Theory Research Design of Novel Automatic Wall Plastering Machine

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***Abstract*—Based on existing market research analysis about plastering machine, the paper presents innovative design for plastering machine according to some problems that currently exist. First, it does analysis of motor selection, which includes reducer motor and external motor; secondly it focuses on various components of gear unit to conduct detailed analysis, modeling, calculation and verification, including distribution and transmission ratio of speed check, reducer shaft calculation about rotating speed, power and torque, and worm gear rack design. The results and conclusions are consistent with theoretical design requirements, and the design theory can provide effective basis for new design of plastering machine.**

***Index Terms*— novel, plastering machine, theoretical research**

I. Introduction

Plastering machine is widely used in construction projects, such as walls, dado, corners etc., which is required for mortar, grout, stone mortar plastering work, can automatically plastering, automatic calendar, automatic lift, adjustable thickness, fast good quality, high efficiency, easy to use. So far, there does not appear fully mature product on market, some construction workers or through manual operation to achieve, both time-consuming and laborious, is a great waste of human resources. After investigation, plastering machinery on market in general consists of two parts: one is plastering machine and the other is equipment. Plastering mechanical transmission mode to use more hydraulic transmission, hydraulic multifunction plastering machine, plastering its rotary drive, up and down, horizontal movement and mortar pump drives are hydraulic. In addition to roof plastering machine driven rotary disc plastering device itself using electric motor-flexible shaft, the other drives are used hydraulic drive. Through nearly a decade of domestic published patent analysis related to plastering machine, plastering machines generally can be divided into two types: one is a handheld plastering machine, which is characterized by plastering device is not permanently installed on plaster machine, where workers plastering work hand control device. Device on plastering walls is along height direction in the width direction to move left and right. Plaster from wall thickness device and relative direction is entirely based on manual control of the operation of workers. The other is a mechanical plastering machine, and characteristics of machine is mounted on a post or the plaster device gantry machines, and post or mast is fixed to the chassis, entire machine to form a rigid whole. Plastering device by means of lifting frame up and down in column or gantry movement; plastering device can be divided into two types: one is a rotating disc plastering device, which by means of rotating disks from mortar plastering mortar tube pressed to wall, and be smooth. Such plastering device-driven approach are: motor-four more than four forms of plastering the device structure flexible shaft type, style motor oil, gas and compressed-air motor directly to the impeller type of work, etc. and they are too complicated. Another is pan-type plaster device which is movable by means of plaster board from wipe to mortar and mortar tube wall be smooth, there is no relative movement between plaster board and board member, where its structure is relatively simple.

By researching the above design, there are some shortcomings, and use of main problems existing in handheld plastering machine is still high labor intensity, can not play a role to reduce labor intensity, and the plaster is difficult to control quality in use. To solve these problems mechanical plastering machine is mostly used. But the presence of flatness and surface gloss to meet national requirements of problems existing in the plastering machine, especially two adjacent faces are not just wipe out ash on the same plane. Mechanical drive with hydraulic drive is easy to produce pulsation, plaster quality is difficult to control, which will also appear flatness and surface gloss to meet national requirements of problem. In order to solve these problems, this paper designs a new type of automatic plastering machine.

II. SELECTION OF MOTOR

A. *Selection of Motor reducer[1]*

(1) Selection of Motor type[2]

Depending on speed plastering devices currently on market, and in this estimate plastering machine take maximum operating speed $V = 0.3m/s$, selection of YCT series three-phase asynchronous motors.

(2) Selection of Motor power

According to actual situation, it is estimated to take plaster device 50Kg, 30Kg activity guide for motor and gear estimate 100Kg.

It is found that plaster device traction：

$$F_{\text{traction}} = (50+30+100)\times 9.8 = 1764N \tag{1}$$

The maximum operating speed： $V = 0.3m/s$ ， (2)

Overall efficiency of plastering machine

$$\eta_a = \eta_1 \cdot (\eta_2)^3 \cdot \eta_3 \cdot \eta_4 \quad (3)$$

(Where η_1 , η_2 , η_3 and η_4 respectively, couplings, worm drive bearings, worm gear, rack and pinion drive efficiency. Set $\eta_1 = 0.97$, $\eta_2 = 0.99$, $\eta_3 = 0.4$ and $\eta_4 = 0.96$)

Then $\eta = 0.97 \times 0.99^3 \times 0.4 \times 0.96 \approx 0.36$ (4)

Motor input power required is ：

$$P_d = \frac{F \cdot V}{1000\eta_\omega} = \frac{1764 \times 0.3}{1000 \times 0.35} kw = 1.5kw \quad (5)$$

(3) Selection of Motor

Due to consider making lightest weight reducer and motor, Y90S-2is selected, and main parameters of Y90S-2 are shown in “Table 1”.

TABLE I.

SELECTION OF MOTOR

Rated power (KW)	Full speed ($r \cdot \min^{-1}$)	Motor shaft diameter (mm)	Installation length of motor shaft extension end (mm)
1.5	2840	24	50

B. Selection of Outer Motor[3]

Take plastering device estimate traction ：

$$F_{\text{traction}} = 500N \quad (6)$$

Maximum operating speed： $v = 0.3m/s$

Effective power required by working machine：

$$P_w = \frac{Fv}{1000} = \frac{500 \times 0.3}{1000} = 0.15KW \quad (7)$$

Motor power required ：

$$P_d = P_w/\eta = 0.15/41.9\% = 0.36KW \quad (8)$$

Therefore selected motor is Y801-4, rated power of 0.55KW, and its parameters are shown in “Table 2”.

