Original Research Article

Analysis of the force on the tooth surface of the cycloid pinwheel of RV reducer considering manufacturing errors

ABSTRACT

The mechanical model of RV reducer considering displacement and stiffness is established, and the RV110E reducer is taken as the research object, and the model is solved by MATLAB computing software without considering the manufacturing error. After the solution is completed, the obtained results are compared with the analysis results based on Hertz contact theory, and the error is within a reasonable range, which verifies the reliability of the mechanical model. Subsequently, the manufacturing errors of the RV reducer were measured based on the high-precision measuring instrument, and the manufacturing errors were imported into the mechanical model for solving, and the results of the cycloidal pinwheel tooth surface force considering the manufacturing errors were obtained, and the maximum value of the load was 1,713N, and the number of engaged teeth was 17 pairs, which was an increase of 89% of the maximum value of the load, and a reduction of two pairs of the number of engaged teeth, compared with that of the case where the manufacturing errors were not taken into account. Based on the existing experimental data, the force on the tooth surface of the cycloid pinwheel after considering the manufacturing error is closer to the actual situation.

Keywords: RV reducer; Cycloid pinwheel; Manufacturing error; Force analysis.

1. INTRODUCTION

RV reducer is a key component of industrial robots, which is a high-performance reducer based on cycloid planetary drive. Cycloid pinwheel as a key component of RV reducer, its running condition and bearing capacity directly affects the overall performance of RV reducer. According to statistics, 90% of RV reducer failure events are caused by transmission gear failure, in which the tooth fatigue failure of the cycloid pinwheel is the main failure mode [1]. Therefore, it is of great significance to investigate the force on the tooth surface of the cycloid pinwheel of RV reducer to solve such problems. The existing studies on the force on the tooth surface of the RV reducer cycloid pinwheel have not considered the influence of the manufacturing error of the RV reducer, and the reference value is limited. In this paper, by establishing the mechanical model of RV reducer considering displacement and stiffness, the manufacturing error of RV reducer is introduced into the force analysis of the cycloid pinwheel tooth surface, and the results of the force on the tooth surface of the cycloid pinwheel which is closer to the actual working condition are obtained. The research results provide a theoretical basis for the design optimization and strength analysis of the cycloid pinwheel of RV reducer.

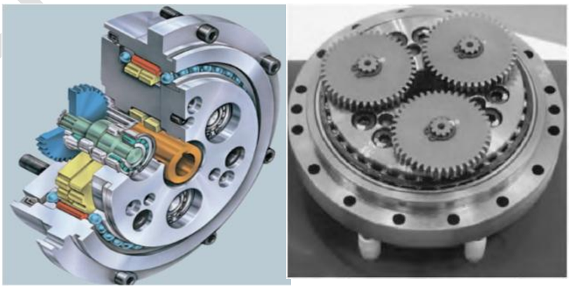

Fig. 1 .RV Reducer

2. INTRODUCTION TO RV REDUCERS

RV reducer is a new type of gear reducer developed on the basis of cycloid planetary transmission. Compared with other reducers, it has a series of advantages such as small size, light weight, large transmission ratio range, high transmission efficiency, smooth transmission, low noise, large overload capacity, less failure, long life, etc., and is widely used in industrial robots, high-precision numerical control machine tools, medical equipment, engineering machinery and other fields. As a two-stage planetary reducer, the first stage of the RV reducer is an involute external meshing planetary gear drive, and the second stage is a cycloid planetary drive, because of the second stage of the cycloid drive, and thus enough to use less volume to achieve a larger transmission ratio in the design of planetary drive [2].

3.MECHANICAL MODELING OF RV REDUCER BASED ON DISPLACEMENT AND STIFFNESS

Comprehensive consideration of part machining error, assembly error, clearance, part contact deformation and load, rotating parts of the inertia load and other factors, the traditional mechanical model can not meet the requirements, the need to establish the consideration of displacement, stiffness of the mechanical model, the RV reducer mechanical model is shown in Figure 2 [3].

