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Theoretical design research of a joy rotating coffee cup

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ABSTRACT

In this paper, based on the survey of the existing large rotary equipment in places of entertainment, a suitable for children's entertainment rotating coffee cup was mainly designed and researched. The structure of the rotating coffee cup has been discussed, and the relevant parts have been selected and checked. Especially, the selection precision material and the related parameters of the main drive gear have carried on the detailed calculation and analysis to make the design of the rotating coffee cup has a reasonable theoretical basis.

KEYWORDS

Rotation; Coffee cup; Gear; Drive.

INTRODUCTION

Rotating coffee cup is a new product which is launched to follow the trend of The Times. In the amusement facilities, because of its clever and beautiful, vivid modeling, the coffee cup has been favored by the majority of children and tourists. Human-computer interaction of the product is very strong, the product is suitable for different age groups of tourists, especially conducive to the cultivation of practice ability and hands-on awareness of children, let children learn from active operation, at the same time exercise children's reaction to the agility.

From children's rotating coffee cup market in recent years, more and more manufacturers especially more professional manufacturer have fully realized the importance of product function. Survey results show that four functions indexes of the rotating coffee cup for convenience, health, safety and energy saving are for four of the results of the survey. Based on the current international energy shortages, people's emphasis on energy and domestic electricity shortage frequently in recent years, the energy saving will be rotating coffee cup a development direction. For example, in the premise of guarantee quality, the lightweight materials are used to reduce weight and the consumption of electrical energy, these energy saving products in the next few years will get promotion and application of the larger social dynamics[1]. In the next few years, light, beautiful, and bright colors will be the main development direction of rotating coffee cup product appearance. In today's era of the consumers pursuit of fashion, personality and refinement, consumer have a special liking to the rotation of the beautiful coffee cup, and about half of the consumer more love the light and lovely rotating coffee cup. Coffee pot and coffee cup are the main part of the rotating coffee cup, it should not be too heavy, so light metal and hard plastic can been chosen. The foundation was with cement fixation, underground, there was a space to install drive mechanism of the rotating "coffee cup", such as motor straight association-like planetary pin wheel reducer, coupling, bevel gear, etc. According to the existing national standard regulation, the main technique indexes of the rotating coffee cup have the following safety and technical requirements, rated voltage, rated power and highest speed gears. When running, the maximum velocity on the circumference of a circle was not more than 270 m/min, dynamic load was not less than 1.3.

SELECTION AND CHECKING OF THE RELATED PARTS

Speed control function is one of the basic requirements for rotating entertainment devices. In the design, the speed control method of the rotating coffee cup was to depend on the motor, which was usually adapted to frequency conversion governor. In order to implement motor drive, frequency conversion speed governor was the use of ac asynchronous motor synchronous speed n0 which was changing over power frequency[2].

Selection of the motor

(1) Power what the rotating coffee cup required P_w

Power P_w what the rotating coffee cup rotation required was 13.5 kW.

TABLE 1 : Basic parameters of the motor 1

Model of the motor	Speed of full load	Reduction ratio
J02-62-4T2/17	1440	15

TABLE 2 : Basic parameters of the motor 2

Rated power (W)	Synchronous speed(r/min)	Speed of full load(r/min)	Voltage (V)	Current (A)	Factor of the power
17	1500	$1440\,{}^{r}\!/_{min}$	380	12.6	0.78
Quality (kg)	Reduction ratio	Torque of full load	Diameter of overhang shaft of the motor (mm)	Length of overhang shaft of the motor (mm)	Center height of the motor (mm)
168	15	2.0	38	80	132

(2) Calculation of total efficiency of the transmission device η

When the power was through a transmission vice or sports, a loss would occur. So, total efficiency of the multi-stage tandem was $\eta = \eta_1\eta_2 \ldots\ldots \eta_w$.

In the formula, η_1 was efficiency of the coupling, taking η_1 =0.98; η_2 was transmission efficiency of the bevel gear, taking η_2 =0.98; η_3 was efficiency of the planetary needle wheel reducer, η_3 =0.90; η_4 was efficiency of the thrust ball bearing, taking η_5 =0.99.

