Principle and Approach for Determining the Reasonable Cross-Section of the Solid-Web Double-beam Component

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Abstract: Solid-web double-beam component is widely used in the dipper handle of excavator, whose mechanical behaviour belongs to compression-bending member and mainly satisfies the strength requirement. In this paper, it starts from the design principles of steel structure and elastic theory, determines the dimension of the main cross-section and the reverse tube under guidance of optimization, and provides initial data for the detailed design. The method has common direct sense.

Foreword

Solid-web double-beam component is widely used in the dipper handle of mechanic excavators, whose mechanical behavior belongs to compression-bending member and mainly satisfies the strength requirement. In initial design, how to use its basic data, determine its reasonable cross-section dimension and provide initial data for the detailed design, designers are got into trouble. Thus, determine the reasonable dimension of the main cross-section from the design principles of steel structure under guidance of optimization and taking the example of excavator dipper handle.

To determine the cross-section of composite beam, the overall design requirement of components shall be comprehensively considered to meet the requirement of strength and economy. It is the issue of optimization.

1. Mechanical model

1.1 Requirements for strength

Where the ratio of length l and height h of a component $l/h > 5$, it can be regarded as a beam. According to strength requirement of beam,

$$\sigma = \frac{M}{W} + \frac{N}{A} \leq [\sigma] \tag{1}$$

1.2 Function expression of beam dead weight

Beams with wide universality in components are composite I beams and composite box beams. Excavator movable arm belongs to composite box beam, generally composed of flange plate, web plate and web stiffener. Gross mass of beam G is the summation of flange plate weight G_1, web plate weight G_2, web stiffener weight G_3 and connecting weld joint weight G_4.

$$G = G_1 + G_2 + G_3 + G_4 = G_1 + n_1 G_2$$
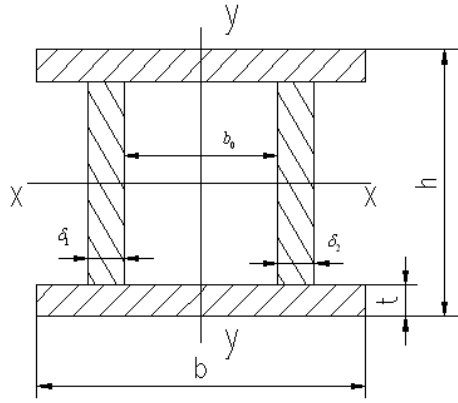$$n_1 = (G_2 + G_3 + G_4)/G_2, \quad n_1 \geq 1 \tag{2}$$

Fig.1   Cross-section of box beam

$$G_1 = 2A_1 \cdot l \cdot r$$

$$G_2 = h \sum \delta_i \cdot l \cdot r$$

$$G = \left(2A_1 + n_1 h \sum_{i=1}^{m} \delta_i \right) l \cdot r \tag{3}$$

Wherein, A_1: Cross-sectional area of a flange plate

$\sum \delta_i$: Total thickness of web plate

n_1 : Structural coefficient of web member

l : Length of dipper handle, determined in population parameter design

r :Steel density, r =7.8 t/m3。

(1) Determine formula G_σ of beam dead weight according to strength requirement:

$$I_\sigma = \frac{Wh}{2} \approx 2A_1 \left(\frac{h}{2} \right)^2 + \frac{h^3 \sum \delta_i}{12} \tag{4}$$

$$A_1 = \frac{W}{h} - \frac{h \sum \delta_i}{6} \tag{5}$$

$$G_\sigma = \left(\frac{2W}{h} - \frac{h \sum \delta_i}{3} + n_1 h \sum \delta_i \right) l \cdot r = \left[\frac{2W}{h} + \left(n_1 - \frac{1}{3} \right) h \sum \delta_i \right] l \cdot r \tag{6}$$

To get   min G_σ, supposing $\dfrac{dG_\sigma}{dh} = 0$, that is

$$h_\sigma = k_\sigma \sqrt{\frac{W}{\sum \delta_i}} \tag{7}$$

$$k_\sigma = \sqrt{\frac{2}{n_1 - \dfrac{1}{3}}} = \frac{\sqrt{2}}{\sqrt{n_1 - \dfrac{1}{3}}}$$

