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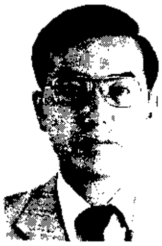

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In recent years, the authors have been endeavoring to revamp steel rolling processes by realizing continuous and syncronized production between two processes in order to achieve higher quality of products. In cold and hot tandem mills or continuous annealing process lines, which required high response and high accuracy to the motor control system, the authors encountered troubles with shaft vibrations caused by interaction between mechanical and electrical control systems, and developed a new power drive technique which was effective in solving the problems. And authors were able to understand the influence of all the digital thyrister motor drive system and the cross current type cycloconverter drive system on the shaft vibration problem through computer simulation analyses and experiments. As a result, the following were found effective in suppressing shaft vibrations: (1) to apply a digital filtering method to speed feedback, (2) to control a speed control loop in high speed sampling time and high accuracy calculation, and (3) to apply the modern control theory.

1 Introduction

In recent years, conspicuous technical progress has been made in realizing continuous, synchronized, and high-speed operation in steel rolling processes and in upgrading the quality of rolled products. Particularly significant has been the contribution of power-electronics (mainly, microcomputers, power conversion elements, and control technologies). On the other hand, frequent problems have arisen regarding shaft vibration between the mechanical shaft system and the motor drive system in the rolling mills. Shaft vibration may present problems because of certain technical considerations related to modern production practices: Equipment systems now use small-diameter rolls and operate at higher speeds, motor drive systems are subjected to fully-digitized control through the use of microcomputers, and the use of ac drive systems has dramatically improved speed response. In addition, rolling mills develop torque ripple with some control systems. This paper discusses the analysis of shaft bivration phenomena in the actual rolling process, measures to cope with such vibration, and, further, shaft-vibration suppression control methods.

2 Causes of Shaft Vibration

Causes of shaft vibration in the mechanical and motor-drive systems include the following:

(1) Torsional Vibration in Mechanical Drive System
The use of smaller rolls, accompanying improvement in rolling characteristics, and the development

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of high strength material for mechanical structures has led to a decrease in shaft cross section, and a decrease in the mechanical torsion natural frequency N_f of equipment. The torque amplifying factor TAF at the resonant point has also been adversely affected.

(2) Flexural Bending Vibration in Mechanical Drive System
When flexure, eccentricity or imbalance exists in the shaft system, the system vibrates at integermultiples of the rotary frequency of the roll- and motor-shafts. If, because of higher operating speeds, N_f lies within the actual-use rotary frequency, thorough examination is indicated.

(3) Vibration of Mechanical Structures and Foundation

(4) Vibration of Strip Being Rolled
Clogging, slipping, changes in deformation characteristics, changes in the inertia (GD^2) of steel, and load imbalance between upper and lower rolls may cause vibration.

(5) Vibration of Motor Drive System
To improve strip-thickness and strip-width accuracy and achieve high-speed and stabilized tracking in rolling, advances in power electronics have been applied, and fully-digitized control and ac drive control have been introduced in the drive system, realizing higher accuracy and higher response. Causes of vibration in the motor drive system include the following:

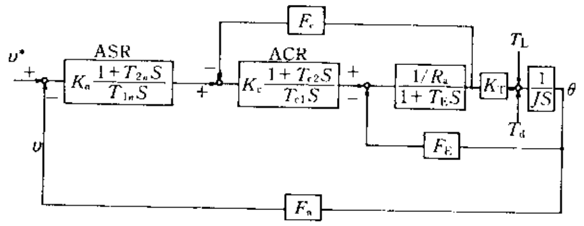
(a) With major improvements in speed response, cross-over frequency ω_c has increased from 5–10 rad/s to 30–60 rad/s.
(b) Fully-digitized control involves a problem of the sampling-hold of signals and in some circumstances such control causes deterioration of the vibration suppression effect to levels lower than with conventional analog continuous control.
(c) Torque ripples from the inverter drive and from the cycloconverter non-circulating current drive may also cause vibration.

Here, the following items are important and require examination: the relation between ω_c and N_f, the sampling time, speed detection accuracy, filtering, a low torque ripple ac drive system, and vibration suppression control measures.

