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Estimation of Impact Input

Torque in Rolling Mill Drive System when Material enters the rolls*

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The drive shaft system of a steel rolling mill usually vibrates torsionally when a material gets into the rolls. In designing a drive system, it is necessary to grasp the magnitude of transient torque. TAF (Torque Amplification Factor = maximum torque/steady-state torque) is usually applied for this purpose but it occasionally varies with each rolling process. This statistical TAF may depend on variable impact input torque.

This paper proposes the following three functions with the parameter β to describe the statistical input torque.

$$\begin{aligned} 1 - \exp(\omega_1 t / \beta) & \quad \beta < 0 \\ u(t) & \quad \beta = 0 \\ u(t) + \beta \delta(t) / \omega_1 & \quad \beta > 0 \end{aligned}$$

where $u(t)$: unit step function, $\delta(t)$: delta function, ω_1 : natural frequency of shaft. β may be determined based on the measurement in site and TAF is theoretically estimated from β and the above three functions.

According to the measurement of a practical hot strip mill system, mean and standard deviations of β are 0.79 ~ 0.86, and 0.89 ~ 1.04, respectively.

1. Introduction

In a steel rolling mill, large stresses are exerted on the drive shaft resulting from a torsional vibration excited in the system when a slab or billet is taken in by the rolls. Therefore, a study on the magnitude of transient torque is important in designing the drive shaft system of a rolling mill.

Petersen⁽¹⁾ reported on a torsional vibration in which the transient responses were computed with such variables as moment of inertia of motors and torsional stiffness of shafts, and discussed the maximum torques caused by torsional vibration in each shaft. Jewik et al.⁽²⁾ developed such a study to introduce a concept of Torque Amplification Factor (TAF). They also calculated the TAF in a drive shaft system with a backlash by the analog computer. Kashay et al.⁽³⁾, and Takeshita et al.⁽⁴⁾ carried out measurement of TAF on practical steel rolling mills and compared the results with the theory, in which the analytical model was composed of two sub-systems, i.e., mechanical spring-mass system and electrical drive control system. Honjyo et al.⁽⁵⁾

discussed, using a simple drive shaft system, on the factors which induce a much larger TAF than the theoretical one for step input torque. With these previous studies the analysis for prediction of torsional response and the TAF in a rolling mill drive shaft system may be thought to have been basically established.

On the other hand, it has been reported⁽⁶⁾ that when the torsional vibration response is measured at the moments of a slab entering a specific steel rolling mill, a considerable variation can be recognized in the TAF derived from that response. This variation may be attributed to the differences in shape of forefront of a given slab and/or relative speed between the rolls and the slab, i.e., such differences may be estimated to cause a statistical variation of impact input torque pattern which in turn affects the torsional vibration response. Therefore, a reliable design of a rolling mill drive shaft system can be attained only when both the method of analysis for torsional vibration system and the behavior of impact input torque are established. Data on the impact input torque pattern are not hitherto available. This may be due to the difficulty in the direct measurement of the input torque.

To cope with such difficulties, the TAF for various impact input torque patterns is analyzed on theoretical basis first, and then the impact torque pattern is estimated from the measured TAF. Since in order to design a successful rolling mill drive shaft system, it is necessary to have adequate knowledge on the impact input torque pattern, 120 measurements on

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a specific finishing hot strip mill are presented to reveal the statistical characteristics of the impact input torque pattern.

2. Modeling of impact input torque pattern

The impact input torque T is represented by three wave patterns shown in Figure 1. They can be also expressed as follows:

- (i) $T = T_0 u(t)$ at $\beta = 0$ (1)
 (ii) $T = T_0 \{u(t) + \beta \delta(t)/\omega_1\}$ at $\beta > 0$ (2)
 (iii) $T = T_0 \{1 - \exp(\omega_1 t/\beta)\}$ at $\beta < 0$ (3)

where, $u(t)$ and $\delta(t)$ are the step function and the delta function respectively, ω_1 is the typical angular natural frequency of the rolling mill drive shaft system, and T_0 is the steady-state torque. The parameter β is introduced to define the impact input torque pattern. The impact input torque expressed by the equation continuously varies from (iii) to (ii) through (i), while β changes from $-\infty$ to $+\infty$ through 0. This means that the impact input torque can be sequentially defined by determining β . Thus, the impact input torque pattern

Fig. 1 Time histories of impact input torque

of practical rolling is estimated from β by a torsional vibration study and therefore deviation of impact input torque can be interpreted as deviation of β .