From expression (7), $\sum \delta_i h_\sigma^2 = k_\sigma^2 W$,   thus

$$W = \frac{\sum \delta_i h_\sigma^2}{k_\sigma^2} \tag{8}$$

Dipper handle of mechanical excavator plays the role of stress and bending mainly, whose torsion is sustained by reverse tube. Thus,

$$\frac{N}{A} + \frac{M_x}{W_x} \leq [\sigma]$$

(9)

$$A = 2A_1 + \sum \delta_i h$$

Wherein, A: Effective area of cross section

N: Axial force acting on cross section

W_x: Flexural modulus of cross section x axis

M_x: x axis bending moment of cross section

From formula (5) and (8), determine functional relationship $A = f(h_\sigma, \delta_i)$. Through combination, get

$$A = \frac{2\sum \delta_i h_\sigma (3 + k_\sigma^2)}{3k_\sigma^2}$$

(10)

Make substitution (13) into (12) and (14),

$$\frac{N}{\dfrac{2\sum \delta_i h_\sigma (3 + k_\sigma^2)}{3k_\sigma^2}} + \frac{M}{\dfrac{\sum \delta_i h_\sigma^2}{k_\sigma^2}} \leq [\sigma]$$

(11)

Then cross section h_σ, determined by external load and basic parameters, can be obtained.

If $i = 2$, and $t = \delta_1 = \delta_2 = \delta$

$$h_\sigma = \frac{3k_\sigma^2 \cdot N \pm \sqrt{9k_\sigma^4 N^2 + 32\delta(3 + k_\sigma^2)^2 \cdot k_\sigma^2 \cdot [\sigma] \cdot M_x}}{8\delta(3 + k_\sigma^2)[\sigma]}$$

(12)

A_1 can be calculated by making substitution of h_σ into (8) and (5).

2.   Determine the reasonable cross section of hopper handle
2.1 Determine the center width between two hopper handles

Inside width of hopper handle is determined by the distance between hopper handle and hopper hinge point which exerts a tremendous effect on hopper intensity. Thus, the center width of two hopper handles can be determined by hopper hinge point distance.

Hinge point between hopper handle and hopper is located the rear wall of hopper. Thickness of rear wall is much smaller than width and length of rear wall, simplified as simply support overhanging beam structure of both ends. Normal component of force of load (Including soil counter-force) acts at hopper rear wall and is considered as uniformly distributed load, as shown in Fig 2.

$R_A = R_B = q(\frac{l}{2} + a)$, shearing force Q,

$0 \leq z \leq a, \quad Q = -qz$

$a \leq z \leq a + l, \quad Q = R_A - qz = q(\frac{l}{2} + a) - qz = \frac{ql}{2}\left(1 + \frac{2a}{l} - \frac{2z}{l}\right)$

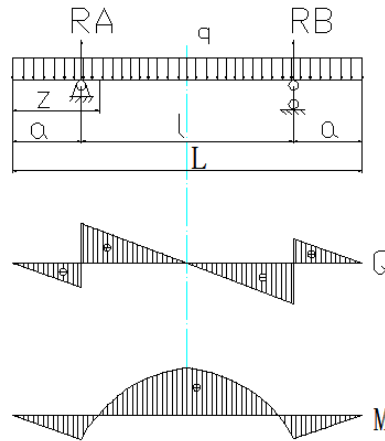$a + l < z \leq l + 2a, Q = R_A + R_B - qz = q(\frac{l}{2} + a) + q(\frac{l}{2} + a) - qz = q(l + 2a - z)$

Fig.2 Shearing force and bending moment

$0 \le z \le a$, $M_I = -\dfrac{qz^2}{2}$

$a \le z \le a+l$,

$$M_{II} = R_A(z-a) - \frac{1}{2}qz^2 = q(\frac{l}{2}+a)(z-a) - \frac{1}{2}qz^2 = \frac{ql^2}{2}\left[\left(1+\frac{2a}{l}\right)\left(\frac{z}{l}-\frac{a}{l}\right)-\frac{z^2}{l^2}\right]$$

To evaluate $M_{II\,Max}$

$$\frac{dM_{II}}{dz} = \frac{d}{d_z}\left\{\frac{ql^2}{2}\left[(1+\frac{2a}{l})(\frac{z}{l}-\frac{a}{l})-\frac{z^2}{l^2}\right]\right\} = 0, \quad \text{that is}$$

$$Z = \frac{l+2a}{2} = a + \frac{l}{2}$$

$$M_{II\,Max} = \frac{ql^2}{2}\left[\frac{1}{4}-\frac{a^2}{l^2}\right]$$

Provided that the beam is uniform cross section beam, then $M_1 = M_{II\,Max}$,

That is $\dfrac{qa^2}{2} = \dfrac{ql^2}{2}[\dfrac{1}{4}-\dfrac{a^2}{l^2}], l^2 = 8a^2, l = 2\sqrt{2}a$

Center distance between two hopper handles

$a = L/(2\sqrt{2}+2)$ (13)

Suppose $l+2a = 2\sqrt{2}a + 2a = L = 4.86m$ (Inside hopper width)