TABLE II.

SELECTION OF MOTOR

Rated power (KW)	Synchronous speed (r/min)	Full speed (r/min)	Motor shaft diameter (mm)
0.55	1500	1390	19

III. EACH COMPONENT DESIGN OF REDUCER

A. Ratio distribution and speed check[4]

Using a worm gear reducer and requires self-locking, when it is required self-locking $Z_1 = 1$,and number of teeth on worm gear should be more than 29, namely $i = \frac{Z_2}{Z_1} > 29$.

So speed of worm wheel axle is $n < \frac{2840}{29} = 97.4 r/\min$

And diameter of gear is

$$D > \frac{0.3 \times 1000 \times 60}{3.14 \times 97.4} \text{mm} = 58.9\text{mm} \quad (9)$$

Take gear pitch diameter of 63mm, modulus m = 6.3, and teeth of 31.

Then actual worm gear speed is

$$n = \frac{v \times 1000 \times 60}{\pi \cdot \text{D}} = \frac{0.3 \times 60 \times 10^3}{3.14 \times 63} \text{r/min} = 91\text{r/min} \quad (10)$$

and ratio is $i = \frac{2840}{91} = 31.04$ $i = 31$ (11)

Then actual speed of worm gear is

$$n_\omega = \frac{2840}{31} r/\min = 91.61 r/\min \quad (12)$$

Check speed error

speed error is $\Delta_{n_\omega} = \left|\frac{n - n_\omega}{n}\right| = \left|\frac{91 - 91.61}{91}\right| = 0.67\% < 5\%$, which meets the requirment.

B. Speed,power and torque calculation of resuder[5]

(1) speed of each shaft

worm shaft $n_{\text{I}} = 2840 r/\min$ (13)

worm wheel shaft $n_{\text{II}} = 91.61\text{r/min}$ (14)

(2) Input power of each shaft

worm shaft $P_{\text{I}} = P_{\text{d}} \cdot \eta_1 = 1.5 \times 0.97 = 1.46$ (15)

worm wheel shaft $P_{\Pi} = P_{\text{I}} \cdot \eta_2 \cdot \eta_3 = 1.46 \times 0.99 \times 0.4 = 0.58$ (16)

(3) Input torque of each shaft

Input torque calculation of motor shaft

$$T_d = 9550 \frac{P_d}{n_{满}} = 9550 \times \frac{1.5}{2840} = 5.04 N \cdot m \quad (17)$$

worm shaft $T_{\text{I}} = T_d \cdot \eta_1 = 5.04 \times 0.97 = 4.89 N \cdot m$ (18)

worm wheel shaft

$$T_{\Pi} = T_{\text{I}} \cdot i_1 \cdot \eta_2 \cdot \eta_3 = 4.89 \times 30 \times 0.99 \times 0.4 = 58.09 N \cdot m \quad (19)$$

Calculation results of movement and dynamic parameters are shown in “Table 3”.

TABLE III.

TRANSMISSION PARAMETERS[6]

Shaft / Parameter	Motor shaft	Worm shaft	Worm wheel shaft
Rotate speed n/（r/min）	2840	2840	91.61
Input power P/kW	1.5	1.46	0.58
Input Torque T/（N·M）	5.04	4.89	58.09
Ratio i	1		31
Efficiency η	0.97		0.397

C. Worm and worm wheel design

(1) material selection[7]

45 steel is chosen as worm tooth, where surface requires hardening and hardness 45-55HRC.

Worm wheel selects ZCuSn10P1, metal mold manufactured. In order to save material, gear ring selects bronze and wheel core HT100 gray cast iron.

(2) Design according to contact fatigue strength calculation[8]

Closed worm drive according to design calculation, press tooth surface contact fatigue strength calculation for design, and then proofread tooth root bending fatigue strength. Drive center distance

$$a \geq \sqrt[3]{KT(\frac{ZeZ\rho}{\sigma H})^2}$$

a. Confirm torque acting on worm wheel T

According to Z=1, estimate $\eta = 0.73$,

then

$$T = 9550\times1000\frac{P_2}{N_{\Pi}} = 9550\times1000\frac{P\cdot\eta}{N_1/\mathrm{i}} = 9550\times1000\times\frac{1.5\times0.73}{2840/31} = 114149.66N\cdot\mathrm{mm}$$

(20)

b. Confirm loading coefficient K

Relatively stable work, taking uneven load distribution coefficient as $K_\beta = 1.0$;

Select performance coefficient; $K_A = 1.0$ Because of slow working speed and little impact, desirable dynamic load can be $K_v = 1.05$;

then $K = K_\beta K_A K_v = 1.0\times1.0\times1.05 = 1.05$ (21)

c. Z_E Confirm elastic effect coefficient

Because choices are 45 steel worm and worm wheel with matching sake ZCuSn10P1, there is $Z_E = 160MPa^{\frac{1}{2}}$

d. $Z\rho$ Confirm contact coefficient

Assume ratio of worm pitch diameter d1 and center distance a of $d_1/a = 0.48$, it can be searched $Z_\rho = 2.32$

e. Confirm allowable contact stress $[\sigma H]$[9]

