Research Article

Research of Crop Production Positioner Theory Design

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Abstract: Based on existing crop production positioner analytical investigation and according to working principle of crop production positioner, this study did a research of crop production positioner transmission design. The paper mainly designed worm and worm gear retarding mechanism of working table rotary direction. Aiming at mechanical structure of crop production positioner, related parts were calculated and checked, including checking calculation keyed shaft, selection and calculation of sliding bearing. Calculated results are of certain theoretical significance for crop production positioner design study.

Keywords: Crop production positioner, exploration, theoretical design

INTRODUCTION

Crop production positioner is auxiliary equipment, which is operation machine and it is called the three machines together with wheel frame and auxiliary equipment. Crop production positioner is to be mechanized and automated industry and development needs generated by rotating and tilting mechanism, workpiece placed in ideal position displacement machinery. Crop production positioner can be composed and operate machines and other supporting automatic machine; robot can be used as peripheral equipment supporting automation and robotics; it can also be used alone and independently. It is used as equipment, mainly used for lateral pipe butt welding, pipe and flange weld seam both inside and outside ring, all-position of pipes tubes. Crop production positioner horizontal flip angle, rotating and flipping through movement of workpiece table so that weld is in best position for welding, thus greatly improving weld quality, to reduce labor intensity welder, especially suitable for various shaft, disc type, such as rotating cylinder workpiece. And crop production positioner has a compact structure, small size, attractive appearance, light weight, etc., is a pressure vessel, metallurgy, electric power, chemical machinery, metal structures and other industries ideal equipment (Wang and Liu, 1995).

CROP PRODUCTION POSITIONER WORKING PRINCIPLE

Crop production positioner rotary table mechanism consists of work platform, slewing mechanism, tilt mechanism, base, electronic control devices, conductive devices and other components. It is driven by a DC motor, variable speed DC motor using SCR, to obtain a continuously adjustable rotary table speed. Its transmission routes are: DC motor;belt drive;first stage of worm rotation;second stage worm gear-gear, enabling rotary table. Table tilt mechanism AC brake motor, transmission line as: AC motor;belt drive;worm reducer;transition gear;worm reducer;gear, to achieve tilt table. Tilt shaft is equiped with limit switches, inclination angle strictly controlled between 0° and 90°, table and base unit to avoid collision.

This study did a research of crop production positioner transmission design. The paper mainly designed worm and worm gear retarding mechanism of working table rotary direction. Aiming at mechanical structure of crop production positioner, related parts were calculated and checked, including checking calculation keyed shaft, selection and calculation of sliding bearing. Calculated results are of certain theoretical significance for crop production positioner design study.

Crop production positioner drive deive:

• Direction of rotary table worm, worm gear transmission mechanism design (Pu and Ji, 2005):
• Selected worm, worm type, precision grade, material and number of teeth, head number
• Figure 1 shows crop production positioner structure (Yu and Zou, 2004), choice of involute worm (Z1) is transmission.
• Variable-bit machine designed for general operation of machine, speed is not high, so selection of 45 steel 8 Accuracy (GB/T 10085-1988).
Material selection, due to high efficiency of these hopes, wear better, so requirements of worm tooth surface hardening, hardness 45-55HRC. Turbine with a cast bronze ZC S 10P1, metal film casting.

According to tooth surface contact fatigue strength calculation

According to design criteria for a closed worm drive, press tooth surface contact fatigue strength design, verification tooth root bending fatigue strength. Calculated by calculation formula, which is shown as follows:

\[ a \geq \frac{\sqrt{KT_a(Z_{\rho}/Z_{\nu})^2}}{\sigma_h} \]

- Determine the formula for calculating value of each calculation

a. Selected test load factor \( K \).

Because work load is stable, they chose uneven load distribution coefficient \( K_\beta = 1 \); Select from Table 1 using coefficient \( K_d = 1.15 \); As speed and impact is not high, desirable dynamic load factor can be \( K_\nu = 1.05 \); then:

\[ K = K_aK_dK_\beta = 1.15 \times 1 \times 1.05 \approx 1.21 \]

b. Determine role of torque on worm \( T_2 \).