3 Models of Motor Control System and of Mechanical System with Torsional Vibration in Rolling Mills[1,2]

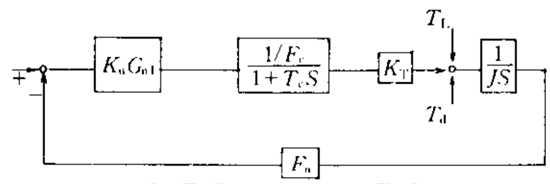
3.1 Vibration Model with Control System and Mechanical System Coupled

Figure 1 shows a block diagram of the basic motor speed control system when the mechanical system is considered as a rigid body. A block diagram prepared



v^*: Speed reference (rpm)
v: Speed feedback (rpm)
$\dot{\theta}$: Angular velocity (rpm)
T_d: Torque disturbance (kgf·m)
T_L: Load torque (kgf·m)
R_a: Armature circuit resistance (Ω)
F_E: Voltage coefficient (V/rpm)
F_n: Speed feedback gain (mpm/rpm)
F_c: Current feedback gain
S: Laplacian
K_T: Torque coefficient
$K_c\frac{1+T_{c2}S}{T_{c1}S}$: Transfer function of ACR (automatic current regulator)
$K_n\frac{1+T_{2n}S}{T_{1n}S}$: Transfer function of ASR (automatic speed regulator)
$J(=J_1+J_2)$: Total moment of inertia of motor and mechanical system (kgf·m²)
J_1: Moment of inertia of motor
J_2: Moment of inertia of mechanical system

Fig. 1 Block diagram of main motor speed control system



$G_{n1} = \frac{1+T_{2n}S}{T_{1n}S}$ $\qquad T_c = \frac{T_{c1}R_a}{K_c F_c}$

Fig. 2 Simplified block diagram of Fig. 1

by equivalent-conversion of Fig. 1 is shown in **Fig. 2.** In this system, the open-loop transfer function $G_I(S)$ is obtained by

$$G_I(S) = G_{n0} \times \frac{1}{JS}$$

where

$$G_{n0} = \frac{F_n K_n G_{n1} K_T}{F_c(1 + T_c S)}$$

The Bode diagram of this system is shown in **Fig. 3.** In general, the fact that the mechanical torsion natural frequency of a mechanical system is three times or more the cross-over frequency ω_c of the control system constitutes a condition for determining stability in practical use.

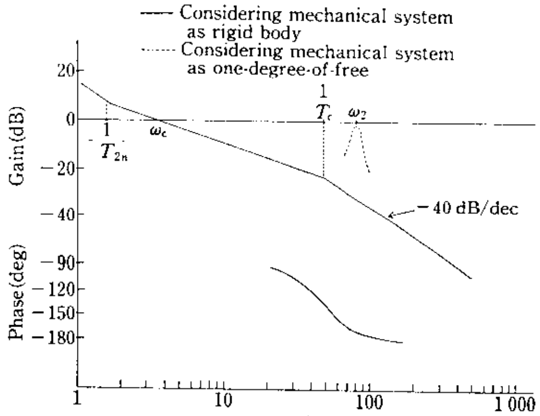


Fig. 3 Bode diagram of speed control system

3.2 Vibration Model of Mechanical System Considering Torsion

A block diagram of this model is shown in **Fig. 4**; a model taking into consideration the torsion of the mechanical system is shown in **Fig. 5**. In the system, the open-loop transfer function $G_{\text{II}}(S)$ is obtained by

$$G_{\text{II}}(S) = G_{n0} \times \frac{1}{J_1 S} \times \frac{S^2 + 2\zeta_2\omega_2 S + \omega_2^2}{S^2 + 2\zeta_1\omega_1 S + \omega_1^2}$$

where

$$\omega_1 = \sqrt{K_{12}\left(\frac{1}{J_1} + \frac{1}{J_2}\right)}$$

$$\omega_2 = \sqrt{\frac{K_{12}}{J_2}}$$

The Bode diagram of this mechanical system is given in Fig. 3. This system becomes unstable at $1/T_c > \omega_2$ because its gain exceeds 0 dB, while it is stable at $1/T_c < \omega_2$. As mentioned earlier, ω_2 in the mechanical system tends to drop when ω_c in the control system increases. If the values of ω_c and ω_2 become too close, the system becomes unstable. Further, if there is any little wasted time, systems become unstable and generate vibration. Excessive play of couplings and gears in the mechanical system equivalently lowers mechanical system rigidity.