According to material choice of worm ZCuSn10P1, mold manufacturing, helical worm tooth surface hardness＞45HRC, Basic allowable stress of worm is

$[\sigma H]' = 268MPa$

Stress number of cycles $N = 60jn_\omega L_h = 60\times1\times91.61\times12000 = 6.60\times10^7$ (22)

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(22)

Life factor $K_{HN} = \sqrt[8]{\frac{10^7}{6.60\times10^7}} = 0.79$ (23)

then $[\sigma H] = K_{HN}\times[\sigma H]' = 0.79\times268MPa = 212MPa$ (24)

f. Center distance calculation

$$a \geq \sqrt[3]{1.05\times1.14\times10^5\times(\frac{160\times2.32}{212})^2} = 105.58mm \quad (25)$$

Because of self-locking, set a=160mm, from i=31, m=4, d1=71mm

it can be gotten $Z\rho' = 2.30$, as $Z\rho' < Z\rho$, above algorithm is effective.

(3) Main parameters and geometry of worm and worm wheel

a. worm

Axial foot distance $P_a = 12.56mm$ Diameter coefficient $q = 17.75$

Tip diameter $d_{a1} = 75mm$ Root diameter $d_{f1} = 57.4mm$

Indexing round lead angle $\gamma = 3^\circ13'28''$

Worm axial tooth thickness $s_a = 6.28mm$

b. worm wheel

worm gear number $z_2 = 31$, modification coefficient $x_2 = 0.125$

Check ratio i=$31/1 = 31$, ratio error is $\frac{31-30}{30} = 0.033$ =3.3%, which is allowable.

Worm pitch diameter

$$d_2 = mz_2 = 4\times31 = 124mm \quad (26)$$

Throat circle diameter

$$d_{a2} = d_2 + 2h_{a2} = 124 + 2\times2.5 = 129mm \quad (27)$$

Root diameter $d_{f2} = d_2 - 2h_{f2} = 124 - 2\times6.3 = 111.4mm$ (28)

Mother throat radius

$$r_{g2} = a - \frac{1}{2}d_{a2} = 160 - 0.5\times129 = 95.5mm \quad (29)$$

(4) Checking of tooth root bending fatigue strength[6]

$$\sigma_F = \frac{1.53KT_2}{d_1 d_2}Y_{Fa2}Y_\beta \leq [\sigma_F] \quad (30)$$

Equivalent teeth is $z_{v2} = \frac{z_2}{\cos^3\gamma} = \frac{31}{\cos^3 3.22^\circ} = 31.15$ (31)

According to $x_2 = 1.25, z_{v2} = 31.15$ toothed coefficient can be gotten that $Y_{Fa2} = 2.44$.

Helix angle factor is

$$Y_\beta = 1 - \frac{\gamma}{140^\circ} = 1 - \frac{3.22^\circ}{140^\circ} = 0.977 \quad (32)$$

allowable bending stress

according to some certain materials, basic promise worm gear bending stress made of ZCuSn10P1 is

$[\sigma_F]'$ =56Mpa.

Life factor is $K_{FN}=\sqrt[9]{\frac{10^6}{6.60\times10^7}}=0.63$ (33)

$[\sigma_F]=[\sigma_F]'\cdot K_{FN}=56\times0.63=35.28MPa$ (34)

$$\sigma_F=\frac{1.53\times1.05\times114149.66\times2.44\times0.977}{71\times124\times4}=12.42MPa \quad (35)$$

Bending strength is satisfied.

(5)Seek worm circumferential speed and check efficiency

$$\eta=(0.95-0.96)\frac{\tan\gamma}{\tan(\gamma+\varphi_v)} \quad (36)$$

As is know $\gamma=3.22^\circ$; $\varphi_v=\arctan f_v$; f_v is related to sliding speed v_s

$$v_s=\frac{\pi d_1 n_1}{60\times1000\cos\gamma}=\frac{3.14\times71\times2825}{60\times1000\cos3.22^\circ}=10.51m/s \quad (37)$$

From table with difference method：

$f_v=0.0279;\ \varphi_v=1.6971$ Substitute it into formula, it should be larger than original estimate. So there is no need of recalculation.

(6) Calculate main dimensions of worm drive

Transmission ratio i, head number of worm Z1 and worm gear number Z2

i=31 Z1=1 Z2=31

Worm lead angle γ $\tan\gamma=\frac{z_1}{q}=\frac{1}{q}\le\tan3.5^\circ$ (38)

It gets that $q\ge16.35$ (39)

Worm pitch diameter d1 and worm diameter factor q

Take $d_1=71$ $q=17.75$ $m=4$

Worm pitch diameter is $d_2=124mm$ (40)

Center distance

$$a=0.5(d_1+d_2)=0.5m(q+z_2)=0.5\times4\times(17.75+31)=97.5\text{mm} \quad (41)$$

Rounding 100mm

Modification coefficient x2

The main purpose of ordinary cylindrical worm gear displacement is equipped Minato center distance to conform to standard or recommended values. Displacement method with worm drives gearing and cutting, where tool is relative to worm gear shift[10]. When Minato center distance for worm gear modification coefficient x2

$$x_2=\frac{a}{m}-\frac{1}{2}(q+z_2)=\frac{a'-a}{m}=\frac{100-97.5}{4}=0.625 \quad (42)$$

Between 0.4 and 0.7 it meets requirements.

