was conducted experimentally and the impact noise was reduced by more than 10 dBA. Using the analytical and experimental results, the validity of the index noise was checked. Regression showed that although an accurate measurement of noise could not be obtained in this study, a rough estimate of the noise level by the proposed noise index is possible.

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