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On a Method to Prevent Chatter in Boring Operations*
(Manufacture and Test of New Anti-Chatter Boring Bar)

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Various kinds of boring bars to prevent chatter in boring operations have been developed, but they are not successful enough. This paper deals with a new boring bar which is composed of a body on which a tool is mounted and an internal plate fitted in a slit of the body. Its structure and dimensions are determined, based on the theoretical analysis in order to achieve the highest performance, and the stability limits of the new boring bar are compared experimentally with those of the conventional bar with the circular solid section of the same diameter and length to diameter ratio. The experimental results show that the new bar can perform stable boring operations in a wide range of cutting speeds.

1. Introduction

In boring operations chatter occurs very often. This is because boring operations usually are practiced using bars which have a length to diameter ratio l/d more than 3 to 1, or in an extreme case more than 10 to 1, and as a result of it the stiffness of the bar becomes small. In order to prevent chatter, the boring bar with a Lanchester damper⁽¹⁾ and one which has different mode directions of the bar^{(2),(3)} have been developed. Unfortunately, those bars are not improved sufficiently on the performance in practical use.

It is known that chatter or self-excited vibration results from a vibration of a tool/work in the direction of the thrust force⁽⁴⁾. Therefore, in order to prevent chatter the vibration in this direction must be suppressed at first. The stiffness of the bar, however, can not be increased by increasing the diameter, once the size of a hole to be bored is given. Hence the damping must be improved by some means.

From this point of view, this paper deals with a new boring bar which is constructed to improve damping performance effectively. First the dynamic performance of the bar was theoretically analysed, then the optimum shape and dimensions of the bar were designed and the bar was made. At the same time a conventional bar with the circular solid section which had the same diameter and ratio l/d as the bar of

the new type was made, and their dynamic performances were compared with each other experimentally.

2. Design and analysis

A bar of a new type, which is called N-bar and one of a conventional type called S-bar hereafter, are shown in Fig.1. A body which is provided with a slit and an internal plate which has the same thickness as the width of the slit and is fitted into the slit, are clamped together at the fixing side. Consequently each of them can deflect independently in the direction of the slit (x direction in

Fig.1 Boring bars tested

Fig.2 Tool setting angle

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Fig.2), while they deflect together in the direction normal to the slit (y direction in Fig.2). The bar near its free end is constructed as shown in the section A-A. The internal plate is provided with a recess, where a spring and a disk are put in and the disk pressed by a set screw against the spring. By adjusting the set screw, the damping force between the body and the internal plate can be varied. This design causes reduction of the static stiffness, but if the damping effect is increased sufficiently, the dynamic stiffness may be able to increase enough to expect a higher dynamic performance.

As is evident from studies by Kuchma⁽²⁾ and Kato et al.⁽³⁾, the dynamic performance of the bar which has the directional mode characteristics depends on a tool setting angle. Since the N-bar has the directional mode characteristics, this is designed to be tested for changing the tool setting angle θ (refer to Fig.2). At the free end of the body, a head is fixed, on which a tool is clamped. That is, the head has a boss which is fitted into a hole of the body and fixed by a set screw. And the head has also holes into which a pin of the body is fitted to resist the torque caused by the cutting force. By means of the holes and the pin, θ can be changed from 0° to 180° with an increment of 30° . In this study, for the sake of convenience to compare the dynamic performances easily, both bars were tested with an l/d of 6 to 1 aiming at easy occurrence of chatter.

Taking x, y axes of rectangular coordinates and θ as shown in Fig.2 which is seen from the free end of the bar, vibration models of the N-bar are represented by Fig.3 (a) for x direction and Fig.3 (b) for y direction. That of the S-bar is represented also by Fig.3 (b) for both directions. In the figures, m_1, m_2 are the equivalent masses of the body and internal plate respectively, and it is assumed that there exists a viscous damper C_{12} at their interfaces. In fact, this damper is a frictional one, but in order to simplify the analysis it is assumed as viscous.