As \( Z_1 = 1 \), then the efficiency is \( \eta = 0.7 \).

\[ T_2 = 9.55 \times 10^6 \frac{P_a}{N_2} = 1000N \cdot mm \]

c. Confirmation of contact coefficient \( Z_{\nu} \).

First hypothesis and worm indexing circle diameter \( d_1 \) and ratio of transmission center distance \( a \) is:

\[ \frac{d_1}{a} = 0.35 \]

From where it can be gotten that.

d. Confirmation of elastic modulus influence coefficient \( Z_E \).

For selecting cast tin phosphor bronze worm gear and worm matching of steel.

Then:

\[ Z_E = 160MPa^{0.25} \]

e. Confirmation of allowable contact stress \( [\sigma_h] \).

According to the worm gear material for casting tin phosphor bronze ZC u S n 10P1, metal film casting, worm helical gear tooth surface hardness >45HRC,
Table 2: Cast bronze worm gear’s basic need contact stress $[\sigma_n](\text{MPa})$

<table>
<thead>
<tr>
<th>Worm gear material</th>
<th>Casting method</th>
<th>Working one side $[\sigma_n]$</th>
<th>Working both sides $[\sigma_n]$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cast Tin Bronze</td>
<td>Sand casting</td>
<td>40</td>
<td>29</td>
</tr>
<tr>
<td>ZCuSn10PI</td>
<td>Metal mould casting</td>
<td>56</td>
<td>40</td>
</tr>
</tbody>
</table>

Calculating transmission ratio is $i = \frac{z_2}{z_1} = 40$, then transmission ratio error $\frac{40-38}{38} = 0.05262 = 5.26\%$, is allowable:

- Worm gear dividing circle diameter is $d_2 = mz_2 = 5 \times 40 = 200$ mm
- Worm gear throat diameter is $d_{n_2} = d_2 - 2h_2 = 125$ mm
- Worm gear tooth root circle diameter is $d_{j_2} = d_2 - 2h_{j_2} = 185.5$ mm
- Worm gear generating circle throat radius is $r_{g_2} = a - \frac{1}{2} d_{a_2} = 33.55$ mm

- Check of tooth root bending fatigue strength:

  $\sigma_f = \frac{1.53KTJr}{d_jdm} \gamma_{1.2} \gamma_{0.6} \leq [\sigma_n]$ 

  Equivalent teeth number is:

  $z_2 = \frac{z_2}{\cos \gamma} = \frac{40}{\cos 11.31^\circ} = 50.398$

  According to $x_2 = -0.5$, $z_{c_2} = 43.398$, tooth form factor is $Y_{fa_2} = 2.48$.

  Spiral angle coefficient is:

  $Y_\beta = 1 - \frac{\gamma}{140} = 1 - \frac{11.31^\circ}{140} = 0.9197$

  Permissible bending stress $[\sigma_f] = [\sigma_f]_g \cdot K_{FN}$.

  According to Table 3, basic allowable bending stress of worm gear made of ZCuSn10PI is $[\sigma_f]_g = 40$ MPa:

  $K_{FN} = \sqrt{\frac{10^9}{6.3072 \times 10^{15}}} = 1.053$

  Life factor is

  $[\sigma_f] = [\sigma_f]_g \cdot K_{FN} = 40 \times 1.053 = 42.12$ MPa

  $\sigma_f = \frac{1.53 \times 1.05 \times 960.4}{22.4 \times 58 \times 2} \times 2.48 \times 0.7987 \times 42.12 = 1.176$ MPa

  Bending strength meets the requirements.