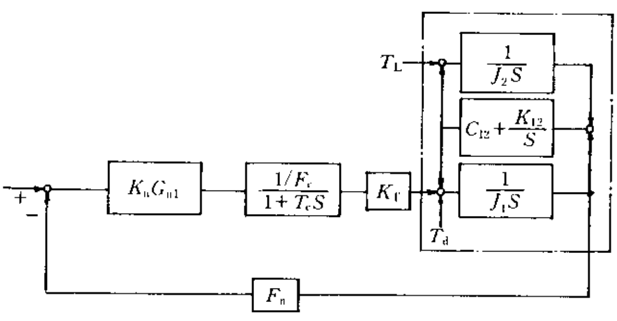
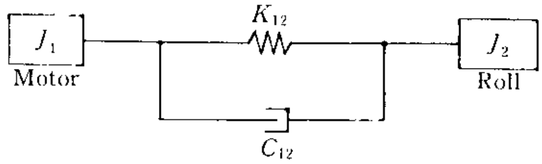


Fig. 4 Block diagram of speed control system considering mechanical system as one-degree-of-free



K_{12}: Equivalent torsional spring constant
C_{12}: Equivalent damping coefficient

Fig. 5 Mechanical resonance system

4 Effect of Fully-Digitization of Thyristor Leonard Drive System on Shaft Vibration in Rolling Mills

4.1 Shaft Vibration in Cold Tandem Mill and Its Analysis

4.1.1 Shaft vibrations of cold tandem mill

To improve strip thickness, accuracy, and rollability, analog-type thyristors in Mizushima Works' cold tandem mill main drives were revamped with fully-digital-type thyristor drive systems[3]. In particular, the No.5 stand was mechanically modified by decreasing the diameters of rolls and remodelling the speed-up gears. **Table 1** shows the general specification of the

Table 1 Specifications of No.5 stand

	Conventional	After replacement
Mill Type	4Hi	6Hi
Motor power	1 350 kW × 2 × 2	1 350 kW × 2 × 2
Motor drive	Twin drive (Mechanical tie)	
Motor (rpm)	200/580	200/580
Gear ratio	1.354/1	2.524/1
Control	ASR	ASR
Roll size (mm)	610 × 1 730 1 520 × 1 730	420 × 1 730 530 × 1 730 1 350 × 1 730
Screw-down	Hydraulic	
Strip thickness (mm)	1.8～6.0/0.15～3.2	
Strip width (mm)	600～1 600	
Coil weight (tf)	50 max	
Line speed (m/min)	1 500	1 930

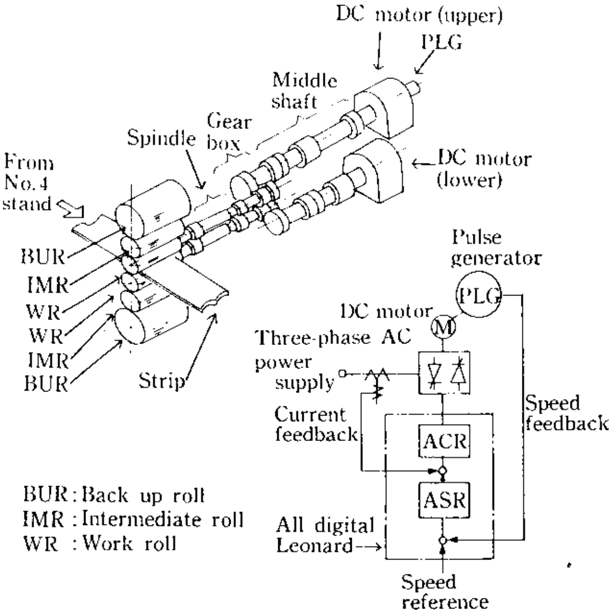


Fig. 6 Configuration of No.5 stand in the 5-stand cold tandem mill at Mizushima Works