3. Relation between TAF and parameter β in a simple drive shaft system

The relation between TAF and β in a simple one degree of freedom system is discussed in this section to provide essential knowledge which is necessary to study a more complicated rolling mill drive shaft system in actual use. The system is shown in Figure 2. TAF can be expressed in the following equation:

Fig. 2 Simplified model

$$\text{TAF} = \frac{T_{\max}}{T_0} = \frac{\text{Maximum torque}}{\text{Steady-state torque}} \quad \dots (4)$$

The responses to the impact input torques by equations (1) ~ (3) which are exerted on the left end roll shown in Figure 2 were calculated to obtain TAF. Then the results were summarized below, with details given in the Appendix.

- (1) For impact input torque (i)...($\beta=0$)
 TAF is determined from the following equations.

$$\text{TAF} = (\text{TAF})_0 = T_{\max}/T_0 = 2\gamma \quad \dots (5)$$

$$\gamma = I_M/(I_M + I_R) \quad \dots (6)$$

where, I_M and I_R represent the moments of inertia of motor armature, and rolls respectively, and $T_{\max}$ represents the maximum shaft torque

- (2) For impact input torque (ii)...($\beta>0$)
 TAF is determined in the same manner as above from the following equation.

$$\text{TAF} = \gamma(1 + \sqrt{1 + \beta^2}) \quad \dots (7)$$

with reference to $(\text{TAF})_0$ of equation(5),

$$(\text{TAF})_L = \text{TAF}/(\text{TAF})_0 = (1 + \sqrt{1 + \beta^2})/2 \quad \dots (8)$$

- (3) For impact input torque (iii)...($\beta<0$)

$$\text{TAF} = \gamma(1 + 1/\sqrt{1 + \beta^2}) \quad \dots (9)$$

$$(\text{TAF})_L = (1 + 1/\sqrt{1 + \beta^2})/2 \quad \dots (10)$$

These relationships are schematically summarized in Figure 3, where $(\text{TAF})_L$ is

Fig. 3 Diagram of $(\text{TAF})_L - \beta$ (One degree of freedom system)

plotted against β . The $(TAF)_L$ is continuous for $\beta=0$ and is a monotonic single-valued function with respect to β . The result shows that $(TAF)_L$ sharply increases in the positive domain of β . For example, $(TAF)_L$ is equal to 2.0 in the impact input torque pattern at $\beta=2.83$. This implies that 2.0 should be used as safety factor when the drive shaft system is designed with a simple step torque which corresponds to $\beta=0$.

The relationships between impact input torque pattern and TAF are schematically illustrated in Figure 4. The impact input torque characterized by the parameter β is translated into the response torque through the torsional vibration system. It is hardly possible to directly measure actual impact input torques. However, when the torsional vibration characteristics of a drive shaft system is carefully modeled on theoretical basis, the impact input torque can be estimated by measuring the response of the shaft torque.

Fig. 4 Relation between impact input torque and TAF

4. Parameter β in a rolling mill drive system

Deviation of impact input torque patterns is discussed in this section for a rolling mill drive system with a complicated configuration shown in Figure 5 (a). In this example, a finishing hot rolling mill with two DC motors of 3,750kw arranged in tandem is used.

4.1 Relation between TAF and β

The relation between TAF and β was first examined. The value of inertia and the torsional stiffness in the rolling mill drive system shown in Figure 5 (a)

are listed in Table 1. The drive system is modeled in a 6-degree of freedom system, as shown in Figure 5 (b). The rotational speed of the rolls in the rolling mill is normally controlled by the motor. A block diagram of the speed control system and the electrical specification of the motor in this example are given in Figure 6 and Table 2 respectively.