Various theories for the analysis of chatter have been put forward, and they

are classified principally into two groups; the one group pertains to primary chatter or the theory which does not contain regenerative effect, and the other pertains to regenerative chatter or the theory which contains regenerative effect. In the theory of primary chatter, the chatter is attributed to the phase lag of the force to the relative displacement between a tool and a workpiece. Considering the fact that regenerative effect can be replaced by a concept of a phase lag, both kinds of chatter can be analysed consistently. Hence in order to anticipate the dynamic performance it is sufficient to analyse chatter only on the basis of the theory of primary chatter. Then assuming the case where the cutting edge is vibrating, the fluctuation of the cutting force F_t and thrust force F_n at a time t can be expressed by the displacements x, y from a point of equilibrium as

$$\begin{aligned} F_t &= K_t \{ x(t-h_1) \cos \theta + y(t-h_2) \sin \theta \} \\ F_n &= K_n \{ x(t-H_1) \cos \theta + y(t-H_2) \sin \theta \} \end{aligned} \quad (1)$$

where h_1, h_2 and H_1, H_2 are phase lags of the cutting force and thrust force respectively, and K_t, K_n are the coefficients of proportion representing the cutting force and thrust force per unit depth of cut respectively. The phase lags in the calculations of this kind have been assumed such that $h_1 = h_2$ and $H_1 = H_2$. But since the frequencies of vibrations of x, y differ from each other, h_1 for x and h_2 for y should differ. Thus the phase lags are distinguished as above.

Using F_t and F_n , the fluctuations of the forces F_x and F_y in the directions of x and y respectively can be written from Fig.2,

$$\begin{aligned} F_x &= F_n \cos \theta - F_t \sin \theta \\ F_y &= F_n \sin \theta + F_t \cos \theta \end{aligned} \quad (2)$$

Substituting Eq.(1) into (2), F_x and F_y can be expressed by the displacements x and y which include a phase lag. Now it is assumed that $h_i, H_i (i=1,2)$ are sufficiently small, then the following approximations can be obtained, i.e.,

$$\begin{aligned} x(t-h_1) &\approx x - h_1 \dot{x}, y(t-h_2) \approx y - h_2 \dot{y} \\ x(t-H_1) &\approx x - H_1 \dot{x}, y(t-H_2) \approx y - H_2 \dot{y} \end{aligned} \quad (3)$$

As for equations of motion, because of the small mass and the low speed of revolution, the gyromoment, centrifugal force and Coriolis effect are all ignored. Therefore the equations of motion of the S-bar can be written,

$$\begin{aligned} m_0 \ddot{x} + C_0 \dot{x} + k_0 x + F_{x0} &= 0 \\ m_0 \ddot{y} + C_0 \dot{y} + k_0 y + F_{y0} &= 0 \end{aligned} \quad (4)$$

where m_0 is the equivalent mass, C_0 is the damping coefficient and k_0 is the spring constant. Since the S-bar has no directional mode characteristics, F_x and F_y are expressed by substituting $\theta = 0^\circ$ into Eqs. (1) and (2).

Likewise the equations of motion for the N-bar can be written

$$\begin{aligned} m_1 \ddot{x}_1 + C_{12}(\dot{x}_1 - \dot{x}_2) + C_1 \dot{x}_1 + k_1 x_1 + F_x &= 0 \\ m_2 \ddot{x}_2 + C_{12}(\dot{x}_2 - \dot{x}_1) + C_2 \dot{x}_2 + k_2 x_2 &= 0 \\ m_0 \ddot{y} + C_0 \dot{y} + k_0 y + F_y &= 0 \end{aligned} \quad (5)$$

where m_1, m_2 are the equivalent masses, C_1, C_2 and k_1, k_2 are damping coefficients and spring constants respectively which are shown in Fig.3, and the suffix 1 stands for the body and 2 for the internal plate. Also C_{12} is the damping coefficient between the body and the internal plate. In addition-

Fig.3 Vibration models

tion $m_0 = m_1 + m_2$, $C_0 = k_1 + k_2$ are the equivalent mass, damping coefficient and spring constant in the direction of y respectively, and it is assumed that m_0 , k_0 are the same as those of the S-bar. Equations (4) and (5) are rewritten as

$$\begin{cases} \ddot{x} + 2S_0 P_0 \dot{x} + P_0^2 x + f_{x0} = 0 \\ \ddot{y} + 2S_0 P_0 \dot{y} + P_0^2 y + f_{y0} = 0 \end{cases} \quad (6)$$

