



Research and exploration of sheet material bending machine

Qiulei Du^{1*} and Lin Jin²

¹Mechanics Engineer College of Changchun University, Jilin, China

²Jilin Teachers' Institute of Engineering & Technology, Jilin, China

ABSTRACT

Based on survey of existing bending machine and mechanism of sheet material bending processing, the paper researches design mechanism and principle of sheet material bending machine, mainly including design of overall structure, rear stopper and transmission part, and then carries out calculation and check of bending machine mechanical structure, including workbench intensity and connecting bolt check.

Key words: Sheet material, Bending machine, Structure, Stress, Workbench

INTRODUCTION

Sheet material bending machine is one of the most widely used sheet metal cold forming machine, which bends sheet metal to various components in different angles by simple general mold under cold condition. Along with the popularity of bending machine's application in various industries there were continuously higher requests for bending precision, productivity and automatic operation. Therefore, since the 60's, increasing new technology was widely used in hydraulic bending machine design and has rapidly improved bending machine technology level, making it a kind of precise metal forming machine [1].

In the mid 80's AMADA developed F/NE&BENDER precise downside-action bending machine, which adopts parallel pressurized series to make top and bottom beams deform uniformly under different load and bending length, and uses two or three sets of rollers to guide and improve partial load resistance. As the world's largest forging machine company, for more than 20 years, AMADA has been committed to developing and producing of downside-action bending machine, developed dedicated 5-axis robot and built sheet metal bending flexible manufacturing cell. The British company, Press & Shear, always manufactured upside-action bending machine, but developed E/GaForm downside-action bending machine series in 1991. All of the above show that downside-action bending machine has very strong vitality and a big market, and it will coexist with upside-action bending machine for a long time in the future. However, downside-action bending machine has its limitations. It is generally believed that downside-action bending machine is appropriate for the situation where pressure is lower than 1200 kN, working length is less than 3200 mm and stroke is short. Its is suitable for thin plate bending and components of small size and easy to support, where there is no need to place multiple sets of mold on workbench [2].

CONFIRMATION OF OVERALL STRUCTURE SCHEME

Based on hydraulic bending machine in the design, the technical specification is shown in table 1.

Table.1 Technical specification of bending machine

No.	Parameter	Value
1	nominal pressure	500kN
2	workbench length	2500mm
3	throat depth	400mm
4	slide stroke	100mm
5	column distance	2000mm
6	machine size	2600mm×1060mm×2030mm
7	descending speed of oil cylinder	80mm/s
8	pressure speed of oil cylinder	20mm/s
9	return stroke speed of oil cylinder	65mm/s
10	main motor power	5.5kw
11	weight	3550kg

According to the above design technical parameters, structure design is confirmed as follows.

(1) Overall design: This design uses whole cast steel structure, where the main components are separately cast and then welded. The advantages of the structure are high intensity, good rigidity and large motion space for rear stopper, where fuel tank and some other important components can be placed on the machine body. But there is a disadvantage that joining precision demands for each component are very high due to the joining structure.

(2) Rear stopper design: The position of rear stopper is presented in figure 1. There are two kinds of rear stoppers, namely hand-actuated and motor-driven adjusting types. The former is of simple structure and low cost, but it needs operator's high quality and a great deal of experience. What's more, when bending length changes, artificial adjustment for stopper is needed, and this will bring much difficulty especially when bending machine is in continuous working state. The latter is controlled by CNC, combined with manual fine tuning. Its structure is a little more complicated, but it is much easier for operators, who only need some simple training. This design adopts the latter rear stopper.

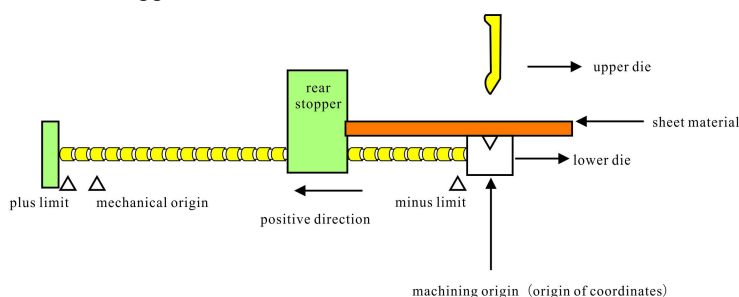


Fig.1 Rear stopper position

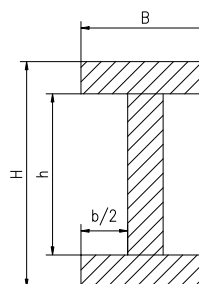


Fig.2 Workbench intensity check

(3) Transmission structure design: The design adopts double cylinder structure. The two cylinders' piston rod tops are connected with circular flange, and then walking beam is also connected to the flange through joint rod, which is equivalent that two cylinders directly drive walking beam, making the transmission more direct and efficient. But its oil system is complex, which makes higher requirement of electro-hydraulic synchronization technology.