- Calculating efficiency $\eta$:
Table 4: Ordinary cylindrical worm transmission value of \( v_s, f_s \) and \( \varphi_s \\

<table>
<thead>
<tr>
<th>Worm gear ring material</th>
<th>Tin bronze</th>
<th>Tin-free bronzes</th>
<th>Gray pig iron</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \geq 45 \text{HRC} )</td>
<td>( f_s )</td>
<td>( \varphi_s )</td>
<td>( f_s )</td>
</tr>
</tbody>
</table>
| \( \geq 45 \text{HRC} \) | 0.10 | 0.12 | 0.14 | 0.18 | 0.10 | 0.18 | 0.19 | 0.10
| Others | 0.12 | 0.14 | 0.16 | 0.18 | 0.10 | 0.18 | 0.19 | 0.10

Table 5: Several kinds of commonly used material \([\varphi_s]\) of shaft:

<table>
<thead>
<tr>
<th>Shaft material</th>
<th>Q235-A, 20</th>
<th>Q275.35 (1Cr18Ni9Ti)</th>
<th>45</th>
<th>40Cr, 35SiMn, 38SiMnMo, 3Cr13</th>
</tr>
</thead>
<tbody>
<tr>
<td>([\varphi_s]/\text{MPa})</td>
<td>15~25</td>
<td>20~35</td>
<td>25~45</td>
<td>35~55</td>
</tr>
</tbody>
</table>

\[ \eta = (0.95 \sim 0.96) \frac{\tan \gamma}{\tan(\gamma + \varphi_s)} \]

It is known that \( \gamma = 28.18^\circ \); \( \varphi_s = \arctan f_s; f_s \) is related with \( v_s \).

\[ v_s = \frac{\pi d \eta}{60 \times 1000 \cos \gamma} = \frac{\pi \times 22.4 \times 3}{60 \times 1000 \cos 28.18^\circ} = 0.01 \text{m/s} \]

According to Table 4, \( f_s = 0.110 \), \( \varphi_s = 6^\circ \); substitute it into the equation, it can get \( \eta = 0.81 \), which is higher than original estimate and thus repetitive computation is not needed.

- **Confirmation of precision grade tolerance and surface roughness**: Considering design of power transmission, worm drive type belongs to general machinery gear reducer, to choose 8th precision accuracy magnitude from cylindrical worm and worm gear of GB / T10089-1988 type of lateral clearance is \( f \), which is labeled as 8 \( f \) of GB / T10089-1988. And then tolerance requires related manual check items and surface roughness and it is omitted here.

- **Check and calculation of belt key shaft**: Axis is under both torque and bending moment. 45 steel is selected as shaft material, quenched and tempered.

- **Check and calculation of shaft hardness**: According to torsional strength check and calculation, shaft torsional strength conditions should be:

\[ \tau_f = \frac{T_c}{W_{ji}} \leq [\tau_f] \]

where, \( T_c = T_1 \eta = 1764 \times 0.98 \text{N} \cdot \text{mm} = 1728.72 \text{mm} \), \( \eta \) is gear transmission efficiency.

\[ W_{ji} = \frac{\pi d^3}{16} \]

According to Table 5, \([\tau_f] = 35 \text{MPa}\) , substitute it into the equation:

\[ d \geq 89.994 \text{mm} \]

Minimum diameter of shaft is: \( d = 90 \text{mm} \), Select:

\[ d_{sb} = 170 \text{mm}, d_{bc} = 90 \text{mm}, d_{cd} = 100 \text{mm}, d_{de} = 90 \text{mm} \]

Check of section AB.

Where, \( T_{sb} = 1764 \text{N} \cdot \text{mm} = \frac{\pi d_{sb}^3}{16} \frac{bt(d_{sb}-t)^2}{2d_{sb}} \)

Checking and getting \( b = 3 \text{mm} \), \( t = 1.8 \text{mm} \), value \([\tau_f] = 25 \text{MPa}\). Numerical generation into above equation and getting:

\[ \tau_f = \frac{T_c}{\frac{\pi d_{sb}^3}{16} \frac{bt(d_{sb}-t)^2}{2d_{sb}}} = \frac{1764}{3 \times 1.8 \times (10-1.8)} 9.91 \text{MPa} \leq [\tau] \]

Shaft is obtained by selected diameter, which meets strength requirements.

- **Check and calculation according to shaft torsional rigidity condition**: Torsion deflection of shaft is presented by torsion angle \( \varphi \) per meter. Calculational formula of circular shaft torsional angle \( \varphi \) (with unit of \( ^\circ / \text{m} \)) is:

\[ \varphi = 5.73 \times 10^4 \frac{1}{LG} \sum_{i=1}^{n} \frac{T_{fi}}{I_{pi}} \]

Where, \( T_{sb} = T_{bc} = T_{cd} = 1764 \text{N} \cdot \text{mm} \), \( G = 8.1 \times 10^4 \text{MPa} \), \( L = 300 \text{mm} \), \( z = 1 \).