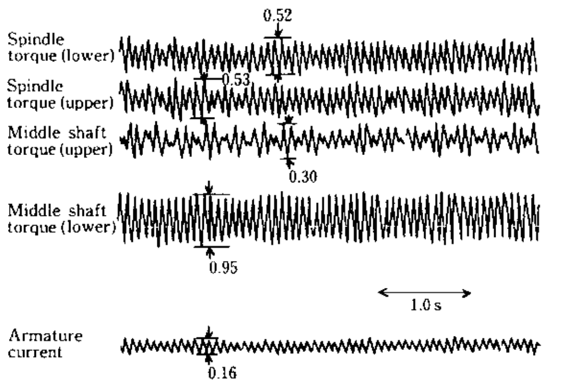


Fig. 7 An example of shaft vibration in 5-stand cold tandem mill at Mizushima Works (No.5 stand shaft torque and armature current measured at line speed of 1 018.3 m/min during natural deceleration)

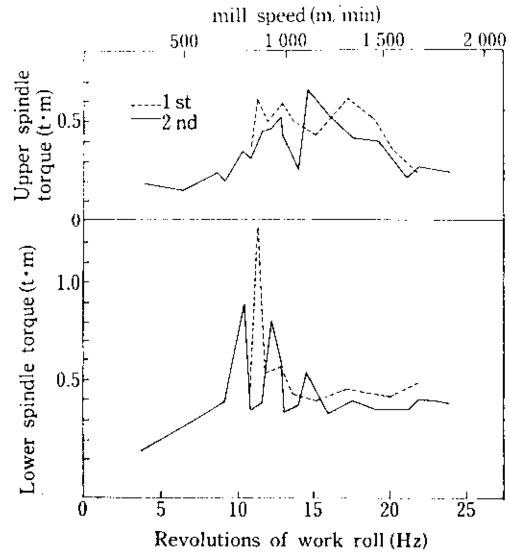


Fig. 8 Relationship between amplitude of vibration and revolutions of work roll

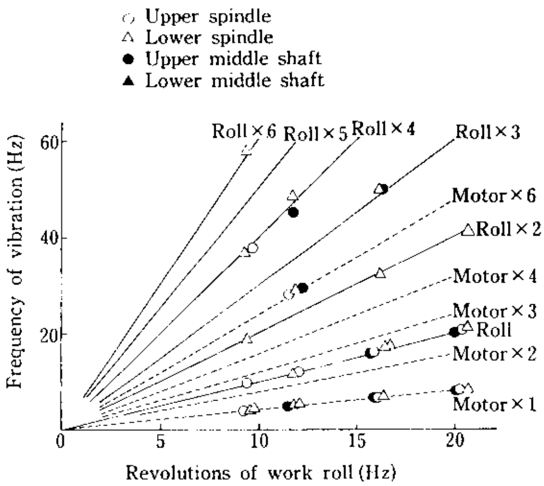


Fig. 9 Relationship between revolutions of work roll and frequency of vibration

No.5 stand; **Fig. 6** shows its mechanical configuration and a schematic diagram of the speed control system. After the modification, it was found that the shaft torque vibrated at a certain roll-rotation frequency. **Figure 7** indicates that torque vibration with a frequency of about 13 Hz and an amplitude of 0.5 to 0.6 (assuming a mean torque of 1 at such time) occurred at a roll speed of about 1 000 m/min.

Therefore, rolls were isolated from the effect of the drive system, roll speed was decelerated naturally from top speed, and the relation between spindle torque and roll speed was investigated. The results are shown in **Figs. 8** and **9**. This experiment yielded the following conclusions:

(1) Upper and lower spindle torque values show a large degree of vibration in the roll rotation frequency range of 11 to 13 Hz; this agrees very closely with the calculated value of mechanical torsion natural frequency of 13.1 Hz.

(2) Vibration frequency is an integer-multiple of the roll-spindle and motor-shaft rotation frequency. Vibration phenomena include the flexural vibration of the extended axes of the respective spindles.

(3) Vibration phenomena, in which (1) and (2) mentioned above are superimposed one upon another, are further affected by use of a digital-type thyristor drive.

