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Research Article

Research of Crop Production Positioner Theory Design

[1]Qiulei Du, [1]Liai Pan and [2]Zixuan Cheng
[1]Mechanics Engineer College of Changchun University, Changchun 130022,
[2]Institute of Automation, Chinese Academy of Sciences, Beijing, 100190, China

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, in order 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 (ZI) 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).

Corresponding Author: Qiulei Du, Mechanics Engineer College of Changchun University, Changchun 130022, China

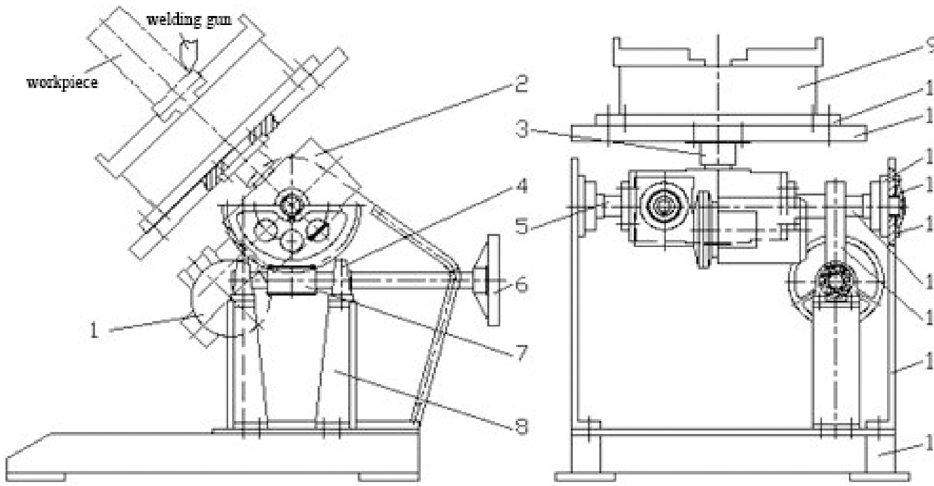

Fig. 1: Crop production positioner structure; 1: Motor; 2: Multi-stage worm gear reducer; 3: Spindle; 4: Bearing; 5: Coupling shaft; 6: Handwheel; 7: Worm; 8: Supports; 9: Jaw chuck; 10: The coupling plate; 11: Table; 12: Bearing; 13: Dial; 14: Fixed disk; 15: The coupling shaft; 16: Worm; 17: Side; 18: Base

Table 1: Tooth width coefficient of cylindrical gear K_A

Working type	I	II	III
Load properties	Uniform, no impact	Nonuniform, little impact	Nonuniform, great impact
Number of starts per hour	<25	25~50	>50
Start load	Low	Medium	High
K_A	1	1.15	1.2

- 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:

$$a \geq \sqrt[3]{KT_2(\frac{Z_E Z_\rho}{[\sigma_H]})^2}$$

- 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 $K_A = 1.15$; As speed and impact is not high, desirable dynamic load factor can be $K_V = 1.05$; then:

$$K = K_A K_V K_\beta = 1.15 \times 1 \times 1.05 \approx 1.21$$

b. Determine role of torque on worm T_2.

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

$$T_2 = 9.55 \times 10^6 \frac{p_2}{N_2} = 1000 N \bullet mm$$

c . Confirmation of contact coefficient Z_ρ.

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. Donfirmation of elastic modulus influence coefficient Z_E.

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

$$Z_E = 160 MPa^{\frac{1}{2}}$$

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 $[\sigma_H]'(MPa)$

Worm gear material	Casting method	Worm surface screw hardness	
		≤45*HRC*	>45*HRC*
Cast Tin Bronze	Sand casting	150	180
ZCuSn10PI	Metal mould casting	220	268

Table 3: Worm gear basic allowable bending stress $[\sigma_F]'$

Worm gear material	Casting method	Working one side $[\sigma_{0F}]'$	Working both sides $[\sigma_{-1F}]'$
Cast Tin Bronze	Sand casting	40	29
ZCuSn10PI	Metal mould casting	56	40

from Table 2 it can show basic allowable stress of worm gear $[\sigma_H]' = 268MP_a$.