Shaft torsional rigidity condition is:

\[ \varphi \leq [\varphi] \]

According to related materials, value \([\varphi] = 0.5-1^\circ / \text{m} \)

It can be gotten: \( \varphi = 0.43^\circ \leq [\varphi] \).

- **Choice and calculation of plain bearings (Zeng and Hu, 2002; Feng, 2005).**
Table 6: Key connection allowable compressive stress and allowable stress

<table>
<thead>
<tr>
<th>Allowable compressive stress</th>
<th>Connection ways of working</th>
<th>Materials of key, hub or shaft</th>
<th>Static load</th>
<th>Slight shock</th>
<th>Shock</th>
</tr>
</thead>
<tbody>
<tr>
<td>$[\sigma_p]$</td>
<td>Static connection</td>
<td>Steel</td>
<td>120–150</td>
<td>100–120</td>
<td>60–90</td>
</tr>
<tr>
<td>$[P]$</td>
<td>Dynamic connection</td>
<td>Steel</td>
<td>70–80</td>
<td>50–60</td>
<td>30–45</td>
</tr>
</tbody>
</table>

The main structures of sliding bearing are integral radial bearings, split journal bearings and thrust sliding bearing. In the study, select monolithic neck sliding bearing sequence described above and actual situation.

Because bearing pedestal only bears radial force with no impact and speed is small, through force analysis $F = Mg(N)$ and there are two bearing and thus half of force in gravity, it can be gotten that:

$$f = \frac{1}{2} F = \frac{1}{2} \times 1000 \times 9.8 = 4900 \text{ N}$$

Choice and calculation of key:

Choice of key: Due to displacement of machine, key is only under shear force to choose ordinary flat key (type A). Key of both sides is working by key and keyway profile extrusion to transfer torque. Bond between surface and wheel hub keyway is on the underside of gaps. Flat key connection has a simple structure, convenient installation and advantages of better to neutral and widely used. But unable to bear axial force, thus shaft parts effect was less than the axial fixation.

- Calculation of key:
  - Connection strength calculation of flat key:
    Assuming load distributed evenly on working plate of key and connection strength calculation of common flat key is:

$$\sigma_p = \frac{2T \times 10^3}{kld} \leq [\sigma_p]$$

where,

- $T$ = Transfer torque ($T = F_y \approx F \frac{d}{2}$), N·m
- $K$ = Contact height between key and hub, $k = 0.5h$
- $L$ = Working length of key, mm, ball peen flat key $l = L - b$

where,

- $L$ = Nominal length of key, mm, $b$ is width of key, mm
- $d$ = Diameter of shaft, mm
- $[\sigma_p]$ = Allowable bearing stress of weakest material among key, shaft and hub, MPa, as is shown in Table 6

$[P] = $ Allowable pressure of weakest material among key, shaft and hub, MPa, as is shown in Table 6

Transfer torque is

$T = F_y \approx F \frac{d}{2} = 509 \times 10^3$ ; $k = 0.5 \times 14 = 7$; $l = 75 - 25 = 50$; $d = 90$ mm and thus

$$\sigma_p = \frac{2 \times 509 \times 10^3}{7 \times 50 \times 90} 32.317 \leq [\sigma_p] = 70 \sim 80$$

so it is up to standard.

CONCLUSION

Main design of transmission parts crop production positioner a detailed calculation and verification, although this is more complicated, it can be more precise and accurate calculation to select desired components to ensure machine running in good condition and avoid some problems that may be caused by wearing and tearing and can extend machine life. Part of calculation and verification includes checking calculation keyed shaft, selection and calculation of a plain bearing selection and calculation as well as the key. Design after crop production positioner movement than previous products with high accuracy, small inertia, good braking and stability, enabling variable speed, easy to realize advantages of reversing. The obtained results of calculation and design have some theoretical significance for crop production positioner design study.

REFERENCES