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Analysis of Mechanism of Unsteady Load in Strip Rolling Processes and Its Control in Commercial Lines

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In hot rolling process, it is a very important subject to improve and maintain the reliability of equipment under severe conditions arising from heat, water and impacts. Especially, the impact load, which appears at unsteady rolling of the leading and tailing ends of a strip, is an inevitable factor. So it is particularly important to analyze the mechanism of the impact load and estimate the impact load quantitatively, for the design and maintenance of equipment. In this paper, a formulation of the horizontal force to a roll, which is given by material at unsteady rolling, has been carried out by using a mechanical approach. Moreover, by using this formula, the mechanism of the roll movement has been made clear, and the impact load, which is produced by collision between roll chock and housing, has been quantified. In this analysis, it was proved quantitatively by using the mechanical model that a mechanical gap was one of the most effective causes of the amplification of this impact load. Furthermore, it was also shown that controlling the mechanical gap decreased the influence of the impact load. Based on the results of the analysis, improvement of the roll restrictive accuracy of the horizontal direction has been achieved and this improvement has contributed to the operational stabilization.

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Analysis of Mechanism of Unsteady Load in Strip Rolling Processes and Its Control in Commercial Lines*



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In hot rolling process, it is a very important subject to improve and maintain the reliability of equipment under severe conditions arising from heat, water and impacts. Especially, the impact load, which appears at unsteady rolling of the leading and tailing ends of a strip, is an inevitable factor. So it is particularly important to analyze the mechanism of the impact load and estimate the impact load quantitatively, for the design and maintenance of equipment. In this paper, a formulation of the horizontal force to a roll, which is given by material at unsteady rolling, has been carried out by using a mechanical approach. Moreover, by using this formula, the mechanism of the roll movement has been made clear, and the impact load, which is produced by collision between roll chock and housing, has been quantified. In this analysis, it was proved quantitatively by using the mechanical model that a mechanical gap was one of the most effective causes of the amplification of this impact load. Furthermore, it was also shown that controlling the mechanical gap decreased the influence of the impact load. Based on the results of the analysis, improvement of the roll restrictive accuracy of the horizontal direction has been achieved and this improvement has contributed to the operational stabilization.

1 Introduction

In 2000, the hot strip mill at Kawasaki Steel's Mizushima Works reached its 30th year of operation. Up to the present, a number of equipment modernizations have been carried out to improve the functions and reliability of this facility, and today it has secured a position as a high productivity plant. In the hot rolling manufacturing process, the loads for properly maintaining and controlling equipment functions are extremely large because the equipment is used under severe conditions of heat, water, and impact. Among these, the impact loads generated in the unsteady rolling region are unavoidable due to the characteristics of the rolling process. Furthermore, in attempting to improve the mill working function, the establishment of a technology that makes it possible to clarify the mechanism of such impact loads and measure them quantitatively, and countermeasures for their reduction, are important. In this report, the movement of rolls in the horizontal direction at the instant when the material bites into the rolling rolls and the impact forces generated by the collision between the roll chocks and housing are expressed in model equations using a dynamic approach. Confirmation of the effectiveness of the model from measured data and the results achieved in reducing impact forces by adopting of a stricter standard for mechanical gap control are also discussed. Finally, improvement of the roll restrictive function in the horizontal direction is described as a concrete example of equipment improvement.

2 Relationship between Mechanical Accuracy and Equipment Failures

The representative loads that act on a rolling mill in steady rolling, as shown in **Fig. 1**, are rolling torque,

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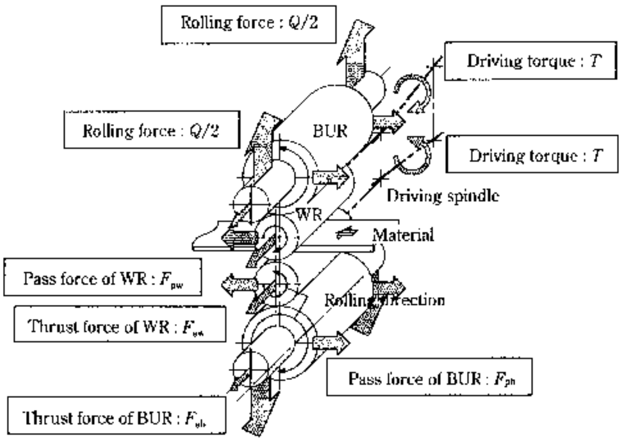