f. Cycle number calculation of stress:

$$N_1 = 60n_1 jL_h = 60\times4\times1\times(2\times365\times24) = 4.2048\times10^6$$

$$N_2 = \frac{4.2048\times10^6}{3} = 1.4016\times10^6$$

g. Coefficient of contact fatigue life $K_{HN1} = 0.9$; $K_{HN2} = 0.95$。

h. Calculation of contact fatigue allowable stress

Failure probability is 1% and safety factor S = 1, then:

$$[\sigma_H]_1 = \frac{K_{HN1}\sigma_{\lim1}}{S} = 0.9\times600MPa = 540MPa$$

$$[\sigma_H]_2 = \frac{K_{HN1}\sigma_{\lim2}}{S} = 0.95\times550MPa = 522.5MPa$$

i. Calculation of center distance:

$$a \ge \sqrt[3]{KT_2(\frac{Z_E Z_\rho}{[\sigma_H]})^2} = \sqrt[3]{1.21\times1000(\frac{160\times2.9}{218})^2}\text{mm} = 131.724\text{mm}$$

Center distance is a = 140 mm and because manualspeed is 70 r/min, worktable rotate speed is 0.1~2, transmission ratio is i = 38, access to relevant information modulus is m = 5, worm indexing circle diameter is d_1 = 70 mm. There d_1/a = 0.5, access to relevant information contact coefficient is Z_ρ^1 = 2.56, because $Z_\rho^1 < Z_\rho$, the above calculation results are available:

- Main parameters and geometric dimensions of worm and worm gear
  - **Worm:** Axial pitch is P_a =15.70 mm; diametral quotient is q = 14; tip diameter is d_{a1} = 77 mm; root diameter is d_{f1} = 57 mm; lead-angle is $\gamma = 11^\circ 18' 36''$; worm axial tooth thickness is S_n = 12.5667 mm.
  - **Worm gear:** Worm gear teeth is Z_2 = 40 changed to coefficient X_2 = -0.5.

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 $d_{a2} = d_2+2h_{a2} - 125$ mm
Worm gear tooth root circle diameter is $d_{f2} = d_2 - 2h_{f2} = 185.5mm$
Worm gear generating circle throat radius is $r_{g2} = a - \frac{1}{2}d_{a2} = 33.5mm$

- Check of tooth root bending fatigue strength:

$$\sigma_F = \frac{1.53KT_5}{d_1 d_2 m} Y_{Fa2} Y_\beta \le [\sigma_F]$$

Equivalent teeth number is:

$$z_{v2} = \frac{z_2}{\cos^3\gamma} = \frac{40}{(\cos 11.31^\circ)^3} = 50.398$$

According to x_2 = -0.5, z_{v2} = 43.398, tooth form factor is Y_{Fa2} = 2.48.
Spiral angle coefficient is:

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

Permissible bending stress $[\sigma_F] = [\sigma_F]' \bullet K_{FN}$.

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

Life factor is $K_{FN} = \sqrt[9]{\frac{10^6}{6.3072\times10^5}} = 1.053$

$$[\sigma_F] = [\sigma_F]' \bullet K_{FN} = 40\times1.053MPa = 42.12MPa$$

$$\sigma_F = \frac{1.53\times1.05\times960.4}{22.4\times58\times2}\times2.48\times0.7987MPa = 1.176MPa$$

Bending strength meets the requirements.

- Calculating efficiency η:

Table 4: Ordinary cylindrical worm transmission value of v_s, f_v and φ_v

Worm gear ring material	Tin bronze				Tin-free bronzes		Gray pig iron			
Worm gear tooth surface hardness	≥45HRC		Others		≥45HRC		≥45HRC		Others	
Sliding speed v_s/(m/s)	f_V	φ_v	f_v	φ_v	f_v	φ_v	f_v	φ_v	f_v	φ_v
0.01	0.110	6º 17′	0.120	6º 51′	0.180	10º 12′	0.180	10º 12′	0.190	10º 45′
0.05	0.090	5º 0.9′	0.100	5º 49′	0.140	7º 58′	0.140	7º 58′	0.160	9º 0.5′
0.10	0.080	4º 34′	0.090	5º 0.9′	0.130	7º 24′	0.130	7º 24′	0.140	7º 58′
0.25	0.065	3º 43′	0.075	4º 17′	0.100	5º 43′	0.100	5º 43′	0.120	6º 51′
0.50	0.055	3º 0.9′	0.065	3º 43′	0.090	5º 0.9′	0.090	5º 0.9′	0.100	5º 43′