4.1.2 Simulation analysis comparison of analog control and digital control of thyristor Leonard control

To make a strict comparison, a coupled model was prepared. This model consisted of the motor control system and mechanical system of an actual program (the actual processing system of an 8086 microcomputer). Denoting the excitation source by x_d and generated torque by y, vibrations were compared and evaluated as y/x_d. The results of comparison of the analog-type and digital-type thyristor speed control systems are shown in **Fig. 10**. This simulation analysis clarified the following:

(1) Analog speed control shows smaller y/x_d than digital control, and has a greater vibration suppression effect. This suggests that since sampling time for the analog type is 0, a better vibration suppression effect can also be obtained in digital control by minimizing sampling time.

(2) With digital control, an increase in the current control system response speed, ACR ω_c (rad/s), from 40 to 60 improves the vibration suppression effect.

(3) With digital control, vibration tends to be amplified if the response speed, ASR ω_c (rad/s), of the high-speed control system is increased.

Next, the filtering method in the speed detection feedback system was altered, and a simulation analysis was made using the actual thyristor Leonard program. The following three filtering methods were examined (**Fig. 11**):

(1) Mean speed detection method

$$N_i = \frac{n_i + n_{i-1}}{2}$$

(2) Two-point speed detection filtering method

$$N_{\mathrm{II}i} = N_i + \frac{N_i - N_{i-1}}{2}$$

(3) Three-point speed detection filtering method

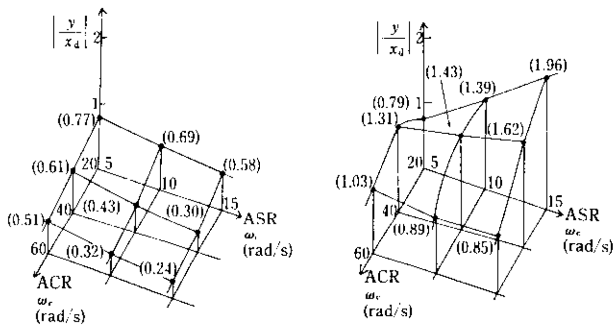
$$N_{\mathrm{III}i} = N_i + \frac{N_i - N_{i-1}}{2} + \left[\frac{N_i - N_{i-1}}{2} - \frac{N_{i-1} - N_{i-2}}{2}\right]$$

The results of analysis by these three methods are shown in **Fig. 12**, and indicate that the three-point speed detection filtering method has a superior vibration suppression effect.

Based on the results of this analysis, the following measures were taken in the actual process:

(1) Speed feedback using the three-point speed detection filtering method was applied.

(2) The response of the current control system was enhanced from 40 to 60 rad/s.



(a) Analog ASR (b) Digital ASR

Fig. 10 Comparison of the effect of analog ASR on shaft vibration with that of digital ASR in simulation test

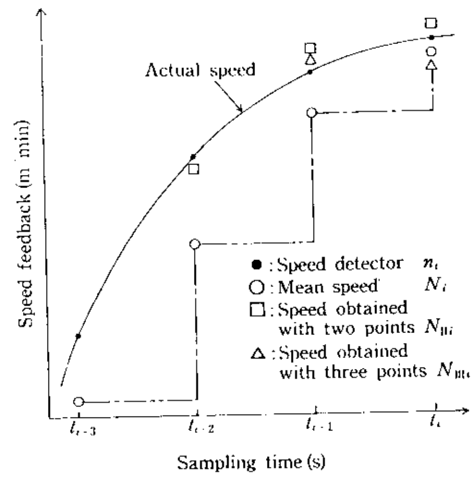


Fig. 11 Filtering methods by detecting three points of speed

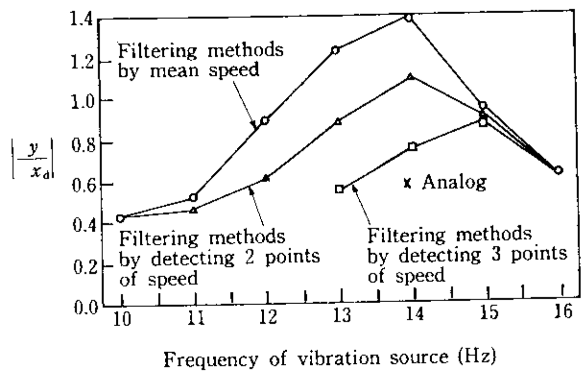


Fig. 12 Influence of filtering methods on value of $|y/X_d|$

(3) The response of the speed control system was not altered (7 rad/s).