Fig. 1 Active load of roll of rolling mill

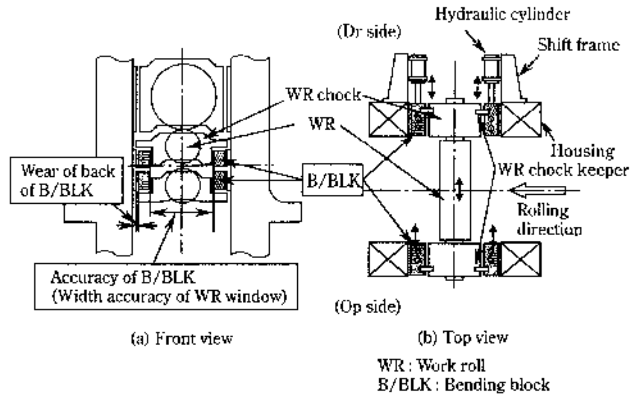


Fig. 2 Structure of finishing mill WR shift device

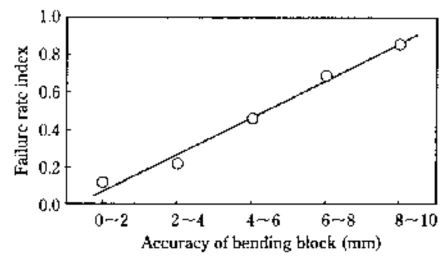


Fig. 3 Relation between mechanical accuracy and failure rate of WR shift device

rolling force, the axial force of the roll (thrust), and the rolling direction force of the rolls. In actual operation, the unsteady loads at metal-in (when the leading end of the material bites into the rolls) and metal-off (when the tail end leaves the roll bite) are larger than those in steady rolling, and are important for strength design. The existence of these unsteady loads is generally recognized and has been the object of a variety of research, both experimental and analytical. However, it has not been possible to obtain adequate knowledge in the sense of a quantitative evaluation of impact forces. This is because it is difficult to clarify systematically the mechanism of the occurrence of the impact forces themselves and observe the phenomenon of occurrence, in spite of the fact that actual measurements[1] and partial studies of impact forces have been conducted.

If the mechanical devices comprising a rolling mill are broadly divided into the driving system and the housing system, in the driving system, it is known that excessive torsional vibration occurs due to the effects of the torsional rigidity and mechanical gap of the respective parts at the time of material biting,[2-4] and this is used in the strength design of the driving shaft. On the other hand, the unsteady loads that act on the roll chock and the surrounding equipment that restrains it are considered to be a problem. Among these loads, in the rolling direction, it is considered that the gap between the roll chock and the housing window is a factor that amplifies the load, and this causes excessive impact force. However, the detailed mechanism of this phenomenon has not been elucidated.

In this chapter, in order to investigate the relationship between the gap (mechanical accuracy) and failure rate, we will make a correlative evaluation of the dimensional accuracy of the gap at the finishing mill housing window and the failure rate of accessory parts of the bending block.

The structure of a finishing mill work roll (WR) shift device is shown in **Fig. 2**. The results of arranging the relationship between the measurement accuracy of the bending block and the failure rate index based on data from a five year period are shown in **Fig. 3**. Here the bending block accuracy shows the amount of sliding abrasion on the back side of the bending block. When the amount of abrasion is small, the housing window gap is also small. As is clear from Fig. 3, as the accuracy of the bending block deteriorates, the failure rate increases. It is therefore considered that an increase in unsteady loads due to reduced accuracy plays a role in the root cause of failures.

3 Development of Model Equations for Evaluation Unsteady Loads

3.1 Measurement of Roll Behavior and Impact Forces

When material which is being rolled bites into the rolls, the rolls receive force from the material in the rolling direction, and an impact force is generated by collision between the roll chocks and the mill housing. In order to understand the movement of the rolls and the impact forces at this time, measurements were made at the upper and lower intermediate rolls (IMR) of the back stand of a finishing mill. The measurement items were the amount of gap displacement between the upper roll chock and housing and the value of the vibration acceleration of the roll changing rails at the entry and delivery sides (**Fig. 4**). It should be mentioned that the roll changing rails have a structure that enables vertical movement during roll changes, and these devices also serve as the window block of the lower IMR. A chart of the vibration acceleration at the roll changing rails is shown in **Fig. 5**, and a gap displacement chart is shown in **Fig. 6**.