(7) Structure design of worm

Worm and shaft are integrally formed, namely worm shaft. Worm uses integrated casting.

(8)Heat balance check

preliminary estimate of cooling area A

$$A=0.33(\frac{a}{100})^{1.75}=0.33\times(\frac{200}{100})^{1.75}=1.11m^2 \quad (43)$$

Ambient air temperature t

Take t=20°C

Coefficient of thermal spreader K from $k=14\sim17.5$

Take $k=17W/m^2\cdot C$

heat balance check

According to

$$t_1=\frac{1000p_1(1-\eta)}{KA}+t=\frac{1000\times3.57\times(1-0.85)}{17\times0.923}+20=54.13^\circ C<85^\circ C \quad (44)$$

It should meet conditions.

D. Gear and pinion design

(1) Selection of materials

Select 40Cr (quenching) as rack, where hardness is 280 HBS , gear material 45 steel (quenching) with hardness of 240. There is 40 HBS difference between them. Accuracy class is 7-level precision. Gear number is $Z_1=17$.

(2) Calculation design according to tooth contact strength[11]

By spreadsheet, there is

$$d_{1t}\ge2.32\sqrt[3]{\frac{KT}{\phi d}\frac{u\pm1}{u}(\frac{Z_E}{[\sigma_H]})^2} \quad (45)$$

1) Determine calculated values

a. Test select $K=1.3$, tooth width coefficient is $\phi_d=0.8$

b. According some materials , elasticity coefficient is $Z_E=189.8MPa^{\frac{1}{2}}$

c. Contact pinion fatigue limit is $\sigma_{H\lim1}=600mPa$

Contact gear fatigue limit is $\sigma_{H\lim2}=550MPa$

d. Calculation of stress factor

$$N=60njL_n=60\times91.61\times12000=6.6\times10^7 \quad (46)$$

e. Take contact fatigue life factor $K_{HN1}=0.95$

f. Calculation of contact fatigue allowable stress

Take failure probability as 1%, safety factor is

$$[\sigma_H]_1=\frac{K_{hN1}\sigma\lim_1}{S}=0.95\times600=570MPa \quad (47)$$

$$[\sigma_H]_2=\frac{K_{KN2}K_{\lim2}}{S}=0.95\times550=523MPa \quad (48)$$

g.Computing gear transmission pitch diameter dlt, into smaller one of $[\sigma_H]$.

$$d_{1t}\ge2.32\sqrt[3]{\frac{KT}{\phi d}\frac{u\pm1}{u}(\frac{Z_E}{[\sigma_H]})^2}=2.32\sqrt[3]{\frac{1.3\times6\times10^4}{1}\cdot(\frac{189.8}{522.5})^2}mm=50.47mm$$

(49)

h. Calculating peripheral speed v.

$$v=\frac{\pi d_{lt}n}{60\times1000}=\frac{\pi\times50.47\times2825}{60000}m/s=7.5m/s \quad (50)$$

$$v=\frac{\pi d_{lt}n}{60\times1000}=\frac{\pi\times50.47\times2840}{60000}m/s=7.5m/s \quad (51)$$

i. Calculating tooth width b.

$$b=\phi_d\cdot d_{lt}=0.8\times50.47mm=40.37mm \quad (52)$$

j. Calculating tooth width and tooth height ratio $\frac{b}{h}$.

modules $m_t=\frac{d_{lt}}{z}=\frac{50.47}{17}mm=2.97mm$ (53)

tooth height

$$h=2.25m_t=2.25\times2.97\text{mm}=6.68\text{mm} \quad (54)$$

$$\frac{b}{h}=\frac{50.47}{6.68}=7.56$$

k. Calculating load factor.

According to $v=0.3m/s$, 7-level precision, dynamic load factor is $K_v=1.05$;

Spur gear, $K_{H\alpha}=K_{F\alpha}=1$;

utilization factor is $K_A=1$;

by interpolation method there is $K_{H\beta}=1.423$

From $\frac{b}{h}=7.56$, $K_{H\beta}=1.423$ it can be gotten $K_{F\beta}=1.35$;

then load factor is

$$K=K_AK_vK_{H\alpha}K_{H\beta}=1\times1.05\times1\times1.423=1.49 \tag{55}$$

l. According to actual load factor, correct calculated pitch diameter

$$d_1=d_{lt}\sqrt[3]{\frac{K}{K_t}}=50.47\times\sqrt[3]{\frac{1.49}{1.3}}=52.8 \tag{56}$$

m. Calculating modulus m.

$$m=\frac{d_1}{z}=\frac{52.8}{17}mm=3.1mm \tag{57}$$

2) Design according to tooth root bending strength

Design formula of bending strength is

$$m\geq\sqrt[3]{\frac{2KT}{\phi_d z^2}(\frac{Y_{Fa}Y_{Sa}}{[\sigma_F]})}$$

Confirm calculation value of each formula.

a. By looking up: strong bending fatigue limit of rack is $\sigma_{FE1}=500MPa$;

Strong bending fatigue limit of gear is $\sigma_{FE1}=380MPa$;

b. Take bending fatigue life factor $K_{FN1}=0.95$, $K_{FN2}=0.98$;

c. Calculating bending fatigue allowable stress.