So, $\eta = \eta_1\eta_2\eta_3\eta_4$ =0.86

Efficiency of the motor: $P_r = \frac{P_w}{\eta} = 15.7kW$

The basic parameters for the selected motor as shown in TABLE 1 and TABLE 2:

Distribution of the transmission ratio

Transmission ratio of the gear reducer $i_f = 15$

In this design, the motor straight association-like planetary pin wheel reducer has been selected. The model was WBD17-5 - I, reduction ratio was 15, the output speed was 96 RPM. Speed of the rotating coffee cup required was 6 RPM. Considering the lubrication problem of the two stage gear, the cone gear should have the similar oil immersion depth. The total transmission ratio: $i_f = n_1 / n_2 = 96/6 = 16$. Considering the lubrication problem of the two stage gear, two levels of the wheel should have the similar oil immersion depth. High speed transmission ratio i_1 and low speed transmission ratio i_2 of two stage gear reducer were the ratio of 1.3, namely $i_1 = 1.3 i_2$.

The reduction ratio of the bevel gear: $i_1 = i_f / i_2 = 16 / i_2$

So, $i_2 = \sqrt{1.31 i_f} = 3.51$; $i_1 = 1.3 i_2 = 4.563$

Calculation of movement and dynamic parameters of the transmission

(1) Calculation of the each shaft speed

$$n_0 = 96\ r/\min \quad n_I == 6\ r/\min$$

(2) Input power of the each shaft

$$P_d = 17kW$$

$$P_I = P_d \cdot \eta_1 \cdot = 17 \times 0.98 = 16.66kW$$

$$P_{II} = P_I \cdot \eta_2 \cdot \eta_3 = 16.66 \times 0.98 \times 0.98 = 16kW$$

$$P_{III} = P_{II} \cdot \eta_4 = 16 \times 0.99 = 15.7kW$$

(3) Input torque of the each shaft

$$T_d = 9550 P_d / n_o = 9550 \times 17 / 1440 = 112.74 n \cdot m$$

$$T_I = 9550 P_I / n_I = 9550 \times 16.66 / 96 = 1657 n \cdot m$$

$$T_{II} = 9550 P_{II} / n_{II} = 9550 \times 16 / 96 = 1591.67 n \cdot m$$

$$T_{III} = 9550 P_{III} / n_{III} \ 9550 \times 15.7 / 6 = 24989.17 n \cdot m$$

Design of the gears[3] [4]

(1) Choose type, precision grade, materials and teeth of the gears

1) According to the transmission scheme, the spur bevel gear transmission has been selected. The conveyor working was generally machinery, speed was not high. So chose 8 precision.

2) Selection of the materials. By the look-up table, material of the pinion selected was 45 steel, the modulation process, the average hardness was 235 HBS. Material of the wheel was 45 steel, the normalizing, hardness was190 HBS.

Difference of the hardness was 45 HBS. Choose teeth of pinion gear $z_1 = 24$, then $z_2 = i_2 z_1 = 3.51 \times 24 = 84.24$. Taking $z_2 = 85$, ratio of the teeth: $u = 85/24 = 3.54$.

According to calculation of fatigue strength of the tooth surface contact:

$$d_1 \geq 2.92 \sqrt[3]{\frac{Z_E}{[\sigma_H]} \cdot \frac{KT_1}{\phi_R (1-0.5\phi_R)^2 u}}$$

$\phi_R = \frac{b}{R}$, generally taking ϕ_R=1/3.

Look-up table to the elasticity influence coefficient of the material:

$$Z_E = 189.8\sqrt{MPa}$$

Calculated: $T_1 = T_d = 1127 \mathrm{n} \cdot \mathrm{m}$; u=3.54

Look-up table to $K = K_A K_V K_\alpha K_\beta = 1.15 \times 1.2 \times 1.4 \times 1.098 = 2.3$

Look-up table to $K_{HN1} = 1.01$, $\sigma_{\mathrm{hlim1}} = 1130 \mathrm{m}Pa$, 取taking S=1

Calculating $[\sigma_H] = \frac{K_{HN1} \cdot \sigma_{H\mathrm{lim1}}}{S} = 1131.5 \mathrm{m}Pa$