As to the behavior of rolls, during roll racing, it can be understood from the charts in Figs. 5 and 6 that the rolls, which had been pressed against the entry side housing by the roll offset force component, move from the entry side to the delivery side in a period of approximately 40 ms after metal-in, after which they return to the entry side. Moreover, it can also be understood that an excessive vibration acceleration of 30 to 40 G is generated in the roll changing rails by the impact with the chocks when the rolls move from the entry side to the delivery side and vice versa.

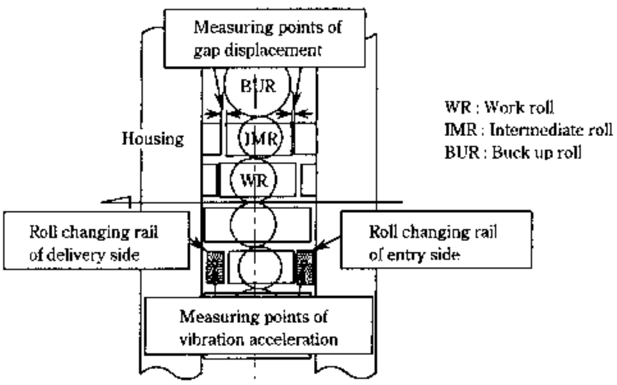


Fig. 4 Measuring points at finishing mill

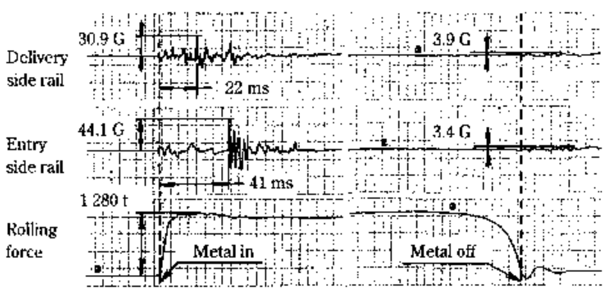


Fig. 5 Chart of vibration acceleration at roll changing rail

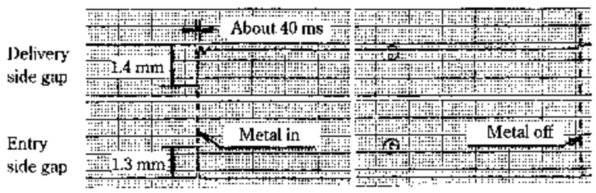


Fig. 6 Chart of gap displacement at upper IMR

3.2 Model of Horizontal Roll Movement in Steady and Unsteady Rolling

Based on the knowledge obtained by the measurements described in the previous section, model equations that enable dynamic evaluation of the impact phenomena involving the housing and roll chocks were developed. A conceptual diagram of the model is shown in **Fig. 7**.

When a roll assembly having mass M receives force F in the pass direction (horizontal direction), after moving across the gap δ while accelerating, the roll achieves a velocity v immediately before impact with the housing. When the roll chock strikes the housing, the kinetic energy of the roll chock and the elastic deformation energy of the housing are in balance, as shown in Eq. (1), and elastic bending e occurs in the housing. The impact force P that generates this bending e acts between the housing and the chock as an impact force at the time of collision with the housing. P_t in Eq. (2) is the resultant force that acts in the pass direction.