Bending fatigue safety factor is S = 1.4.

$$[\sigma_F]_1=\frac{K_{FN1}\sigma_{FE1}}{S}=\frac{0.95\times500}{1.4}MPa=339.3MPa \tag{58}$$

$$[\sigma_F]_2=\frac{K_{FN2}\sigma_{FE2}}{S}=\frac{0.98\times380}{1.4}MPa=266MPa \tag{59}$$

d. Calculating load factor K.

$$K=K_AK_vK_{F\alpha}K_{F\beta}=1\times1.05\times1\times1.35=1.42 \tag{60}$$

e. Looking up for toothed coefficient.

It is gotten that $Y_{Fa1}=2.06$, $Y_{Fa2}=2.97$

f. Looking up for stress correction factor

It can be gotten $Y_{Sa1}=1.97$, $Y_{Sa2}=1.52$

g. Computing $\frac{Y_{Fa}Y_{Sa}}{[\sigma_F]}$ of gear and pinion and comparing them.

$$\frac{Y_{Fa1}Y_{Sa1}}{[\sigma_F]_1}=\frac{2.06\times1.97}{339.3}=0.012 \tag{61}$$

$$\frac{Y_{Fa2}Y_{Sa2}}{[\sigma_F]_2}=\frac{2.97\times1.52}{266}=0.017 \tag{62}$$

Value of gear is higher than that of pinion.

Design and calculate

$$m\geq\sqrt[3]{\frac{2\times1.42\times6\times10^4}{1\times17^2}\times0.017}mm=2.16mm \tag{63}$$

Compare results from tooth surface contact fatigue strength calculation of modulus greater than modulus m from tooth root bending fatigue strength calculations, due to size gear modulus m mainly depends on carrying capacity of bending strength decision, and tooth surface contact fatigue carrying capacity strength decision, only related with gear diameter, where it is desirable modulus calculated by bending strength of 2.16 and rounded to nearest standard value m = 2mm, according to contact strength calculated pitch diameter $d_1=52.8mm$, calculate gear number $z=\frac{d_1}{m}=\frac{52.8}{2}\approx26$.

3) Calculation ofgeometric dimensioning

a. Calculationof pitch diameter:

$$d=z\cdot m=26\times2mm=52mm \tag{64}$$

b.Calculation of gear width:

$$b=\phi_d\cdot d=0.8\times52mm=41.6mm \tag{65}$$

E. Worm shaft design

(1) First calculate torque shaft diameter

45 steel is chosen for quenching and tempering, and its hardness is $217-255HBS$.

Set $C=110$

$$d\geq C\sqrt[3]{\frac{P}{n}}mm=110\sqrt[3]{\frac{1.46}{2840}}=8.8 \tag{66}$$

Considering keyway, diameter increases by 7%, then

$$d=8.8\times(1+7\%)mm=9.4mm \tag{67}$$

So select $d=10mm$.

(2) Shaft structure design

First level worm gear can be arranged in middle of box, two symmetrical bearings, worm shaft shoulder by positioning, worm circumferential direction flat key connection and positioning of users. Meanwhile on the other side with a sleeve fixed bearing sleeve and cap positioning. Worm shaft structure is shown in "Fig.1".

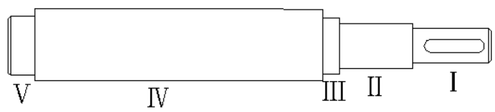

Figure 1. Worm shaft structure schematic diagram

Section I: Minimum diameter d_1 of coupling shaft is installed, so while choice of coupling torque calculation is $T_{ca}=K_AT_I$, check textbook 14-1, taking into account torque change is very small, then set

$$K_a=1.3, \text{then} T_{ca}=1.3\times4.89=6.4N\cdot m \tag{68}$$

According to calculation, torque should be less than nominal coupling torque, check mechanical condition of manual, as is shown in "Table 4".

TABLE IV.

SELECTION OF COUPLING[12]

Model	Nominal torque $(N\cdot m)$	Allowable rotating speed $(r/\min)$	L1	L	Shaft diameter (mm)	D (mm)
HL1	160	7100	27	32	14	90

Select $HL1$ type elastic sleeve as pin coupling.
Thus Section I is $d_1 = 14mm,$ and length $L_1 = 27 - 2 = 25mm,$, where keyway width, key height and key length are $5\times5\times25$, and total keyway is 4, where $L'' = 25 + 6 = 31mm$.
Total length of bearing cap is 20mm, taking cover outside according to convenience of removable coupling between end face on right side of distance is $L = 30mm$.
Thus $L_2 = 30 + 20 = 50mm$.
Section Ⅲ : Primary choice of single row tapered roller bearings, referring to requirements, diameter $d_3 = 20mm$ is selected. Check mechanical manual select 30204 type roller bearings, namely $d\times D\times B = 20\times47\times14$.
And
$T = 15.25, C = 12$, namely $d_3 = 20mm, L_3 = 11.2mm.$
Roller bearing is for shaft right shoulder positioning. Check manual and 30204 type bearing shaft locating shoulder height should be $h = 4.5mm$, and then dimensionof section IV can be determined.