Through these vibration suppression measures, the magnitude of shaft vibration, y/x_d, was reduced to 54% of the pre-measurement value.

4.2 Effect of Speed Feedback Filter of Digital-type Thyristor Leonard Control System on Speed Response and Its Shaft Vibration Suppression Effect

4.2.1 Characteristics of speed feedback filter (mean value filter)

Speaking in micro terms, actual speed fluctuates in tens of milliseconds of sampling time. When the speed fluctuates at motor rotating speed (feedback) $N = A + B \sin \omega_t$, the speed at time t is given by the following formula if the mean value of the speed during T sec is assumed to be feedback value n_t:

$$n_t = \frac{1}{T}\int_{t-T}^{t} (A + B \sin \omega t)dt$$

$$= A + \left(\frac{2B}{\omega T} \sin \frac{\omega T}{2}\right) \sin \left(\omega t - \frac{\omega T}{2}\right) \ .$$

Therefore, the AC-fraction amplitude which is superimposed on the speed feedback is damped to $(2/\omega T) \sin (\omega t/2)$, and the phase is delayed by $\omega T/2$. These characteristics are shown in **Fig. 13**.

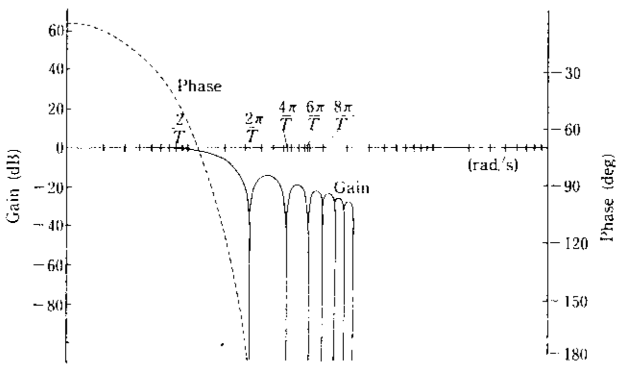


Fig. 13 Bode diagram of speed feedback with filter

4.2.2 Influence on speed response and shaft vibration suppression effect

A Bode diagram of the open loop transfer function of speed control is shown in **Fig. 14**. Without the filter, the gain is damped at 40 dB/dec at a break point of 200 rad/s, while, with a filter (time constant: 52 ms), gain is damped at a break point of 120 rad/s, where the mechanical torsion natural frequency is approximately 20 Hz. The gain near the resonance point is lowered by about 10 to 15 dB. Without a filter, the phase margin is 69°; with a filter, the phase margin becomes smaller, to 46°, but this does not pose a problem in actual use. In this way, speed response performance was maintained at

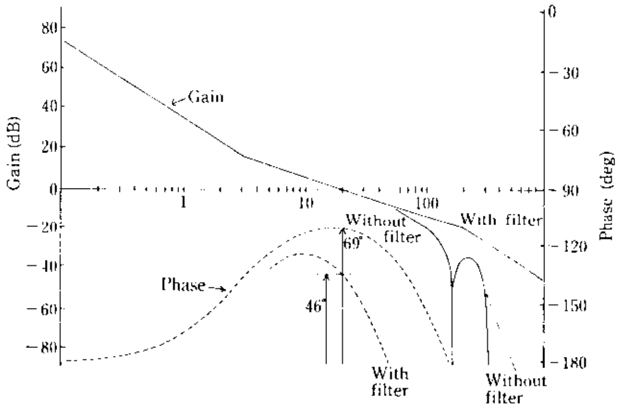


Fig. 14 Bode diagram of open loop (ASR)

20 rad/s, and shaft vibration was suppressed without causing deterioration of rolling characteristics.

5 Increased Response of Speed Control System by Cold Tandem Mill AC Drive and Shaft Vibration Suppression Control under New Control Theory

To improve strip thickness accuracy in the cold tandem mill, it is necessary to enhance the responsiveness of the drive speed control system of the mill main drive system. However, the mechanical torsion vibration frequency of the roll-spindle and motor-shaft system is in the range of about 10 Hz, and may cause shaft vibration as a result of recent trends toward smaller roll diameter and higher-speed rolling operation. And also, in a dc thyristor Leonard system, speed response is limited to about 10 rad/s because of the commutator's current increase rate limit. The solution to these problems is explained below.