$$\frac{Mv^2}{2} = Pe \cdots\cdots (1)$$

$$P_t = F + \sqrt{2K_h F\delta} \cdots\cdots (2)$$

Figure 8 shows the results of a numerical calculation

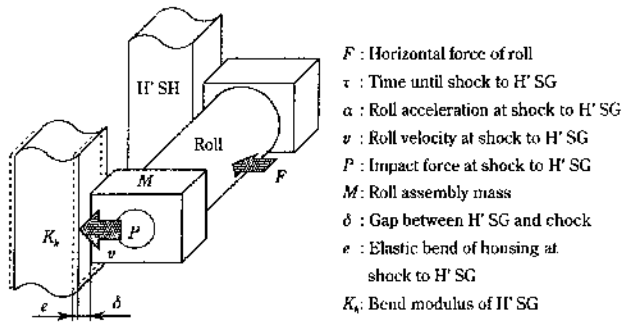


Fig. 7 Backlash impact model between housing and chock

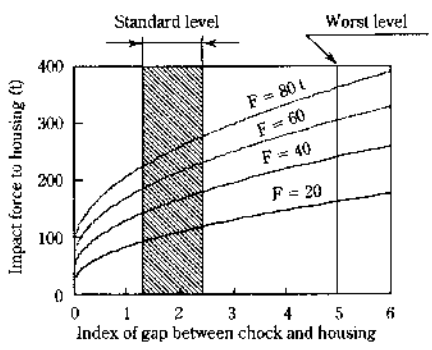


Fig. 8 Impact force between chock and housing

when the transverse rigidity K_h of the housing of a hot rolling finishing mill and the horizontal force F that acts on the roll are input as parameters. These calculated results were obtained for the lower intermediate roll of the back stand (6Hi) of the finishing mill.

The knowledge obtained here is summarized below.

(1) The impact force that acts on the housing increases in proportion to the 1/2 power of the gap δ.
(2) At standard gap levels, the resultant impact force increases by 30–70% under a condition in which the gap is increased by approximately 4 times.
(3) At standard gap levels, the resultant force of impact is reduced by 60–80% if the housing gap can be reduced to 0.

An equation can be derived for the relationship between the housing gap and impact force in the pass line direction. Moreover, if the horizontal force of the roll F can be quantified, it is also possible to calculate the impact force in an actual rolling mill.

In the following, an equation was developed for the horizontal force F.

The model was developed for a 6Hi mill, which has 6 rolls. The horizontal forces that act on the rolls during rolling can be divided into the following components.

(1) Horizontal component force of tangential force of the torque that acts in a tangential direction to the area of contact between the rolls when the rolls are driven
(2) Horizontal component force of the rolling force due to roll offset
(3) Horizontal component force due to forward and backward tension from the strip acting on the WR

First, a conceptual diagram of the model of the forces during roll racing and steady rolling is shown in **Fig. 9**. During roll racing, the horizontal component torces obtained from the equilibrium equation of the tangential components of the torque acting between each pair of rolls can be expressed as shown in Eqs. (3)∼(5).

$$\text{BUR: } R_b = F_r \cos\phi_{bi} \quad \cdots\cdots (3)$$

$$\text{IMR: } R_i = -F_r \cos\phi_{bi} - G_r \cos\phi_{wi} \quad \cdots\cdots (4)$$

$$\text{WR: } R_w = G_r \cos\phi_w \quad \cdots\cdots (5)$$

Here, the directions of the horizontal forces are defined as positive in the delivery side direction and negative in the entry side direction. Because the torque during roll racing can be considered to be virtually 0, the horizontal component forces are also 0.

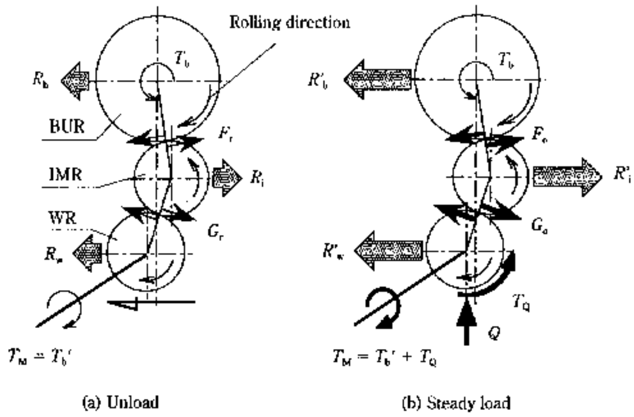
For steady rolling, the horizontal component forces of the rolling force due to the roll offset angle were added to Eqs. (3)∼(5), resulting in Eqs. (6)∼(8).

$$\text{BUR: } R_b = (F_r + F_o)\cos\phi_{bi} \quad \cdots\cdots (6)$$

$$\text{IMR: } R_i = -(F_r + F_o)\cos\phi_{bi} - (G_r + G_o)\cos\phi_{wi} \quad \cdots\cdots (7)$$

$$\text{WR: } R_w = (G_r + G_o)\cos\phi_w \quad \cdots\cdots (8)$$

Here, the horizontal component forces were calculated from Eqs. (6)∼(8) by inputting the values of the diameter of the respective rolls, the amount of offset,