CALCULATION OF MECHANICAL STRUCTURE

(1) Workbench intensity check is presented [3] in figure 2.

1) The formulas used are as follows:

$$I = \frac{BH^3 - bh^3}{12}$$

$$W_z = \frac{BH^3 - bh^3}{6H}$$

$y(x)$ Uniformly distributed $= ql^4(5-24\lambda^2)/384EIz$

$B=40\text{cm}$ $H=100\text{cm}$ $h=92\text{cm}$ $b=16\text{cm}$

Inertia moment:

$$I_z = \frac{BH^3 - bh^3}{12} = \frac{20 \times 92^3 - 16 \times 92^3}{12} = 628416 \text{ cm}^4$$

Uniformly distributed load:

$$q = \frac{F}{L} = \frac{500 \times 10^3}{250} = 200 \text{ N/mm}$$

F: Native nominal pressure $5 \times 10^5 \text{ N}$

L: Workbench total length 250 cm

$$\text{Modulus of anti-bending section: } W_z = \frac{BH^3 - bh^3}{6H} = \frac{20 \times 100^3 - 16 \times 92^3}{6 \times 100} = 12568.32 \text{ cm}^3$$

$$\lambda = m/l = 250/2000 = 0.175$$

Fulcrum counterforce:

$$R_A = R_B = ql(1+2\lambda)/2 = 200 \times 2000(1+2 \times 0.175)/2 = 2.5 \times 10^4 \text{ N}$$

$$\text{Bending moment: } M_A = M_B = -\frac{1}{2} qm^2 \quad 0 \leq x \leq m$$

$$M_A = -\frac{qx^2}{2} = -6.25 \times 10^6 \text{ N} \cdot \text{mm}$$

$$M_x = \frac{qL^2}{2} \left(\frac{1}{4} - \frac{m}{L} \right) \quad m < x < m+1$$

$$M = -\frac{q}{2} [x(x-L) + Lm]$$

$$= ql^2(1 - 4\lambda^2)/8$$

$$= 200 \times 250^2(1 - 4 \times 0.175^2)/8$$

$$= 1464843.75 \text{ N} \cdot \text{mm}$$

Shear stress:

$$\text{Section AC: } Q_A = -qx \quad 0 < x < m$$

$$Q_A = -200 \times 250 = -50000 \text{ N}$$

$$\text{Section AB: } Q = q \left(\frac{l}{2} - x \right) \quad m < x < m+1$$

$$\text{Point A: } \frac{qL}{2} = R_A = 2.5 \times 10^3 \text{ N}$$

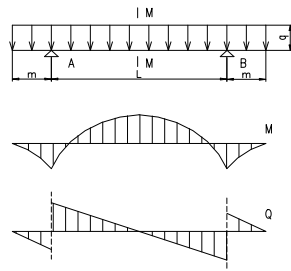


Fig.3 Stress diagram

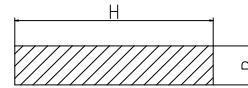


Fig.4 Slide diagram

2) Bending stress [4]:
It is shown in figure 3.

Bending stress in Section M-M maximizes.

$$\sigma_{m-m} = M_{\max} Wz = 1464843.75 \times 12568.32 = 1160.55 \text{ N/mm}^2$$

Material: 16Mn $\sigma_s = 2750 \text{ N/mm}^2$

$$[\sigma] = \sigma_s / n \quad n=2$$

$$= 2750 / 2$$

$$= 1375 \text{ N/mm}^2$$

$$\sigma_{m-m} < [\sigma]$$

Shear stress in Point A and B maximizes.