The impact input torques corresponding to the equations (1) ~ (3) were given to the left end rolls of the drive system in the same manner as in the previous section 3, and the transient response was calculated to obtain TAF for a range $-3 \leq \beta \leq 3$. As this system is an intricate multi-degree of freedom system with the control and the response cannot be obtained in an analytical manner, a digital computer-based block diagram simulator⁽⁷⁾ is employed. The dimensionless $(TAF)_L$ was derived with the reference $(TAF)_0$ for the simple step torque

Table 1 Moment of inertia and torsional stiffness (for Fig. 5)

Moment of inertia, kgf·m·s ²		Torsional stiffness kgf·m/rad	
I_1, I_5	49.6	K_1, K_5	$\times 10^6$ 1.905
I_2, I_6	2.5	K_2, K_6	1.713
I_3, I_7	3.1	K_3, K_7	3.774
I_p	20.2	K_4	11.77
I_8	49.4	K_8	18.88
I_9	49.4	K_9	15.84
I_{10}	1233.5	K_{10}	94.82
I_{11}	318.6	K_{11}	110.26
I_{12}	1911.0	K_{12}	45.25
I_{13}	1911.0	K_{13}	45.25
I_{14}	1911.0	—	—

Table 2 Main specifications of DC motor

Rated power	3750 kw/Motor
Revolutions per minute	115/300 rpm
Rated voltage	750 V
Rated current	5450 A/Motor
Rated torque	95.38 t.m/3 Motors

Fig. 5 (a) Schematic mill drive system showing major components and measurement locations

Fig. 5 (b) Model of spring-mass system

Fig. 6 Block diagram of control system

Fig. 7 Diagram of $(TAF)_L - \beta$ (for Fig. 5)

($\beta=0$), as adopted in section 3. Figure 7 shows the relationship between $(TAF)_L$ and β . The response of vibration is dependent on the observed point. In the example the calculation is shown at two different points, i.e., (1) floating shaft (hereinafter referred to as A shaft) and (2) motor output shaft (referred to as B shaft). The $(TAF)_L - \beta$ curve in the rolling mill drive system shows a trend similar to that in the simple drive system described in section 3. In the present example, the curve obtained from the A shaft is similar to that obtained from the simple drive system, while the curve obtained from the B shaft somewhat differs from that obtained from the simple drive system. In order to explain the trend, the transient response wave was examined in detail, and it should be noted here that the main vibration component in the A shaft consists of the first natural frequency, while the vibration component in the B shaft consists of both the first and the second natural frequencies.

4.2 Measurement of TAF and estimation of impact input torque pattern

The transient responses in the fin-

ishing hot rolling mill shown in Figure 5 were actually measured. Shaft torque due to torsional vibration was measured by applying strain gauges on the A and B shafts. The signals from the gauges are led to an external measuring device by an FM telemeter. An example of measured wave pattern is shown in Figure 8. The response wave sharply rises when the rolls bite the slab, and reaches the maximum value T_{max} , then the wave settles a steady-state value T_0 . The natural frequency of the torsional vibration is superimposed on the wave pattern. In this example, the first natural frequency of the system is 16.0 Hz. The previous observation is also demonstrated in Figure 8 where the first natural frequency of the system is dominant in the A shaft, while higher natural frequencies are remarkable in the B shaft.

TAF was determined from the maximum value T_{max} and the corresponding steady-state value T_0 and then $(TAF)_L$ was calculated in reference to $(TAF)_0$. The impact input torque pattern parameter β can be estimated from these calculation results

(a) A shaft $(TAF)_L = 1.15, \beta = 0.95$
B shaft $(TAF)_L = 1.26, \beta = 0.95$

(b) A shaft $(TAF)_L = 0.85, \beta = -0.95$
B shaft $(TAF)_L = 0.86, \beta = -0.99$

Fig. 8 Shaft torque of field test

and the relationship shown in Figure 7. The estimation should yield the same value whatever data on the system are used. β was estimated at 0.95 in both the A shaft and B shaft in Figure 8 (a), and at -0.95 and -0.99 in the A shaft and B shaft respectively in Figure 8 (b), thus a significant difference in β was not noted between the A shaft and the B shaft. Figure 8 shows a typical example of the positive and negative β .