D_o, D_i, D_w: Diameter of each roll
Φ_{bi}, Φ_{wi}: Offset angle of each roll
T_M: Motor driving torque on WR shaft
T_b: Frictional resistance of BUR mogoil bearing
T_b': Frictional resistance of BUR mogoil bearing on WR shaft
F_r: Tangential force of driving force between BUR and IMR
G_r: Tangential force of driving force between IMR and WR
Q: Rolling force
T_Q: Rolling torque
F_o: Tangential force of rolling force between BUR and IMR
G_o: Tangential force of rolling force between IMR and WR

Fig. 9 Principle model of steady roll horizontal force

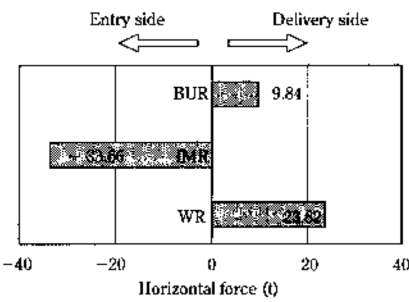


Fig. 10 Horizontal force of each roll at steady rolling

and the average rolling force at an actual mill. The results of this calculation are shown in **Fig. 10**. From this, it can be understood that, during steady rolling, the WR and BUR receive horizontal force in the delivery side direction, and the IMR receives horizontal force in the entry side direction.

Next, the horizontal forces acting on rolls during unsteady rolling will be considered. A conceptual diagram of the model is shown in **Fig. 11**. In this case, if a kinetic equation is obtained for the respective rolls at the instant of material biting, and assuming hypothetically that no slipping occurs between the rolls, F_d and G_d can be obtained as Eqs. (9) and (10), respectively.

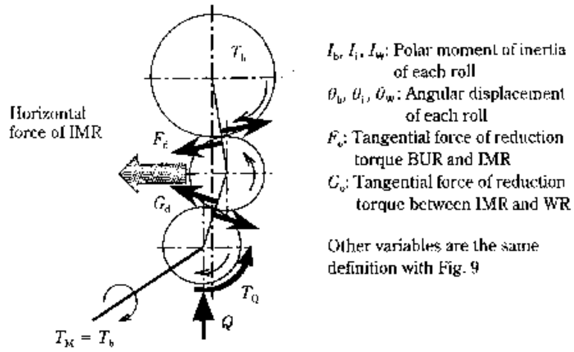


Fig. 11 Principle model of unsteady roll horizontal force

$$F_d = \frac{\frac{D_i}{D_w}(T_M - T_Q) + \left(\frac{I_iD_b}{I_bD_i} + \frac{I_wD_iD_b}{I_bD_w^2}\right)T_b}{\frac{D_b}{2}\left(\frac{I_iD_b}{I_bD_i} + \frac{I_wD_iD_b}{I_bD_w^2}\right) + \frac{D_i}{2}} \cdots\cdots (9)$$

$$G_d = \frac{\left(\frac{I_iD_w}{I_wD_i} + \frac{I_bD_wD_i}{I_wD_b^2}\right)(T_M - T_Q) + \frac{D_i}{D_b}T_b}{\frac{D_b}{2}\left(\frac{I_iD_b}{I_bD_i} + \frac{I_wD_iD_b}{I_bD_w^2}\right) + \frac{D_i}{2}} \cdots (10)$$

Considering also the horizontal component forces of the force acting between the rolls as a result of deceleration torque in Eqs. (9) and (10), the horizontal component force of each roll can be expressed by Eqs. (11)~(13).

$$\text{BUR: } R_b = (F_d + F_r + F_o)\cos\phi_{bi} \cdots\cdots\cdots (11)$$

$$\text{IMR: } R_i = -(F_d + F_r + F_o)\cos\phi_{bi} - (G_d + G_r + G_o)\cos\phi_{wi} \cdots\cdots (12)$$

$$\text{WR: } R_w = (G_d + G_r + G_o)\cos\phi_w \cdots\cdots\cdots (13)$$

Here, the horizontal component forces were calculated by inputting the respective numerical values into Eqs. (11)~(13) in the same manner as with steady rolling, with the results shown in **Fig. 12**. From this figure, it can be understood that the horizontal component forces act on the respective rolls act in the opposite directions from those in steady rolling.

The results of calculations using these model equations are summarized below.