Table 5: Several kinds of commonly used material [τ_T] of shaft

Shaft material	Q235-A, 20	Q275, 35 (1Cr18Ni9Ti)	45	40Cr, 35SiMn, 38SiMnMo, 3Cr13
[τ$_T$]/*MPa*	15~25	20~35	25~45	35~55

$$\eta = (0.95 \sim 0.96)\frac{\tan\gamma}{\tan(\gamma+\varphi_v)}$$

It is known that γ = 28. 18°; φ_v = arctan f_v; f_v; is related with v_s.

$$v_s = \frac{\pi d_1 n_1}{60\times1000\cos\gamma} = \frac{\pi\times22.4\times3}{60\times1000\cos28.18^\circ} \approx 0.01\text{m}/s$$

According to Table 4, f_v = 0.110, φ_v = 6º7́; substitute it into the equation, it can get η = 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[2]:** 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:

$$\tau_T = \frac{T_2}{W_{T_2}} \le [\tau_T]$$

where, $T_2 = T_1\eta$ = 1764 × 0.98 N•mm = 1728.72 mm, η is gear transmission efficiency, $W_{T_2} = \frac{\pi d^3}{16}$.

According to Table 5, $[\tau_T] = 35MPa$, substitute it into the equation:

$$d \ge 89.994mm$$

Minimum diameter of shaft is: d = 90mm, Select:

$$d_{AB} = 170mm, d_{BC} = 90mm, d_{CD} = 100mm, d_{DE} = 90mm$$

Check of section AB.

where, $T_{AB} = T_2 = 1764N\bullet mm$, $W_{T_{AB}} = \frac{\pi d_{AB}^{\ 3}}{16} - \frac{bt(d_{AB}-t)^2}{2d_{AB}}$ checking and getting b = 3mm, t = 1.8mm, value [τ$_T$] = 25MPa. Numerical generation into above equation and getting:

$$\tau_T = \frac{T_2}{\frac{\pi d_{AB}^{\ 3}}{16} - \frac{bt(d_{AB}-t)^2}{2d_{AB}}} = \frac{1764}{\frac{\pi\bullet1000}{16} - \frac{3\times1.8\times(10-1.8)}{2\times10}} MPa = 9.91MPa \le [\tau]$$

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 φ per meter. Calculational formula of circular shaft torsional angle φ (with unit of (º)/m) is:

$$\varphi = 5.73\times10^4\frac{1}{LG}\sum_{i=1}^{z}\frac{T_i l_i}{I_{pi}}$$

where,

$T_{AB} = T_{BC} = T_{CD} = 1764N\bullet mm$, $G = 8.1\times10^4 MPa$, L = 300 mm, z = 1.

Shaft torsional rigidity condition is:

$$\varphi \le [\varphi]$$

According to related materials, value [φ] = 0.5-1(º) / m。

It can be gotten: $\varphi = 0.43^\circ \le [\varphi]$.

- Choice and calculation of plain bearings (Zeng and Hu, 2002; Feng, 2005).

Table 6: Key connection allowable compressive stress and allowable stress[5]

Allowable compressive stress	Connection ways of working	Materials of key, hub or shaft	Static load	Load property: Slight shock	Shock
$[\sigma_p]$	Static connection	Steel	120~150	100~120	60~90
		Cast iron	70~80	50~60	30~45
$[P]$	Dynamic connection	Steel	50	40	30

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:

$$\sigma_P = \frac{2T\times 10^3}{kld} \le [\sigma_P]$$

where,

T = Transfer torque $(T = Fy \approx F\frac{d}{2})$, N·m

K = Contact height between key and hub, $k = 0.5$h

L = Working length of key, mm, ball peen flat key $l = L - b$

where,

L = Nominal length of key, mm, b 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 $T = Fy \approx F\frac{d}{2} = 509\times 10^3$; $k = 0.5\times 14 = 7$; $l = 75 - 25 = 50$; $d = 90$ mm, and thus $\sigma_p = \frac{2\times 509\times 10^3}{7\times 50\times 90} 32.317 \le [\sigma_p] = 70 \sim 80$, 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.

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