4.3 Statistical values of impact input torque pattern

The impact input torque pattern parameters β are not always the same, but depend on each rolling process. A difference, or deviation in terms of statistics, can be generally caused by "hard-to-control factors" and "easy-to-control factors". The former include such variables as shape of forefront of given slab, uneven temperature distribution in slab and a slight relative speed difference between the rolling speed and the slab running speed. The latter include such variables as characteristics of slab itself (weight, material, etc.) and rolling conditions (reduction, for example). However, discussion is limited here only to the various statistic values obtained from 120 measurements carried out on a specific finishing hot rolling mill in use.

Figure 9 shows an example of histogram of $(TAF)_L$ obtained from the A shaft.

Fig. 9 Histogram of $(TAF)_L$ (A shaft)

It is noted that the $(TAF)_L$ is a statistical variable but does not always present a normal distribution.

The impact input torque pattern parameter β in the B shaft was estimated and its cumulative distribution was plotted. The results are shown in Figure 10. The data I is plotted from the 120 measurements. In this example, β evidently has a trend to present almost normal distribution.

○: data I, ×: data II
Fig. 10 Cumulative distribution function of β (B shaft)

The above results are summarized in Table 3, where the mean value and the standard deviation concerning TAF and β

Table 3 Value of β (Data I)

		A shaft	B shaft	Mean value of A and B Shaft
Field test TAF	Mean value	2.16	2.61	—
	Standard deviation	0.47	0.81	—
	Coefficient of variation	0.22	0.31	—
β	Mean value	0.79	0.86	0.83
	Standard deviation	0.89	1.04	0.97

are shown. The mean values of TAF are 2.16 and 2.61 in the A shaft and the B shaft respectively. The standard deviations are 0.47 and 0.81 in the A shaft and B shaft respectively. Since shock transmissibility of torsional vibration is a function of position, the comparison of mean values between the A shaft and B shaft is meaningless. As for coefficient of variation (standard deviation/mean value), it also differs with position on the shaft, i.e., 0.22 and 0.31 in the A

shaft and the B shaft respectively. The mean values of β , that is β_m , are 0.79 and 0.86 in the A shaft and the B shaft respectively, while the standard deviations β_s are 0.89 and 1.04 in the A shaft and the B shaft respectively. These statistical values for β are found to be almost the same in the A shaft and the B shaft, giving mean values $\beta_m = 0.83$ and $\beta_s = 0.97$. Strength calculation in designing a rolling mill drive system has been hitherto based on a step function corresponding to $\beta=0$ in most cases. The example shows that the design based on $\beta=0$ is not always reasonable.

4.4 Discussion

Among the statistic data taken from the 120 measurements, the data obtained under the known and specific rolling conditions are picked up and displayed as data II in Figure 10. In comparison of the data II with the data I, it is seen that the rolling conditions affect the mean value and the deviation of impact input torque pattern parameter. Statistic data obtained under the specific rolling condition are listed in Table 4. In this table, it should be noted that β_m ranges

Table 4 Value of β (Data II)

		A shaft	B shaft	Mean value of A and B shaft
Field test TAF	Mean value	2.39	3.06	—
	Standard deviation	0.41	0.67	—
	Coefficient of variation	0.17	0.22	—
β	Mean value	1.26	1.48	1.37
	Standard deviation	0.54	0.56	0.55

from 1.26 to 1.48, and β_s ranges from 0.54 to 0.56. This means that under the specific rolling condition, β might be a positive value i.e., $\beta>0$, in almost every rolling process. From these results, the following are deduced.