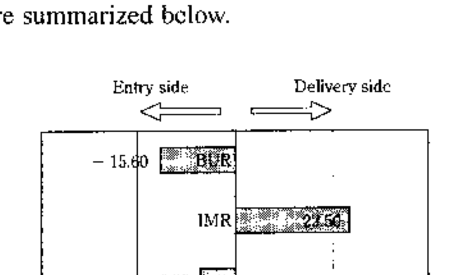


Fig. 12 Horizontal force of each roll at unsteady rolling

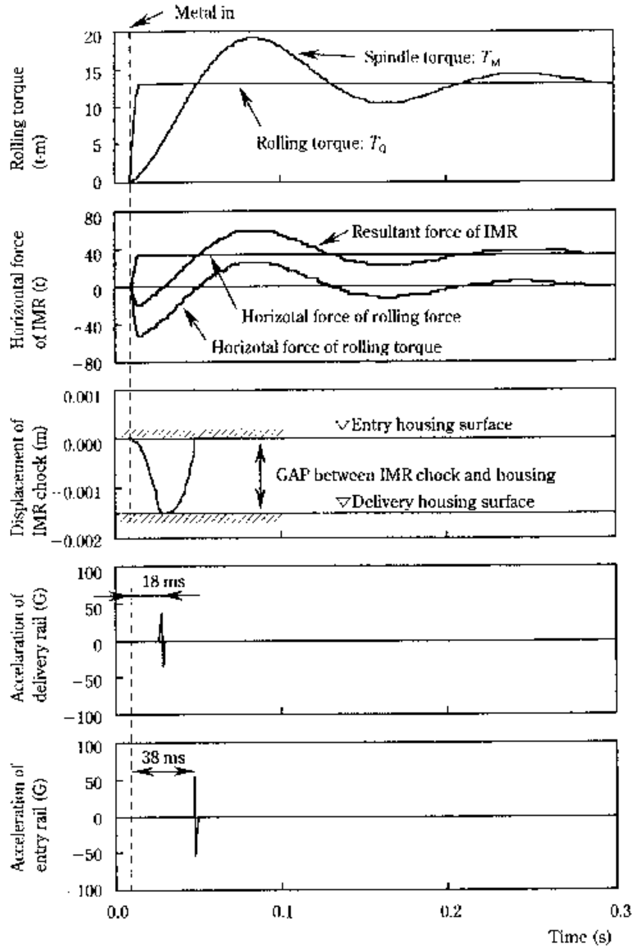


Fig. 13 Simulation of roll displacement and rail acceleration

(1) During mill racing and steady rolling, the rolls receive horizontal forces which are constant in the same direction. In this case, the rolls are pressed against the housing on one side and are stable.
(2) In unsteady rolling during material biting, the rolls receive horizontal forces in the opposite direction from those in the steady state, and are pressed against the housing on the opposite side.
(3) When the mill returns to the steady state after biting is completed, the rolls again receive horizontal forces in the original direction and return to their original positions against the mill housing.
(4) The roll chocks and housing receive impact due to the movements in (2) and (3), and the impact value is proportionate to the 1/2 power of the gap between the housing and chock.

These calculated results provide qualitative support for the measured results of the roll behavior and impact acceleration presented in section 3.1. For a quantitative comparison, a simulation of the IMR horizontal force, roll displacement, and vibration acceleration at the roll changing rails during unsteady rolling was carried out,

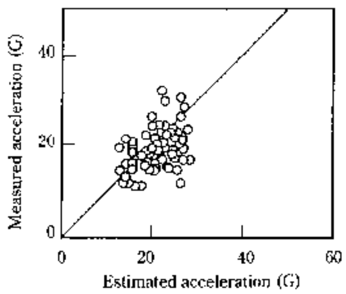


Fig. 14 Comparison of calculated and experimental acceleration at roll changing rail

with the results shown in **Fig. 13**. A comparison of the calculated and measured results of the values of vibration acceleration at the roll changing rails is shown in **Fig. 14**.

In the results obtained here, the vibration acceleration received by the roll changing rails after the material bites, the time required until acceleration occurs, and the absolute values were all virtually the same as the experimental results. Because substantial quantitative agreement can also be seen from the results in Fig. 14, it can be said that it is possible to quantify the impact force in the pass direction using this model.

Furthermore, these results showed that it is necessary to stabilize the behavior of the chocks, in other words, either to eliminate the gap between the housing and the chock, or to maintain a small gap over the long term, in order to reduce the impact force.