F. Design and calculation of output shaft

(1) Power, rotate speed and torque of output shaft[4]

$$P_{\Pi} = P_1\cdot\eta_2\cdot\eta_3 = 0.73\times0.99\times0.4 = 0.29 \quad (69)$$

$$T_{\Pi} = T_1\cdot i_1\cdot\eta_2\cdot\eta_3 = 2.46\times30\times0.99\times0.4 = 29.22N\cdot m \quad (70)$$

rotate speed of shaft

$$n_{\text{Worm}} = \frac{n_{in}}{i} = \frac{2825r/\min}{31} = 91.1r/\min \quad (71)$$

(2) Seeking force acting on worm

$$F_t = F_a = \frac{2T_{\Pi}}{d_2} = 0.47N$$

$$F_a = F_{t1} = \frac{2T_{\Pi}}{d_1} = 0.82N$$

$$F_{r2} = F_{r1} = F_{t2}\times\tan\alpha = 0.47\times0.369 = 0.17N$$

(3) Initial confirmation of minimum diameter of shaft
45# steel is chosen, where hardness is $217 - 255HBS$.
According to fomula $P370(15-2)$, setting $A_0 = 110,$

$$d \ge A_0\sqrt[3]{\frac{P}{n}}mm = 110\times\sqrt[3]{\frac{0.29}{91.1}} = 16.2mm \quad (72)$$

Considering keyway, increase diameter by 10%, and then

$$d = 16.2\times(1\times10\%)mm = 17.82mm \quad (73)$$

Therefore $d = 20mm$.
(4) Structural design of shaft[13]
1) positioning, fixed and assembly of shaft
Single-stage worm reduction gear transmission, worm gear can be installed in housing a central, symmetrical relative two bearings, worm shaft shoulder positioned with left and right end surface with shaft cover positioned axially keyed and fit over, two bearings respectively bearing shoulder and shaft cover positioning, circumferential positioning with use of excessive or interference fit, stepped shaft, bearings from left into right bearing, as is shown in “Fig. 2”.

Figure 2. Worm shaft structure diagram

2)Confirmation of shaft diameter and length in each segment
Start of output design inside, check mechanical design manual, selection HL1 flexible shaft coupling, as is shown in “Table 5”.

TABLE V.

SELECTION OF COUPLING[4]

Type	Nominal torque $(N\cdot m)$	Allowable rotate speed (r/min)	L1	L	Shaft hole diameter (mm)
38	160	7100	27	52	20

Section I and Ⅶ: $d_1 = 20mm$, $L_1 = 38 - 2 = 36mm$ .set key on shaft 6×6, $L = 70mm$
Section II: Selection of single row tapered roller bearings, taken $d_3 = 30mm$ in accordance with the requirements, where model is primaries 30206 tapered roller bearings.
$d\times D\times T = 30\times62\times17.25, \text{and } c = 14mm$, Considering right end of bearing sleeve positioned, take a distance from inner wall of gear casing from a = 10mm, where error should be determined taking into account rolling bearing in housing S, taking S = 8. Known width T = 17.25,
then $L_3 = T + S + a + 2 + 4 = 17.25 + 8 + 10 + 2 + 4 = 41.25mm$
Section Ⅲ: as positioning shaft shoulder height is $h = (0.07\sim0.1)d_1 = 2mm$, $d_2 = 20 + 2\times2 = 24mm$, total width of bearing cap is 20mm, for purpose of easy accessibility, take cover outside outer end of coupling distance from right side of 30mm, so $L_2 = 30 + 20 = 50mm$,
Section Ⅳ: to install worm gear
$d_4 = d_3 + 2\times2.5 = 35mm$,
teeth width of worm gear is
$L_{\text{Worm gear}} = 0.75d_{a1} = 0.75\times75 = 57mm, L_4 = 57 - 2 = 55mm$ S
ection V: right end of section Ⅵ is bearing axial positioning $d_5 = d_4 + 2\times2.5 = 40mm$,

$$L_6 = \frac{d_{a1}}{2} + a - \frac{L_4}{2} = \frac{75}{2} + 10 - \frac{55}{2} = 20mm$$

Section VI: this section is bearing installation, so

$d_6 = d_3 = 30mm$, $L_7 = T + 2 = 17.25 + 2 = 19.25mm$ 3)

Circumferential positioning of shaft parts

Gear coupling half and positioning axes are used to connect flat key. According to $d_4 = 35mm$, it is gotten that flat key section is $b \times h = 10 \times 8$. Keyway milling with a length of 32mm, and in order to ensure that gear and shaft in good symmetry, gear hub and shaft coordination is $\frac{H7}{r6}$; Also connected to shaft coupling halves, respectively, for the choice of flat keys, half-coupling and shaft coordination is $\frac{H7}{k6}$. Rolling to circumference orientation is ensured by selected shaft diameter tolerance m6.

4)Shaft load

It can be seen shaft section is installed worm shaft dangerous section.

$$F_{t1} = F_{a2} = \frac{2T_1}{d_1} = \frac{2.46}{30} \times 10^3 \times 2 = 164\text{N} \quad (74)$$

$$F_{t2} = F_{a2} = \frac{2T_2}{d_2} = \frac{29.42}{124} \times 10^3 \times 2 = 474.5\text{N} \quad (75)$$

$$F_{r1} = F_{r2} = F_{t2} \cdot \tan 20^\circ = 172.7\text{N} \quad (76)$$

$$F_{NV1} = F_{NV2} = 0.5 \times F_r = 86.35\text{N} \quad (77)$$

$$F_{NH1} = F_{NH2} = 0.5 \times F_{t2} = 237.25\text{N} \quad (78)$$

$$T = 29.22 N \cdot m \quad (79)$$

$$M_H = 118.6 \times 105 = 12.5 N \cdot m \quad (80)$$

$$M_V = 43.2 \times 105 = 4.5 N \cdot m \quad (81)$$

TABLE VI.