$$\begin{cases} \ddot{x}_1 + 2S_{12} P_1 (\dot{x}_1 - \dot{x}_2) + 2S_1 P_1 \dot{x}_1 + P_1^2 x_1 + f_{x1} = 0 \\ \ddot{x}_2 + 2S_{12} P_2 (\dot{x}_2 - \dot{x}_1) + 2S_2 P_2 \dot{x}_2 + P_2^2 x_2 + f_{x2} = 0 \\ \ddot{y} + 2S_{n0} P_0 \dot{y} + P_0^2 y + f_{y0} = 0 \end{cases} \quad (7)$$

where

$$\begin{aligned} C_0 &= 2S_0 P_0 m_0, \quad C_1 = 2S_1 P_1 m_1, \quad C_2 = 2S_2 P_2 m_2, \\ C_{12} &= 2S_{12} P_1 m_1, \quad C_{n0} = 2S_{n0} P_0 m_0, \quad \mu = m_1/m_2, \\ P_0^2 &= k_0/m_0, \quad P_1^2 = k_1/m_1, \quad P_2^2 = k_2/m_2, \quad f_{x0} = F_{x0}/m_0, \\ f_{y0} &= F_{y0}/m_0, \quad f_{x1} = F_{x1}/m_1, \quad f_{y1} = F_{y1}/m_1. \end{aligned}$$

Now nondimensional variables λ , β defined by the following equations are introduced,

$$\begin{cases} Kt = \lambda K_n \\ K_n = \beta k_0 \end{cases} \quad (8)$$

If a value of λ is given under certain conditions, β which is expressed by the following equations

$$\begin{cases} f_{x0} = f_{x0}(\beta), \quad f_{x1} = f_{x1}(\beta) \\ f_{y0} = f_{y0}(\beta), \quad f_{y1} = f_{y1}(\beta) \end{cases} \quad (9)$$

determines a stability limit. Here β is called cutting index.

A stability limit can be represented by the critical cutting index β_{sc} . In order to calculate β_{sc} for the S-bar, first a characteristic equation must be obtained by substitution of $x = X \exp(\xi t)$ and $y = Y \exp(\xi t)$ into Eq.(6). By means of an electric computer the critical β_{sc} was found for various β by calculating the Hurwitz equations which can be obtained from the characteristic equation. Likewise β_{nc} for stability limits of the N-bar can be found by substituting $x_1 = X_1 \exp(\xi t)$, $x_2 = X_2 \exp(\xi t)$ and $y = Y \exp(\xi t)$ into Eq.(7). From both β_{nc} and β_{sc} , γ is defined by

$$\gamma = \beta_{nc}/\beta_{sc} \quad (10)$$

with which the dynamic performances of both the N-bar and S-bar can be compared with each other. Here γ is called ratio of the dynamic performance.

To calculate γ , a value of thickness of the internal plate T must have been given. The value of T affects greatly the dynamic performance of the N-bar. In order to obtain the optimum T , γ was investigated for the change of T . In advance of this calculation, a phase lag must be assumed. For the purpose of distinguishing the dynamic performance between them, it is better to assume as large a value of the phase lag as possible, provided that the assumption of Eq.(3) holds. Consequently the phase lag for the thrust force, $\psi = H_i P_i$ ($i=0,1,2$, i have the same meaning hereafter) was taken as $\pi/9$. This procedure is more rational comparing with the one in which all the H_i were given the same value. Because, if the same value is given to each H_i , phase lag varies with the natural frequencies P_i and energy input into the vibratory system of the S-bar during a cycle differs from that of the N-bar, it is impossible to compare the dynamic performances of the bars under the same conditions. The natural frequencies P_i were calculated by means of the Rayleigh method, and the equivalent masses were obtained from $m_i = k_i/P_i^2$. The phase lag in the direction of the cutting

Fig.4 Performance ratio versus thickness of internal plate

Fig.5 Performance ratio versus tool setting angle

force was assumed such that $h_i = 0.8 H_i$. In the test, d was 30 mm and l/d was 6 to 1. Also it was assumed that λ was 3, the damping ratios ζ_0 , ζ_1 , ζ_2 were all 0.010 and S_{n0} was 0.015, while ζ_{12} was taken as a parameter and the dynamic performances were calculated for $\zeta_{12} = 0.20$, 0.40 and 0.60. The result for $\theta = 0^\circ$ is shown in Fig.4. As it is evident from this figure, there is an optimum thickness of the internal plate T . In the case where $\zeta_{12} = 0.60$ and $T = 13$ mm, the N-bar has dynamic performance 12.6 times as large as that of the S-bar. Even when ζ_{12} is 0.20, the N-bar has about 10.2 times dynamic performance at $T = 7$ mm. The larger the damping ratio ζ_{12} becomes, the higher maximum dynamic performance can be obtained and to the thicker side the optimum point of T moves.