(1) For the main drive system, a squirrel cage induction motor driven by a circulaing current cycloconverter was used. Through the use of a circulating current and a 12-pulse rectified current, torque ripple generated by the drive system was virtually eliminated. **Figured 15**, **16**, and **17** illustrate the features of this system.
(2) The speed and accuracy of control and control calculations have been enhanced. **Table 2** shows the conventional and new systems. To ensure a speed response of 60 rad/s, particular ingenuity was applied to reduce sampling time and increase the speed and accuracy of speed detection. In addition, consideration was given to motor shaft roll-eccentricity control.
(3) Mechanical distortion frequencies of the roll-spindle and motor-shaft systems were increased from the previous approximate 13 Hz to 21–

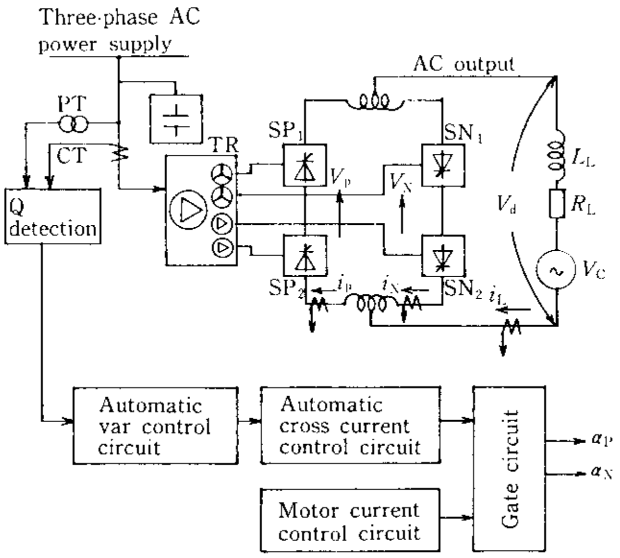


Fig. 15 System configuration (for one phase) of the cross current type cycloconverter

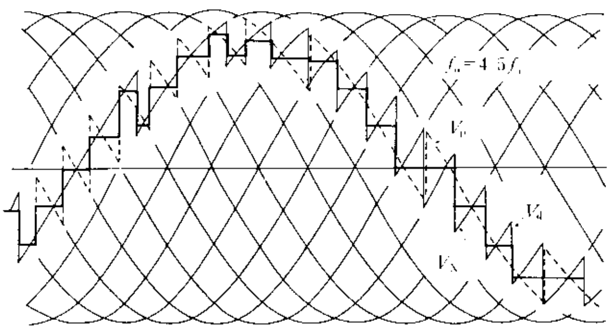
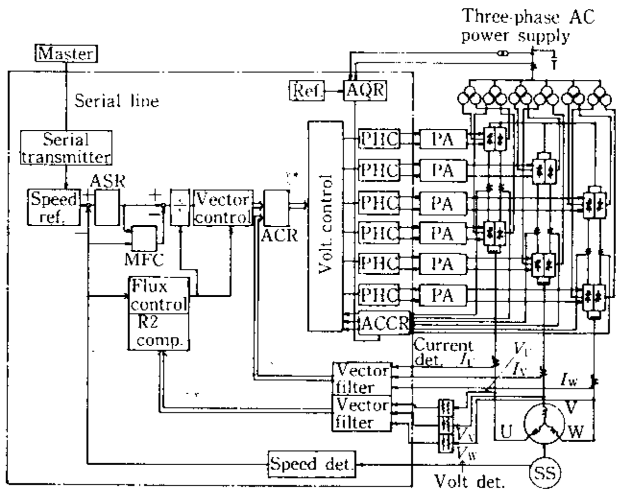


Fig. 16 Output voltage wave form of cycloconverter



MFC: Model following control
AQR: Automatic reactive power regulator
PHC: Phase control
ref.: reference
comp.: compensation
det.: detector
PA: Pulse amplifier
SS: Speed sensor (Resolver)