- (1) The relationship between the rolling condition and β can be established when β is measured systematically for each rolling condition.
- (2) When one sets the period to obtain the deviation of β in which all possible rolling conditions are included, the impact input torque characteristics in the rolling mill can be represented on statistic basis.

With these knowledges, the characteristics of the stress developed on the shaft system should be statistically grasped from the data of impact input torque, so as to estimate a reliable strength and the fatigue life of a rolling mill system.

5. Conclusions

The impact input torques exerted on a steel rolling mill system are statistically

discussed and the following results are obtained.

- (1) The impact input torque is defined by the following three functions with parameter β .

$$\begin{aligned} 1 - e^{-\omega_1 t / \beta} & \quad \beta < 0 \\ u(t) & \quad \beta = 0 \\ u(t) + \beta \delta(t) / \omega_1 & \quad \beta > 0 \end{aligned}$$

where, $u(t)$: step function, $\delta(t)$: delta function and ω_1 : typical angular natural frequency.

- (2) From the data actually measured on a finishing hot rolling mill in use, a statistical value of β is derived. The range of its mean values is from 0.79 to 0.86, and that of standard deviations is from 0.89 to 1.04.

The measurement is carried out under the limited conditions, but it is believed that the statistic data of impact input torque will be helpful in designing a steel rolling mill drive system.

APPENDIX

In the one-degree of freedom system of torsional vibration shown in Figure 2, TAF can be obtained from the transient response analytically for the three impact input torques expressed by the equations (1) ~ (3). Let the impact input torque given to roll be T and disregard the motor output torque because of its large time constant in rising, then the following differential equation of motion can be obtained.

$$\begin{aligned} IR \ddot{\theta}_1 + K(\theta_1 - \theta_2) &= T \\ IM \ddot{\theta}_2 + K(\theta_2 - \theta_1) &= 0 \end{aligned} \quad \dots\dots\dots (11)$$

where, IR and IM represent the moment of inertia of rolls and that of motor armature respectively, K represents the torsional stiffness, θ_1 and θ_2 represent the angle of torsion of roll and that of motor armature.

- (1) For impact input torque(i)...($\beta=0$)
By substituting equation (1) for T in equation (11), shaft torque T_s can be expressed as follows:

$$\begin{aligned} T_s &= K(\theta_1 - \theta_2) \\ &= \gamma T_0 (1 - \cos \omega_1 t) \end{aligned}$$

where, ω_1 is the angular natural frequency in a given system:

$$\sqrt{K(IM + IR) / IM IR}$$

The maximum values of T_s which is referred to $T_{\max}$ and TAF are given by

$$\begin{aligned} T_{\max} &= 2\gamma T_0 \\ \text{TAF} &= T_{\max} / T_0 = 2\gamma \dots\dots\dots (12) \\ \gamma &= IM / (IM + IR) \end{aligned}$$

The TAF in this case is referred to as $(\text{TAF})_0$, and used as reference value for the subsequent calculation.

- (2) For impact input torque(ii)...($\beta > 0$)
By applying equation (2) to equation (11), the following can be introduced.

$$T_s = \gamma T_0 \{ 1 + \sqrt{1 + \beta^2} \sin(\omega_1 t + \psi) \}$$

where, $\tan \psi = -1/\beta$
TAF can be determined from the following equation derived in the same process in paragraph (1).

$$TAF = \gamma \{ 1 + \sqrt{1 + \beta^2} \} \dots\dots\dots(13)$$

- (3) For impact input torque(iii)...($\beta < 0$)
By applying equation (3) to equation (11), the following formula can be introduced.

$$T_s = \gamma T_0 \left\{ 1 - \frac{\beta^2}{\beta^2 + 1} e^{\omega_1 t / \beta} + \frac{1}{\sqrt{\beta^2 + 1}} \sin(\omega_1 t + \psi) \right\}$$

where, $\tan \psi = -1/\beta$

$$TAF = \gamma \left(1 + \frac{1}{\sqrt{1 + \beta^2}} \right) \dots\dots\dots(14)$$

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