As mentioned before, the dynamic performance depends also on the tool setting angle θ , then γ is calculated by varying θ for a different ζ_{12} in the case where T is 10 mm. The result is given in Fig.5, which shows that the maximums of the dynamic performance exist near $\theta = 0^\circ$ and 108° , and the larger ζ_{12} becomes, the higher dynamic performance can be obtained near 108° than near $\theta = 0^\circ$. For example, when ζ_{12} is 0.20, γ is 9.1 at $\theta = 0^\circ$ and $\gamma = 4.2$ at $\theta = 108^\circ$, hence

$\gamma_{\theta=0^\circ} > \gamma_{\theta=108^\circ}$. On the other hand when ζ_{12} is 0.60, $\gamma=11.6$ at $\theta=0^\circ$ and $\gamma=12.2$ at $\theta=108^\circ$, thus $\gamma_{\theta=0^\circ} < \gamma_{\theta=108^\circ}$. The dynamic performance of the N-bar is inferior to that of the S-bar between $\theta=30^\circ$ and 80° conversely.

Now phenomenon of the digging-in of a tool must be noted. The resultant cutting force for a given tool setting angle θ can be divided into R_x and R_y as shown in Fig.6. Since the cross section of the body of the N-bar has principal axes x, y , the deflections δ_x and δ_y occur in each direction of the axis. Let δ_{xn} and δ_{yn} be the projections of δ_x and δ_y onto the axis of n , and the direction shown in the figure be positive, then digging-in does not occur and cutting is stable statically, provided that the following relationship holds,

$$\delta_{xn} + \delta_{yn} \geq 0 \quad (11)$$

Now from the figure, R_x and R_y can be written

$$\begin{aligned} R_x &= R \cos(\theta + \phi) \\ R_y &= R \sin(\theta + \phi) \end{aligned} \quad (12)$$

Hence, considering $\delta_x = R_x/k_x$ and $\delta_y = R_y/k_y$ and using the spring constants aforementioned, δ_{xn} and δ_{yn} become

$$\begin{aligned} \delta_{xn} &= \frac{R}{k_x} \cos(\theta + \phi) \cos \theta \\ \delta_{yn} &= \frac{R}{k_y} \sin(\theta + \phi) \sin \theta \end{aligned} \quad (13)$$

From Eq.(11) and (13) statical stability limits within which digging-in does not occur, are given by

$$\frac{k_y}{k_x} \cos(\theta + \phi) \cos \theta + \sin(\theta + \phi) \sin \theta \geq 0 \quad (14)$$

When T is 10 mm, k_0/k_1 becomes 2.15. If it is assumed that λ is 3, ϕ becomes 71.6° , hence the range of the statical instability

Fig.6 Cutting forces and displacement of tool

Fig.7 Performance ratio versus thickness of internal plate

can be defined by Eq.(14) as $40^\circ \leq \theta \leq 70^\circ$. This range is also shown in Fig.5. Degradation of the dynamic performance in the range of θ between 10° and 90° may be attributed to this statical instability.

Since it is evident that there is a peak of the dynamic performance near $\theta=108^\circ$, next the performance ratio γ is calculated changing the value of T at this tool setting angle. The results are shown in Fig.7 for different ζ_{12} . Some different tendency can be perceived in this figure comparing with Fig.4. That is, the dynamic performance for $\theta=108^\circ$ falls rapidly on the side of the larger values of T . Although the maximum values of γ for $\theta=108^\circ$ are nearly equal to those for $\theta=0^\circ$, the maximum values for $\theta=108^\circ$ become slightly larger than those for $\theta=0^\circ$ for an increasing ζ_{12} . In Fig.5, γ for $\zeta_{12}=0.40$ takes the larger values at $\theta=0^\circ$ and 108° than those for the other ζ_{12} . Though this result seems to be contradictory to the results in Fig.4 and 7, it is not so, because Fig.5 is the limited case of $T=10$ mm. Hence if the dynamic performances in Fig.5 are compared with those in Fig.4 and 7 under the condition of the optimum T , γ increases for an increasing ζ_{12} and also γ at $\theta=108^\circ$ is higher than that at $\theta=0^\circ$.