LOAD PARAMETERS OF SHAFT

Load	H		V	
Bearing reaction N	F_{NH1}	F_{NH2}	F_{NV1}	F_{NV2}
	118.6	118.6	43.2	43.2
Bending moment M $N \cdot m$	$M_H = 12.5$		$M_V = 4.5$	
Total beding moment M	$M_1 = M_2 = \sqrt{M_H^2 + M_V^2} = 13.3 N \cdot m$			
Torque $T = 29.22 N \cdot m$				

$$\sigma_{ca} = \frac{\sqrt{M_1^2 + (\sigma T)^2}}{0.1 d_2^3} = 0.016MPa < [\sigma_{-1}] = 60MPa \quad (82)$$

Therefore security, load parameters of shaft are shown in “Table 6”.

(5) Accuracy check of shaft fatigue strength[14]

Since minimum diameter of shaft is determined by torque strength so there is no need to check each section. Keyway stress concentration factor than interference fit small, so just check shaft bearing is installed at shaft section.

Cross section resistance coefficient is

$$W = 0.1d^3 = 0.1 \times 30^3 = 2700mm^3 \quad (83)$$

Section modulus in torsion is

$$W_r = 0.2d^3 = 0.2 \times 30^3 = 5400mm^3 \quad (84)$$

Bending moment on left side of section E is

$$M = 13.3 \times \frac{30 - 24}{30} = 2.66 N \bullet m \quad (85)$$

Torque on section E is T=29.22 (86)

$$\sigma_b = \frac{M}{W} = \frac{2.68}{0.27} = 10MPa \quad (87)$$

$$\tau_T = \frac{T}{W_T} = \frac{29.22}{0.54} = 54MPa \quad (88)$$

Shaft material is 45 steel, quenched and tempered treatment by checking table.

$\sigma_B = 640MPa$, $[\sigma_{-1}] = 60MPa$, $\sigma_{-1} = 275$, $\tau_{-1} = 155$ (89)

Theoretical stress shaft cross-section due to concentration factor formation α_σ and α_τ

As $\frac{r}{d} = \frac{2.0}{30} = 0.07$, $\frac{D}{d} = \frac{35}{30} = 1.17$

$\alpha_\sigma = 2.0$, $\alpha_\tau = 1.31$

Material sensitivity coefficients of shaft are $q_\tau = 0.82$, $q_\tau = 0.85$

therefore effective stress concentration factors are

$$k_\sigma = 1 + q_r(\alpha_\sigma - 1) = 1.82 \quad (90)$$

$$k_T = 1 + q_T(\alpha_T - 1) = 1.26 \quad (91)$$

Size factors are $\varepsilon_\sigma = 0.67$, $\varepsilon_\tau = 0.82$, $\beta_\sigma = \beta_T = 0.92$

Shaft without surface hardening treatment

$$K_\sigma = \frac{k_\sigma}{\varepsilon_\sigma} + \frac{1}{\beta_\sigma} - 1 = 2.8 \quad (92)$$

$$K_\tau = \frac{k_\tau}{\varepsilon_\tau} + \frac{1}{\beta_\tau} - 1 = 1.62 \quad (93)$$

and according to characteristics coefficient of carbon steel

$\varphi_\sigma = 0.1 \sim 0.2$ taking $\varphi_\sigma = 0.1$; $\varphi_\tau = 0.05 \sim 0.1$, $\varphi_\tau = 0.05$

Calculate safety factor S_{ca}

$$s_\sigma = \frac{\sigma_{-1}}{k_\sigma \sigma_a + \varphi_\sigma \sigma_m} = 20.21 \quad (94)$$

$$s_\tau = \frac{\tau_{-1}}{k_\tau \tau_a + \varphi_\tau \tau_m} = 10.62 \quad (95)$$

$$s_{ca} = \frac{s_\sigma s_\tau}{\sqrt{s_\sigma^2 + s_\tau^2}} = 9.40 > s = 1.5 \quad (96)$$

Section on right side, cross section coefficient formula is

$$W = 0.1d^3 = 0.1 \times 35^3 = 4287.5mm^3 \quad (97)$$

Section modulus in torsion is

$$W_T = 0.2d^3 = 0.2 \times 35^3 = 8575mm^3 \quad (98)$$

Bending moment T_3 and torsional shear stress are

$$M = 13.3 \times \frac{30-24}{30} = 2.66 N \bullet m \quad (99)$$

$$\tau = \frac{T}{W_T} = \frac{29.22}{8575} \times 1000 = 3.4 MPa \quad (100)$$

$$\sigma_b = \frac{M}{W} = \frac{2.66}{0.43} = 6.2 MPa \quad (101)$$

Interference fit place $\frac{k_\sigma}{\varepsilon_\sigma}$ using interpolation method and set $\frac{k_T}{\varepsilon_T} = 0.8 \frac{k_\sigma}{\varepsilon_\sigma}$