So, $\mathrm{d}_1 \geq 2.92 \sqrt[3]{\left(\frac{189.8}{1131.5}\right)^2 \cdot \frac{2.3 \times 1127 \times 10^3}{0.3 \times (1-0.5 \times 0.3)^2 \times 3.54}} = 32.7mm$

So, The modulus: $\mathrm{m} = \frac{d_1}{z_1} = \frac{32.7}{24} = 1.36$, taking the standard: m=2mm。

The diameter of the dividing circle on the big end:

$$\mathrm{d}_1 = 2z_1 = 2 \times 24 = 48 \geq 32.7$$

(2) Related parameters in the gears

1) Taking Z_1=24, so, $\mathrm{d}_1 = mz_1 = 24 \times 2 = 48$, $\mathrm{d}_2 = mz_2 = 2 \times 85 = 170$.

2)Top moment of the pitch cone $R = \frac{\mathrm{d}}{2}\left(\sqrt{1+\mathrm{u}^2}\right) = \frac{48}{2}\left(\sqrt{1+3.54^2}\right) = 88\mathrm{mm}$

3) $\delta_1 = \mathrm{arc}\tan\frac{1}{u} = \arctan\frac{1}{3.54} = 15°53`33``$ $\delta_2 = 74°6`26``$

4) 大端齿顶圆直径：Diameter of the addendum circle at the big end:

The pinion: $\mathrm{d}_{a1} = d_1 + 2m\cos\delta_1 = 71.85$ mm

The wheel: $\mathrm{d}_{a2} = d_2 + 2m\cos\delta_2 = 231.3mm$

5) Taking $\mathrm{b}_1 = b_2 = 42mm$

(3) According to bending strength of the tooth root, calculating

$$m \geq \sqrt[3]{\frac{4KT_1}{\phi_R(1-0.5\phi_R)^2 Z_1^2 \sqrt{1+u^2}} \cdot \frac{Y_{Fa} \cdot Y_{Sa}}{[\sigma_F]}}$$

Getting $T_1 = 213N \cdot M$; $Z_1 = 24$; $u = 4.042$; $K = K_A K_V K_{F\alpha} K_{F\beta} = 2.3$

Look-up table to $Y_{Fa1} = 2.65$ $Y_{Fa2} = 2.20$ $Y_{Sa1} = 1.58$ $Y_{Sa2} = 1.78$

Look-up figure to $\sigma_{F1} = 420MPa$ $\sigma_{F2} = 320MPa$

According to the formula $N = 60njL_b$, Look up: $K_{FN1} = 0.91$, $K_{FN2} = 0.92$

Calculating $[\sigma_F]_1 = \frac{K_{FN1} \cdot \sigma_F 1}{1.4} = \frac{0.91 \times 420}{1.4} = 273MPa$

$$[\sigma_F]_2 = \frac{0.92 \times 320}{1.4} = 210MPa$$

(4) Calculation and comparison of the bending fatigue strength of the gears

$$\frac{Y_{Fa1} \cdot Y_{Sa1}}{[\sigma_F]_1} = \frac{2.65 \times 1.58}{273} = 0.0153$$

$$\frac{Y_{Fa2} \cdot Y_{Sa2}}{[\sigma_F]_2} = \frac{2.2 \cdot 1.78}{210} = 0.0186$$

The values of the wheel was big.

1) Calculation of the modulus m（According to calculation of the wheel） :

$$m \geq \sqrt[3]{\frac{4KT_1}{\phi_R(1-0.5\phi_R)^2 Z_1^2 \sqrt{1+u^2}} \cdot \frac{Y_{Fa} \cdot Y_{Sa}}{[\sigma_F]}}$$

$$= \sqrt[3]{\frac{4 \times 213 \times 10^3 \times 2.3}{0.3 \times (1-0.5 \times 0.3)^2 \times 24^2 \times \sqrt{1+4.033^2}} \times 0.0186}$$

$$= 3.01mm$$

Consistent with the calculated before m value, then parameters of the gears as follows:

Z1=22, then $d_1 = mz_1 = 22 \times 3 = 66$; $z_2 = z_1 \times i_2 = 22 \times 4.033 = 88.7$, taking $z_2 = 89$,

$d_2 = 3 \times 89 = 247mm$.