$$\tau_A = \tau_B = \frac{R_A}{S_A} = 2.5 \times 10^5 / S_A = 125 \text{ N/mm}^2$$

$$S_A = 120 \times 40 \text{ mm}$$

$$[\tau] = 0.6 \times 1375 = 825 \text{ N/mm}^2$$

$$\tau_A = \tau_B < [\tau]$$

∴ Security

3) Stiffness check: uniformly distributed load

$$y_{\max} = ql^4(5-24\lambda^2)/384EIz$$

$$= 200 \times 200^4(5-24 \times 0.175^2) / 384 \times 2 \times 10^6 \times 8220417$$

$$= 0.0111 \text{ cm}$$

$$y_{\max} \leq [y]$$

∴ License

(2) Slide is shown in figure 4.

$$B = 4 \text{ cm}, H = 250 \text{ cm}$$

Inertia moment: $I_z = \frac{BH^3}{12} = \frac{4 \times 250^3}{12} = 5.2 \times 10^6 \text{ cm}^4$

Modulus of anti-bending section: $W_z = \frac{BH^2}{6} = \frac{4 \times 250^2}{6} = 4.16 \times 10^4 \text{ cm}^3$

Fulcrum counterforce: $R_B = R_C = \frac{P}{2} = \frac{0.5 \times 10^6}{2} = 2.5 \times 10^4 \text{ N}$

Bending moment: AB : Section AB: $M = \frac{PX}{2} \quad 0 \leq x \leq \frac{l}{2}$

Section AC: $M = \frac{P(l - X)}{2} \quad \frac{l}{2} \leq x \leq l$

$M_{max} = \frac{pl}{4} = \frac{0.5 \times 10^6 \times 250}{4} = 3.125 \times 10^7 \text{ N} \cdot \text{mm}$

Stress diagram in figure 5.

Shear stress: Section AB: $Q_x = \frac{P}{2} = 2.5 \times 10^4 \text{ N}$

Section AC: $Q_x = -\frac{P}{2} = -2.5 \times 10^4 \text{ N}$

Bending stress: $\sigma_{max} = M_{max}/W_z = 3.125 \times 10^7 / 4.16 \times 10^4 = 751.2 \text{ N/mm}^2$

$n = \sigma_s / \sigma_{max} = 2750 / 751.2 = 3.67$

Shear stress in Point B and C maximizes.

$S_A = 4 \times 80.5 = 322 \text{ cm}^2$

$\tau_B = \tau_C = R_A / S_A$

$= 2.5 \times 10^5 / 322 = 776.4 \text{ N/mm}^2$

$[\tau] = 8250 \text{ N/mm}^2$

$\tau_B = \tau_C < [\tau]$

∴ Security

Stiffness check: concentrated $y_A = y_{max} = Pl^3 / 48EI_z$

$= 0.5 \times 10^6 \times 250^3 / 48 \times 2 \times 10^6 \times 6.3 \times 10^6 = 0.03 \text{ cm}$

$\lambda = 265 / 400 = 0.066$

Uniform $y_A = y_{max} = ql^4(5 - 24\lambda^2) / 384 EI_z$

$= 1250 \times 200^4(5 - 24 \times 0.066^2) / 384 \times 2 \times 10^6 \times 6.3 \times 10^6$

$= 0.015 \text{ cm}$

∴ License

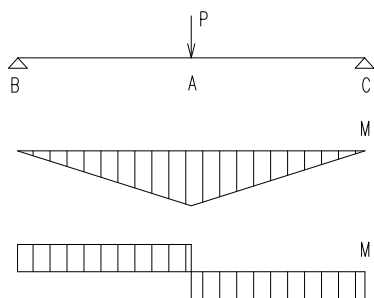


Fig.5 Stress diagram

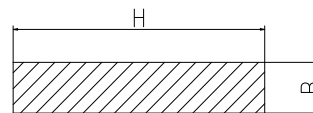


Fig.6 Wallboard diagram

(3) Wallboard is shown in figure 6.

$$H=48 \text{ cm}, B=4 \text{ cm} \quad P = \frac{F}{2} = 2.5 \times 10^5 \text{ N}$$

$$W = \frac{BH^2}{6} \quad B=4 \text{ cm}$$

$$\text{Inertia moment: } I_z = \frac{BH^3}{12} = \frac{4 \times 48^3}{12} = 36864 \text{ cm}^4$$

$$\text{Torque radius: } L = 22 + 48/2 = 46 \text{ cm}$$

$$\text{Modulus of anti-bending section: } W = \frac{4 \times 48^2}{6} = 1536 \text{ cm}^3$$

Wallboard dangerous section is on Section A-A, stress consists of tensile and bending stress.