$a = 4.86/(2\sqrt{2}+2) = 1.0065 \approx 1m$ $l = 2.8469 \approx 2.847m$,

That is, center width between two hoppers is $2.847m$. Actual center width between two hoppers is $2.7m$。

Error: $\eta_1 = \dfrac{2.847\text{-}2.7}{2.7} = 5.4\%$;

2.2 Cross-sectional design of hopper handle

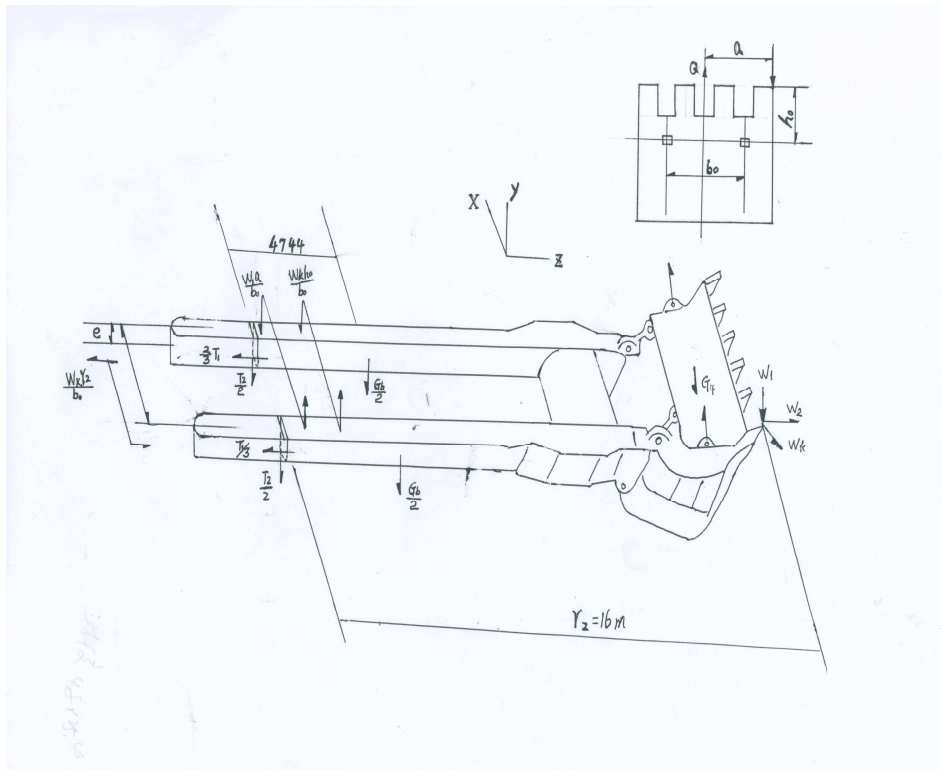
Fig.3:   Force diagram of hopper handle

Hopper handle design[1] includes structure form of hopper handle, hopper handle length, cross-sectional design and cross-sectional design of reverse tube.

Hopper handle is push-type gear rack, solid-web double-beam component, added with ribbed slab along its cross section direction, welded cylinder at front hopper handle next to hopper used for anti-torque.

(1)Assumed conditions

Assumed conditions include the following.

①Considering hopper body stiffness is much larger than hopper handle stiffness. Thus hopper handle shall be regarded as cantilever mechanic model[30];

②Suppose uneven two loaded handle hoppers. And suppose hopper handle at one side is sustained by 2T1/3 and another side T1/3.

③Take the hopper handle sustaining the larger load as computing object.

④For that handle hopper, σ_s =700Mpa;

⑤$[\sigma] = 0.85\sigma_s$, impact coefficient 2.5 and safety coefficient 1.2~1.5。

$$[\sigma] = \frac{700}{2.5 \times (1.2 \sim 1.5)} = 233 \sim 186 Mpa$$

(2)Known data

Tangential resistance W1=1325KN, normal resistance W2=1127 KN, b0=2692mm,a=2.615m; Crossrange excavating resistance Wk=113 KN, lifting force Q =2890KN, h0=4800mm; Pushing force T1=1127 KN; radial force T2=179 KN, r2=16m; Swing arm height of pushing axis: 2380mm; contact length of hopper handle and swing arm: 2380/sin45°=3366mm; Diagonal contact length: 4744mm;