Using the results of above calculations, a dimension of T must be decided. But actually it is not definite what value ζ_{12} takes. Thus ζ_{12} is assumed first to be 0.40 and T is decided to be 10 mm which is optimum for this ζ_{12} .

3. Experimental results

A horizontal milling machine was used for test, and vibration was detected by strain gauges. First the critical depths of cut of the S-bar and N-bar have been determined for $\theta=0^\circ$ and 120° against various cutting speeds. Next the critical depths of cut of the N-bar at the cutting speed of 60 m/min for various θ from 0° to 180° with 30° increment have been investigated. Work material used was Bsp3-1/4H, and material of solid tools was SKH4B. Tools had geometry of $(0^\circ, \alpha, 25^\circ, 10^\circ, 25^\circ, 25^\circ)$, where α was 0° and 20° . Feed was 0.10 mm/rev.

Damping ratios of the bars were measured with strain gauges which were struck near the fixed end of the bars. The case

Fig.8 Position of strain gauges

Table 1 Measured damping ratio

	ζ_0, ζ_{n0}	ζ_1	ζ_2	ζ_{12}
S-bar	0.015	—	—	—
N-bar	0.040	0.015	0.015	0.15

Fig.9 Stability limits

Fig.10 Stability limits

Fig.11 Stability limit

of the N-bar is shown in Fig.8. Free vibration caused by an impact at the free end of a bar was traced on the screen of an oscilloscope through a strain meter and photographed. Measured values are shown in Table 1. The value of ζ_{12} is a presumed one, because all the waves investigated for ζ_{12} were greatly distorted. This result shows that the measured values are in good accordance with the assumed values except ζ_{12} .

Experimental results for $\alpha=0^\circ$ and 20° are shown in Fig.9 and 10. As apparent from the figures, the higher the cutting speed becomes, the lower the limit of stability of the S-bar for both $\alpha=0^\circ$ and 20° becomes.

When the rake angle is 0° , the limit of the upper cutting speed for stability is about 70 m/min, while when α is 20° , it is about 90 m/min. In the ranges above those limits, chatter occurred at any depth of cut and machining was impossible. In the case of $\alpha=0^\circ$, there was also a lower limit of stability concerning depth of cut, and hence under any depth of cut below the limit, chatter again occurred. Consequently stable cut can be realized only at a depth of cut between the upper and lower limit of stability. On the other hand, when machined with the N-bar, there was no such a limit of stability concerning the cutting speed as the S-bar. Stable cutting could be achieved even in the range of high cutting speeds. Due to the limit of the machine capacity the maximum cutting speed tried was about 130 m/min, but no chatter occurred at this speed. When θ was 0° and α was 0° , the dynamic performance of the N-bar was inferior to that of the S-bar at low cutting speeds, but when θ was taken as 120° , the dynamic performance of the N-bar was sufficiently superior to that of the S-bar even in the range of low cutting speeds. In case where α is 20° , the N-bar shows better performance than the S-bar in all the range of cutting speeds irrespective to whether θ is 0° or 120° . Although the length of the cutting edge of the tool was limited to about 7 mm, and depth of cut could not be taken more than that, for $\theta=120^\circ$ chatter could not be perceived at any depth of cut.

Figure 11 which shows the stability limit against θ , indicates the same tendency as the theoretical calculation. That is, there are two peaks of the dynamic performance near $\theta=0^\circ$ and 120° , and also the value at $\theta=120^\circ$ is larger than that at $\theta=0^\circ$. Although the case where θ was 108° was not investigated because of the design of the bar, this experimental result shows that the theoretical analysis is correct at least qualitatively. Moreover the dynamic performance in case of θ existing between 10° and 100° is degraded and especially when θ is 30° and 60° , violent chatter occurs at any depth of cut. This agrees well with the statically unstable region which was deduced theoretically before ($40^\circ \leq \theta \leq 70^\circ$, refer to Fig.5).

4. Conclusion

It was found that the boring bar of the new type, as the theoretical analysis anticipated, has shown a good anti-chatter effect in all the range of cutting speeds tested, and also that there are two peaks in the dynamic performance at the tool setting angles $\theta=0^\circ$ and 120° .

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