$$S_{A-A} = BH = 4 \times 48 = 192 \text{ cm}^2$$

$$\sigma_{Tensile} = \frac{2.5 \times 10^5}{4 \times 48} = 13.02 \text{ N/mm}^2$$

$$\sigma_{Bending} = \frac{M}{W} = \frac{PL}{W} = \frac{250 \times 46}{1536} = 84.8 \text{ N/mm}^2$$

$$\sigma_{max} = 13.02 + 84.8 = 97.82 \text{ N/mm}^2$$

$$[\sigma] = 350 \text{ MPa}$$

$$[\sigma] > \sigma_{max}$$

Therefore wallboard is safe.

(4) Connecting bolt check

1) Each cylinder head is connected by bolts of 16-M16. After being checked: the weight of slide is $G_1=0.5T$, and thus the total weight is $G=1.3G_1=0.65T$.

Bolt tightening coefficient $K=1.5$

$$[\sigma] = \sigma_s / n = \sigma_s / 2.5 = 142 \text{ MPa}$$

According to bolt check formula:

$$d_s \geq \sqrt{\frac{4KF}{\pi Zm[\sigma]}}$$

It can be gained that $d_s \geq \sqrt{\frac{4 \times 1.5 \times 0.65 \times 10^4}{\pi \times 16 \times 2 \times 142}} = 2.7 < 36$

Thus intensity meets requirements.

2) Connecting bolt of vertical plate supporting workbench is 6-M24, under shear force, with bolt performance of Level 8.8, thus allowable stress is $\delta=640\text{MPa}$.

$$[\tau] = \delta/S_\tau = 256\text{MPa}$$

S_τ is security coefficient, assigned value of 2.5.

$$\tau = \frac{F}{An} = \frac{500000N}{6 \times \frac{\pi 24^2}{4}} = 184.3\text{MPa}$$

Where

F—— nominal pressure (N)

A—— bolt cross-sectional area (m²)

n—— bolt number

$$\tau = 184.3\text{MPa} \leq [\tau]$$

Thus requirements are met.

Additionally, that screwing length of bolt is greater than thickness of nut can ensure the stability. Thickness of M24 bolt nut is 22.6 mm~23.9 mm, screwing length should be longer than 24 mm.

3) Connecting bolt of cylinder connecting bed is 8-M20, under shear force, with bolt performance of Level 8.8, thus allowable stress is $\delta=640\text{MPa}$.

$$[\tau] = \delta/S_\tau = 256\text{MPa}$$

S_τ is security coefficient, assigned value of 2.5.

$$\tau = \frac{F}{An} = \frac{500000N}{8 \times \frac{\pi 20^2}{4}} = 199.04\text{MPa}$$

Where

F—— nominal pressure (N)

A—— bolt cross-sectional area (m²)

n—— bolt number

$$\tau = 199.04\text{MPa} \leq [\tau]$$

Thus requirements are met.

Additionally, that screwing length of bolt is greater than thickness of nut can ensure the stability. Thickness of M24 bolt nut is 19 mm~20.3 mm, screwing length should be longer than 20.3 mm.

CONCLUSION

The paper has carried on the detailed calculation and checking aiming at main components of bending machine. Through complicated, accurate calculation can bring great convenience to choosing components precisely so as to guarantee the machine running in good condition, avoid wear and tear caused by some certain problems, and thus increase machine's service life. Calculation and check aim at workbench, wallboard and connecting bolts. Bending machine in the design is of convenient mould replacement, accurate positioning rear stopper and easy adjusting front carrier. Users only need moulds in different shapes to bend them into required shapes. Further it adopts advanced and reliable electro-hydraulic synchronization technology, which is able to guarantee high synchronization accuracy even under the action of partial residual. It is ideal for sheet metal forming, can be widely used in aircraft, automobile, shipbuilding, electrical appliances, machinery and light industry, and the production efficiency is quite high.

REFERENCES

- [1] Yu X.L. *Hydraulic Machine*, Beijing, Mechanical Industry Press, **1982**; 148-149.
- [2] Fan H.C., et al. *Modern forging machinery*, Beijing, Mechanical Industry Press, **1994**; 188-199.
- [3] Liu H.W. *Mechanics of materials*, Beijing, Higher Education Press, **1979**; 27-144.
- [4] Nash W.A. *Mechanics of materials*, Beijing, Science Press, **2002**; 134-137.