Downward component forces of vertical hopper handle center line:

$$p_1 = \frac{w_1 a}{b_0} = \frac{1325 \times 2.615}{2.692} = 1287 KN,$$

$$p_2 = \frac{w_k h_0}{b_0} = \frac{113 \times 4.8}{2.692} = 202 KN,$$

$$p_3 = T_2/2 = 179/2 = 90 KN,$$

$$P = p_1 + p_2 + p_3 = 1579 KN$$

Bending moment M=PL, L is the distance between cross section and pushing axis.
Resultant forces along hopper handle center line:

$$N = \frac{2}{3} T_1 + \frac{w_k r_2}{b_0} = \frac{2}{3} \times 1127 + \frac{113 \times 16}{2.692} = 751.3 + 672 = 1423 KN$$

(3) Computed result of hopper handle economic height h_σ

$k^{[2,3]} = 1.52$, $k_\sigma^2 = 2.3104$, $k_\sigma^4 = 5.33794816$, $3k_\delta^2 = 6.9312$, $3 + k_\delta^2 = 5.3104$;

According to formula (12), substituting related data$^{[\sigma]}$, δ, h_σ can be got. Refer to table1:

Table 1   Computed result of hopper result h_σ :

	Computing position I : L=5m exterior cross section from pushing axis						
	[σ]=200Mpa			[σ]=185 Mpa			
	δ=25	δ=50	δ=75	δ=25	δ=38	δ=50	δ=75
Hopper	1.4	0.98	0.79	1.455	1.173	1	0.83
	Computing position II ; L=11.3m exterior cross section from pushing axis						
handle h_σ (m)	1.46	1.19	2.16	1.76	1.62	1.62	1.32
	Computing position III: L=10m exterior cross section from pushing axis						
	1.96	1.469	1.2	2.03	1.64	1.52	1.24

If$^{[\sigma]}$=200Mpa, δ=75mm, hopper handle root, position close to reverse tube, hopper handle height is 1.2m;

If$^{[\sigma]}$=185Mpa, δ=75mm, hopper handle root, hopper handle height is 1.24m; actual hopper handle height: h=1219mm.

(4)   Width of hopper handle
    Width of single hopper handle is determined by the width of pinion mate.

2.3 Calculation of reverse tube

    Reverse is caused when T1 and $w_k r_2 / b_0$ acting on meshing point between hopper handle and pushing gear is simplified on hopple arm central plane. Function of reverse tube is anti-torque. For easy calculation, regard reverse tube as rectangle uniform section.
 (1)   Calculating reverse tube side length by allowable shearing stress
    Shearing stress can be calculated as per the formula of membrane theory in elasticity mechanics [4].

$$\tau = \frac{M}{2A\delta}$$
(14)

   Wherein,
        M: torque
        M=1423KN×0.6m=8.538×108 N·mm;
        A: Average area value encircling cross section outside of reverse tube

    δ : take wall thickness of reverse tube as δ =10mm

    Considering $\tau = \frac{M}{2A\delta} \leq [\tau]$, then $A = \frac{M}{2[\tau]\delta}$; taking $[\tau]$=22Mpa, then

$$A = \frac{M}{2[\tau]\delta} = \frac{8.538 \times 10^8}{2 \times 22 \times 10} = 1.94 \times 10^6 \, mm^2$$

    If ratio of long and short sides is 2:1, then
    Approximate values of long and short side are 2000mm and1000mm respectively.
(2) Calculating torsion angle
    Torsion angle: $k = \frac{MS}{4A^2 G\delta}$
(15)
    Wherein, S: overall length of tube wall center line;
        G:torsional modulus, G=8.538×10^8 。

$$k = \frac{MS}{4A^2 G\delta} = \frac{8.538 \times 10^8 \times 5960}{4 \times 4 \times 10^{12} \times 8.1 \times 10^4 \times 10} = 4 \times 10^{-6} \ \text{degree}$$
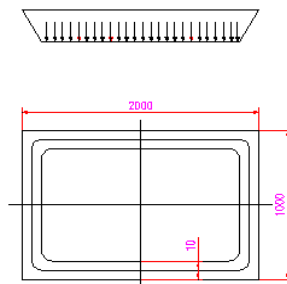

Fig.4 Membrane figure of thin-walled reverse tube

3. Conclusion

Arrive at the following conclusions from computed results:

(1) Center width of two hopper handles is 2.85m, 0.15m larger than the actual value;

(2) When $[\sigma]$=200Mpa, δ=75mm, hopper handle root and hopper handle height is 1.2m; actual hopper handle height is h=1.220m and error is －1.64%;

When $[\sigma]$=185Mpa, δ=75mm, hopper handle root and hopper handle height is 1.24m; actual hopper handle height is h=1.220m and error is +1.64%.

(3) Simplified mechanic model and computing method has guiding significance to initial design.

4. References

[1] S.W.Yan   Design of Mechanical Excavator (China machine Press, pp11-83, Beijing China 1991)

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