2) Top moment of the pitch cone: $R = \frac{d}{2}\left(\sqrt{1+u^2}\right) = \frac{66}{2}\left(\sqrt{1+4.042^2}\right) = 137.408mm$

3) $\delta_1 = \arctan\frac{1}{u} = \arctan\frac{1}{4.042} = 13°53`33``$ , $\delta_2 = 76°6`26``$

4) Diameter of the addendum circle at the big end:

The pinion: $d_{a1} = d_1 + 2m\cos\delta_1 = 71.85$ mm

The Wheel: $d_{a2} = d_2 + 2m\cos\delta_2 = 231.3mm$

5) Breadth: $b = \phi_R \cdot R = 0.3 \times 137.408 = 41.22mm$

Taking $b_1 = b_2 = 42mm$

6) The average pitch circle diameter: $d_m = d(1-0.5\times\phi_R)$

$$d_{m1} = d_1(1-0.5\times\phi_R) = 66\times0.85 = 56.1\text{mm}$$
$$d_{m2} = d_2(1-0.5\times\phi_R) = 267\times0.85 = 226.95\text{mm}$$

7) The equivalent pitch circle radius:

$$r_v = \frac{d_m}{2\cos\delta}$$

$$r_{v1} = \frac{d_{m1}}{2\cos\delta_1} = \frac{56.1}{2\times0.91} = 31.2\text{mm}$$

$$r_{v2} = \frac{d_{m2}}{2\cos\delta_2} = \frac{226.95}{2\times0.09} = 1260.8\text{mm}$$

8) Equivalent number of teeth Zv

$$Z_{v1} = \frac{d_{v1}}{m_{m1}} = \frac{z_1}{\cos\delta_1} = \frac{66}{0.9} = 73.3$$

$$Z_{v2} = \frac{d_{v2}}{m_{m2}} = \frac{z_2}{\cos\delta_2} = \frac{247}{0.1} = 2470$$

9) The equivalent gear transmission ratio

$$u_v = \frac{z_{v2}}{z_{v1}} = \frac{2470}{73.7} = 33$$

10) The average modulus

$$m_m = m(1-0.5\phi_R) = 3\times0.85 = 2.55$$

The size of the high speed gear transmission as shown in TABLE 3

TABLE 3 : Size of the high speed gear transmission

Name	Computational formula	Result
Centre-to-centre spacing	$a = \frac{m(z_1+z_2)}{2\cos\beta}$	192mm
Root diameter	$d_f = d - 2(h_{a*} + c^*)m$	$d_{f1} = 84\text{mm}$ $d_{f2} = 282mm$
Breadth	B	$B_1 = 98\text{mm}$ $B_2 = 93\text{mm}$
Number of teeth	Z_1 Z_2	
Tip diameter	$d_a = d + 2m\cos\delta$	$d_{a1} = d_1 + 2m\cos\delta_1 = 71.85$ mm $d_{a2} = d_2 + 2m\cos\delta_2 = 231.3mm$
Pitch cone apex distance	$R = \frac{d}{2}\left(\sqrt{1+u^2}\right)$	137.408mm
Equivalent pitch circle radius	$r_v = \frac{d_m}{2\cos\delta}$	$r_{v1} = \frac{d_{m1}}{2\cos\delta_1} = \frac{56.1}{2\times0.91} = 31.2\text{mm}$

		$r_{v2}=\frac{d_{m2}}{2\cos\delta_2}=\frac{226.95}{2\times 0.09}=1260.8\text{mm}$
The equivalent gear transmission ratio	$u_v=\frac{z_{v2}}{z_{v1}}$	$u_v=\frac{z_{v2}}{z_{v1}}=\frac{2470}{73.7}=33$

CONCLUSIONS

In the design, the straight association-like planetary pin wheel reducer has been adopted to the rotating coffee cup, and the drive gear has been carried on the detailed analysis, design and calculation. According to the transmission scheme, the spur bevel gear transmission has been selected; conveyor 8 precision has been selected. The material of the pinion was 45 steel, modulations, the average hardness was 235 HBS; the material of the big gear was also 45 steel, the normalizing, hardness was 190 HBS. Difference of the hardness was 45 HBS. In narrative, conclusion of the analysis calculation of gear type, precision grade, material and teeth conformed to the theoretical research of the rotating coffee cup.

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