4 Equipment Technology for Reducing Unsteady Load

4.1 Structure and Problems Before Improvement[5-7]

In the WR shift device at the fore stand of the finishing mill of the Mizushima Works hot strip mill, a sliding bending block type had been adopted. Structurally, as shown in **Fig. 15**(a), this is a device in which the bending block slides in the shift direction, following the roll, during WR shifts. The advantage of this type is that the offset load does not act on the roll neck bearing because the relative positional relationship of the roll and bending block is constant at all times. Accordingly, because this type can be designed with a relative large shift stroke, it has an excellent flatness control function. However, as a disadvantage, the back surface of the bending block serves as the sliding surface, and the large velocity of abrasion deterioration in this position was a problem. As abrasion deterioration of the back surface progresses, the gap between the roll chock and the bending block becomes larger, and the unsteady load in the roll pass direction increases. This increases the danger of failures of the bending block auxiliary equipment and

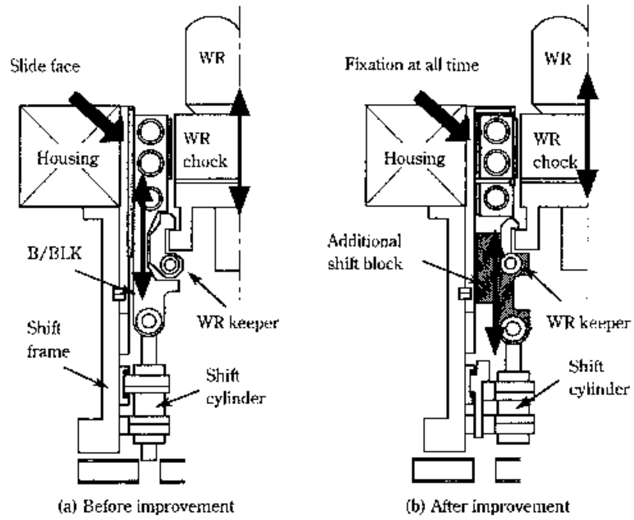


Fig. 15 Structure of WR shift

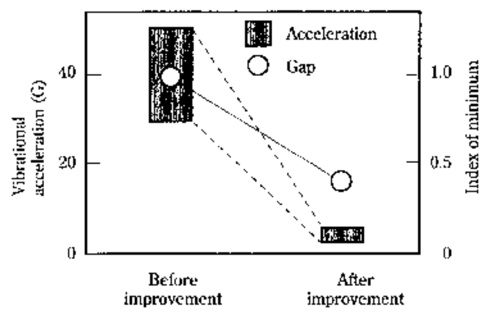


Fig. 16 Comparison of vibrational acceleration and accuracy of housing window

other parts.

4.2 Structure After Improvement

In order to solve this problem, the WR shift device was improved to a fixed bending block type WR shift structure. In fixing the bending block, a new shift block was installed as a medium for linkage between the roll and the shift cylinder. Ingenuity was used in the structural design[8] so that moment loads are not generated in this block by the rolling thrust force. Further, to cope with the offset bender load that acts on the WR neck bearings, an offset load allowance type bearing was developed[9,10] and applied, making it possible to maintain the required flatness control function. This completely eliminated sliding abrasion of the back surface of the bending block, and made it possible to maintain the accuracy of the window gap and greatly reduce unsteady impact forces. After this improvement, the area of contact between the roll chock and the bending block became the sliding surface. However, liners are used to enable easy replacement of this sliding surface, reducing the maintenance load. As one benefit of this improvement, as shown in **Fig. 16**, the vibration acceleration generated in the bending block structure, which had conventionally been 30–40 G, was reduced to 2–5 G after modification of the WR shift device. Following this improvement, the minimum window gap control standard was reviewed, and a stricter standard of less than 1/2 of the conventional value was adopted. This also contributed to improved threading.

5 Conclusion

To solve problems associated with impact forces in the roll pass direction, which occur in the unsteady rolling region when the material bites into the rolls in the hot rolling process, a theoretical study was conducted using a dynamic approach. As a result, the following knowledge and results were obtained.

(1) Roll movements during rolling and the pass direction forces acting on the rolls were quantified by developing equations for a dynamic model of the impact forces in the roll pass direction during steady and unsteady rolling.

(2) The relationship between the gap and impact forces was clarified using an equation for evaluating the effects of pass direction impact forces, with the gap as a parameter.

(3) Because pass direction forces are unavoidably generated due to the characteristics of rolling dynamics, it was shown theoretically that adopting a stricter standard for the gap is an effective means of reducing impact forces.

(4) A large reduction in the pass direction impact force was possible by improving the finishing mill WR shift device.

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