$\frac{k_\sigma}{\varepsilon_\sigma}$ =3.16,

So $\frac{k_T}{\varepsilon_T} = 0.8 \times 3.16 = 2.53$ (102)

Surface quality coefficient is

$$\beta_\sigma = \beta_T = 0.92 \quad (103)$$

Size factor, so overall coefficients are

Shaft without surface hardening treatment

$$K_\sigma = \frac{k_\sigma}{\varepsilon_\sigma} + \frac{1}{\beta_\sigma} - 1 = 3.25 \quad (104)$$

$$K_\tau = \frac{k_\tau}{\varepsilon_\tau} + \frac{1}{\beta_\tau} - 1 = 1.62 \quad (105)$$

Also by characteristics of carbon steel factor $\varphi_\sigma = 0.1 \sim 0.2$ taking $\varphi_\sigma = 0.1$; $\varphi_\tau = 0.05 \sim 0.1$, $\varphi_\tau = 0.05$

Calculate safety factor S_{ca}

$$s_\sigma = \frac{\sigma_{-1}}{k_\sigma \sigma_a + \varphi_\sigma \sigma_m} = 23.27 \quad (106)$$

$$s_\tau = \frac{\tau_{-1}}{k_\tau \tau_a + \varphi_\tau \tau_m} = 8.38 \quad (107)$$

$$s_{ca} = \frac{s_\sigma s_\tau}{\sqrt{s_\sigma^2 + s_\tau^2}} = 7.38 > s = 1.5 \quad (108)$$

Therefore, strength shaft is sufficient.

G. Selection of standard part

(1) Selection of rolling[15]

According to conditions, life expectancy of bearings is: 12,000 hours.

Due to shaft under large axial force, tapered roller bearings are used.

Worm shaft bearing

30204GB/T294-1994 $d \times D \times B = 20 \times 47 \times 14$

Worm gear shaft bearing 30216 GB/T294-1994

$d \times D \times T = 30 \times 62 \times 17.25$

(2) Selection, check and calculation of key joint

1) Coupling1 and worm shaft are connected by flat key connection:

Shaft diameter $d_1 = 14mm$, $L_1 = 25mm$

According to handbook GB/T1096-2003 A-type flat key is chosen

$b = 5mm \quad h = 5mm \quad L = 10mm$

namely key of 5×10

It can be gotten that $l = L - b = 10 - 5 = 5mm$

$k = 0.5h = 2.5mm$

and $T_I = 2.46 N \cdot m$

According to textbook P106 formula (6-1) flat coupling strength conditions are

$$\sigma_p = \frac{2T \times 10^3}{kld} = \frac{2 \times 2.46 \times 10^3}{2.5 \times 5 \times 14} = 28.1 MPa < [\sigma_p](110MPa)$$

(109)

2) Worm shaft and worm wheel connected by flat key connection

Shaft diameter $d_3 = 35mm \quad L_3 = 42mm$

According to handbook GB/T1096-2003 A-type flat key is chosen： $b = 10mm \quad h = 8mm$

$L = 20mm$

namely key of 10×20

It can be gotten that $l = L - b = 20 - 10 = 10mm$

$k = 0.5h = 4mm$

and $T_{II} = 29.22$

Flat coupling strength conditions are

$$\sigma_p = \frac{2T \times 10^3}{kld} = \frac{2 \times 29.22 \times 10^3}{4 \times 10 \times 35} = 41.7 MPa < [\sigma_p](110MPa)$$

(110)

3) Output shaft and couplings connecting 2 flat key connection

Shaft diameter $d = 30mm \quad L = 19.25mm$

According to handbook GB/T1096-2005 A-type flat key is chosen： $b = 10mm \quad h = 8mm$

namely key of 18×70

It can be gotten that $l = L - b = 20 - 10 = 10mm$

$k = 0.5h = 4mm$

and $T_{II} = 29.22$

Flat coupling strength conditions are

$$\sigma_p = \frac{2T \times 10^3}{kld} = \frac{2 \times 29.22 \times 10^3}{4 \times 10 \times 35} = 41.7 MPa < [\sigma_p](110MPa)$$

(111)

Selection of box and all attachments

(1) Selection of box materials

Considering cast iron box easier to obtain reasonable shape and complex structure, good rigidity, easy cutting material was chosen for casting box.

(2)Selection of accessories [15]

1) selection of bearing cap

Because flange bearing cap bearing clearance easy to adjust, good sealing performance, so selection of flange is bearing cover.

2) selection of vision hole and hole cover

As hole rolled steel plate light cover structure, following without machining, so selection of rolling steel plate as hole cover. At the same time should be added to tpaper gasket between tank to prevent oil spills.

3) selection of breather

Due to use of outside world, there is dust, using ventilation chamber M20 1.5

4) selection of oil subject

Because oil dipstick mark compared to circular, elongated oil standard structure is simple, easy to process, saving materials, so selection of an elongated dipstick.

5)Selection of plug hole

Selection is fine tooth hex plug and gasket M14×1.5.

IV. CONCLUSIONS

Design for this new automatic wall plastering machine, driven by a rack and pinion system from top to down plastering. Reducer is a focus design, so each part reducer through detailed selection and calculation and checking results and conclusions of design requirements in line with theory, which can provide new product design plastering machine effective theoretical design basis.

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