Fig. 17 Control block diagram of cross current type cycloconverter drive system

Table 2 Comparison of characteristics of digital vector control system

	Conventional	New vector control
Response		
ACR	100 rad/s	600 rad/s
ASR	30 rad/s	60 rad/s
Sampling time		
ACR	3 ms	1.0 ms
ASR	10 ms	4.0 ms
Speed detector (Pulse/revolution)	65 536 P/R	262 144 P/R

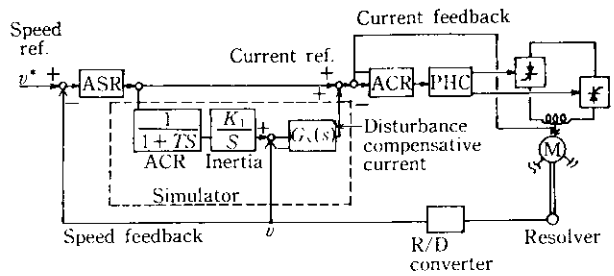


Fig. 18 Block diagram of MFC

22 Hz. Although roll diameters were reduced, consideration was given to the diameter and length of roll spindles and the gear layout to enhance the rigidity of the machine.

(4) A new control theory was introduced for shaft vibration suppression control.

To enhance the speed response of the speed control system to 60 rad/s, it was necessary to solve the problem of resonance at the drive shaft system mechanical torsion frequency of 22 Hz ≒ 138 rad/s. To solve this problem, Model Following Control (MFC) was introduced. MFC calculates ideal speed by simulation of a coupled model of the mechanical system, which has been considered to be rigid, and the motor control system, MFC compares this model simulation speed with the speed actually detected at the motor side and applies feedback to adjust the actually detected speed up to the simulated speed. This new control system is a powerful means of controlling the effects of external disturbances of motor speed such as load fluctuation and shaft and spindle resonance. A block diagram of this control method is shown in **Fig. 18**, and speed responses with and without

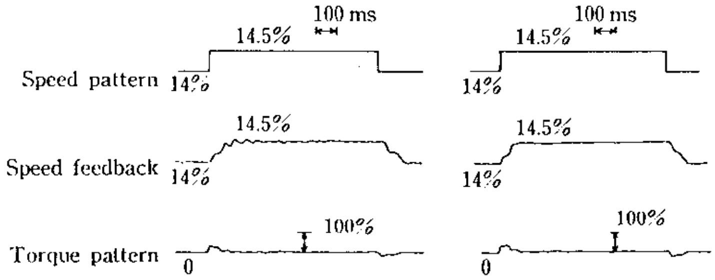


Fig. 19 Step response of speed

the MFC system are shown in **Fig. 19**. Speed feedback signal fluctuations are greatly reduced with the MFC system.

6 Conclusions

The following conclusions were reached in respect of shaft vibrations in the machine drive system for the steel rolling process through analysis and confirmation by tests using actual equipment:

(1) In the mechanical drive system, in which the response and accuracy of speed control are important, preliminary evaluation is necessary. A strict, comprehensive simulation analysis is required of shaft vibration phenomena in the mechanical system, control system, and rolling operation.

(2) Instead of considering the mechanical system as a rigid body, it is necessary to divide the mechanical system into mass points, where required, and to consider the respective torsional spring constants and damping coefficients of the mass points.

(3) Digital speed control should be subjected to simulation analyses taking into strict consideration the sampling hold. The shorter the sampling time and the more accurate and faster the speed detection, the stronger will be the control over shaft vibrations.

(4) As a shaft vibration suppression control method in the speed control system, application of a filter to the speed detection feedback system is effective in ensuring speed responsiveness and suppressing shaft vibrations.

(5) Shaft vibration suppression control by the model following control (MFC) system is effective. This method of control has been realized only through progress in theory and the development of microcomputers. It is important that there be good agreement between the model and the actual machine, and that the condition of the actual machine be accurately detected.

(6) When the mechanical torsion natural frequency falls within the actual-use rotational frequencies of the roll, spindle, and motor, flexural vibrations also occur, requiring particular attention. Excessive play also lowers rigidity equivalently.

Finally the authors would like to express their deep appreciation to the concerned staff of Hitachi, Ltd., Toshiba Corp., and Mitsubishi Electric Corp. for their valuable cooperation given in the course of the vibration analysis.

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