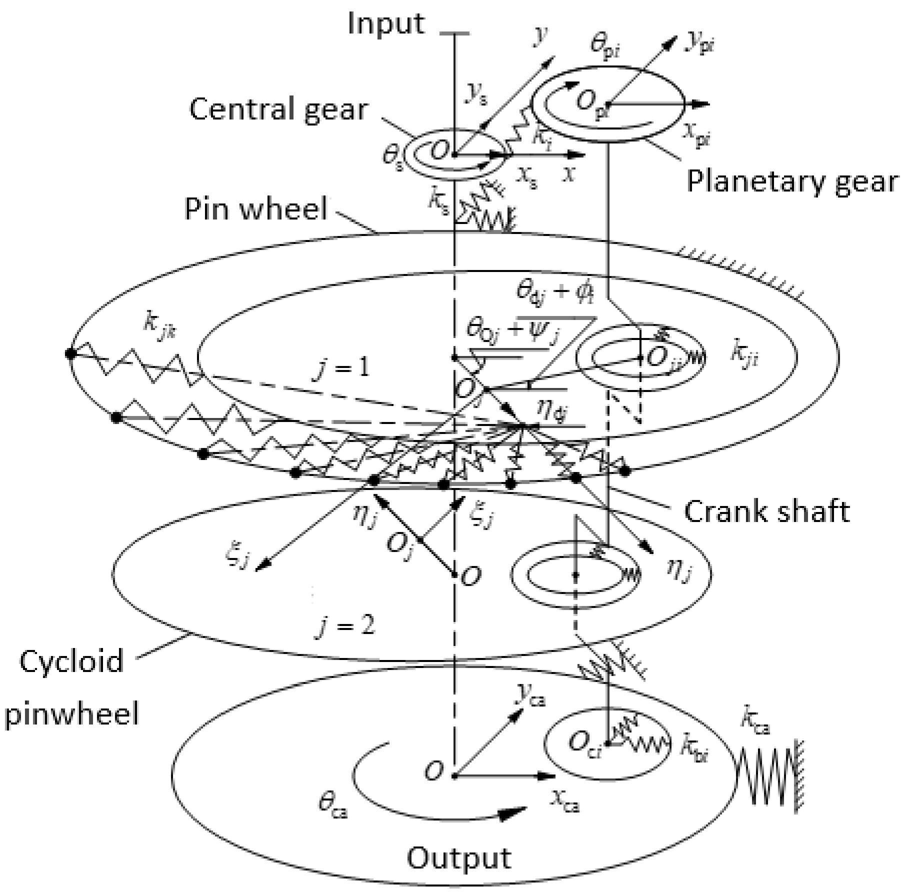

Fig. 2 Mechanical model of RV reducer

In the model, the support (bearing or shaft) stiffness between parts and the gear tooth meshing stiffness are represented by springs, such as the meshing stiffness between the Central gear and the planetary wheelk_i, the bearing support stiffness between the crankshaft and the hole of the balance wheel shaft k_{ji}, the bearing support stiffness between the crankshaft and the hole of the planetary carrier shaft k_{bi}, the bearing support stiffness between the planetary carrier and the pinwheel k_{ca}, and *the meshing stiffness between the balance wheel and the pinion k_{jk} and the Central geark_s. The above stiffnesses are determined as follows: k_i is determined by using the bending deformation of the wheel teeth, and k_s is obtained by calculating the bending deformation of the Central gear shaft, while k_s is determined by using the bending deformation of the wheel teeth k_{jk} and the support stiffness of the Central geark_s. The above stiffnesses are determined as follows: k_i is determined by using the bending deformation of the wheel teeth, k_s is obtained by calculating the bending deformation of the Central gear shaft, and k_{ji}, k_{bi}, k_{ca}, k_{jk} are determined by using the Palmgren formula.*

The next step is to determine the relevant static and dynamic coordinate systems. Taking the center 0 of the fixed pinwheel as the origin, the section perpendicular to its axis is the plane static coordinate system (x, y); starting from the input end, i.e., the end of the Central gear, the two Cycloid pinwheels are numbered j=1, 2, and the theoretical center of mass of the Cycloid pinwheels, 0_j, as the origin, and the eccentric direction of the Cycloid pinwheels is the η_j axis, and the rotation of the Cycloid pinwheel by 90° along the direction of its rotational angle is the ξ_j axis, to build up a dynamic coordinate system of the Cycloid pinwheels $(\eta_j,\ \xi_j)$. In the dynamic modeling process, the η_j axis of the pendulum wheel with j=1 is taken to be in the same direction as the x-axis of the static coordinate system at the starting position.

In addition, in the mechanical model, each spring is set at the positive direction contact of the coordinate axes of each part or at the wheel teeth meshing, and each factor is set so that the spring is pulled positively and pressed negatively; φ_i denotes the relative position of the crankshaft shaft holes on the pendulum wheels (or the planetary carriers), and is taken as $\varphi_i = 2\pi(i-1)/3\ (i = 1,\ 2,\ 3)$, and ψ_j denotes the relative position of the theoretical centers of mass of two pendulum wheels, 0_j, and is taken as $\psi_j = (j-1)\pi(j = 1,\ 2)$.

By analyzing the force condition of each part at any position, based on D Alembert's principle, the mathematical model of the dynamic transmission error of this system is established as follows:

$$m_{\mathrm{s}}\ddot{x}_{\mathrm{s}}+F_{\mathrm{sx}}+\sum_{i=1}^{3}(F_i\cos A_i)=0$$
$$m_{\mathrm{s}}\ddot{y}_{\mathrm{s}}+F_{\mathrm{sy}}+\sum_{i=1}^{3}(F_i\sin A_i)=0$$
$$J_{\mathrm{os}}\ddot{\theta}_{\mathrm{si}}+\sum_{i=1}^{3}(F_iR_{bs})=\frac{T_{\mathrm{out}}}{i_z}$$
$$m_{\mathrm{sp}}\left[\ddot{x}_{\mathrm{p}i}-R_{\mathrm{dc}}\omega_{\mathrm{c}}^2\cos(\theta_{\mathrm{c}}+\phi_i)-R_{\mathrm{dc}}\ddot{\theta}_{\mathrm{ca}}\sin(\theta_{\mathrm{c}}+\phi_i)-2\omega_{\mathrm{c}}\dot{y}_{\mathrm{p}i}\right]$$
$$-F_i\cos A_i+\sum_{j=1}^{2}F_{jix}+F_{cix}=0$$
$$m_{\mathrm{sp}}\left[\ddot{y}_{\mathrm{p}i}-R_{\mathrm{dc}}\omega_{\mathrm{c}}^2\sin(\theta_{\mathrm{c}}+\phi_i)+R_{\mathrm{dc}}\ddot{\theta}_{\mathrm{ca}}\cos(\theta_{\mathrm{c}}+\phi_i)+2\omega_{\mathrm{c}}\dot{x}_{\mathrm{p}i}\right]$$
$$-F_i\sin A_i+\sum_{j=1}^{2}F_{jj}+F_{cij}=0$$
$$J_{0\mathrm{p}}\ddot{\theta}_{\mathrm{p}i}-F_iR_{\mathrm{bp}}-e\sum_{j=1}^{2}\left[F_{jxx}\sin\left(\theta_{\mathrm{p}}+\psi_j\right)+F_{jy}\cos\left(\theta_{\mathrm{p}}+\psi_j\right)\right]=0$$
$$(i=1,2,3)$$
$$m_{\mathrm{br}}\left[\ddot{\eta}_{\mathrm{d}j}\cos\left(\theta_{\mathrm{p}}+\psi_j\right)-e\omega_{\mathrm{p}}^2\cos\left(\theta_{\mathrm{p}}+\psi_j\right)-e\ddot{\theta}_{\mathrm{Q}j}\sin\left(\theta_{\mathrm{p}}+\psi_j\right)\right.$$
$$\left.-2\omega_{\mathrm{p}}\dot{\eta}_{dj}\sin\left(\theta_{\mathrm{p}}+\psi_j\right)\right]-\sum_{i=1}^{3}F_{jkx}+\sum_{k=1}^{\frac{z_k}{2}}\left[F_{jk}\cos\left(\alpha_{jk}+\theta_{\mathrm{p}}+\psi_j\right)\right]=0$$
$$m_{\mathrm{br}}\left[\ddot{\eta}_{\mathrm{d}j}\sin\left(\theta_{\mathrm{p}}+\psi_j\right)-e\omega_{\mathrm{p}}^2\sin\left(\theta_{\mathrm{p}}+\psi_j\right)+e\ddot{\theta}_{\mathrm{o}j}\cos\left(\theta_{\mathrm{p}}+\psi_j\right)\right.$$
$$\left.+2\omega_{\mathrm{p}}\dot{\eta}_{\mathrm{d}j}\cos\left(\theta_{\mathrm{p}}+\psi_j\right)\right]+\sum_{i=1}^{3}F_{jij}+\sum_{k=1}^{\frac{z_r}{2}}\left[F_{jk}\sin\left(\alpha_{jk}+\theta_{\mathrm{p}}+\psi_j\right)\right]=0$$
$$J_{0j}\ddot{\theta}_{dj}-\sum_{k=1}^{\frac{z_r}{2}}\left[F_{jk}R_d\sin\alpha_{jk}\right]-R_{dc}\sum_{i=1}^{3}\left[F_{jw}\cos(\theta_c+\phi_i)-F_{jkx}\sin(\theta_c+\phi_i)\right]=0$$
$$(j=1,2)$$
$$m_{\mathrm{ca}}\ddot{x}_{\mathrm{ca}}-\sum_{i=1}^{3}F_{\mathrm{cix}}+F_{\mathrm{car}}=0$$
$$m_{\mathrm{ca}}\ddot{y}_{\mathrm{ca}}-\sum_{i=1}^{3}F_{\mathrm{c}iy}+F_{\mathrm{cay}}=0$$
$$J_0\ddot{\theta}_{\mathrm{ca}}+R_{\mathrm{dc}}\sum_{i=1}^{3}\left[F_{\mathrm{ccx}}\sin(\theta_{\mathrm{c}}+\phi_i)\right]-R_{\mathrm{dc}}\sum_{i=1}^{3}\left[F_{\mathrm{ccy}}\cos(\theta_{\mathrm{c}}+\phi_i)\right]+T_{\mathrm{out}}=0$$

Where m_s is the mass of the Central gear; m_{sp} is the sum of the masses of the planetary wheel and the crank shaft; m_{bx} is the mass of the balance wheel; J_{op} is the moment of inertia of the planetary wheel and the crank shaft; J_{oj} is the moment of inertia of the balance wheel; ω_c is the theoretical angular velocity of the planetary carrier; ω_p is the theoretical angular velocity of the planetary wheel rotation; T_{out} is the load torque.

Taking the RV110E reducer as the research object, its parameters are inputted into the established mechanical model, and solved based on MATLAB calculation software to obtain the number of teeth and their loads that are simultaneously engaged without considering the manufacturing error. Comparison of the results with the calculation results based on Hertz contact theory, as shown in Figure 3, the two calculation results are basically the same, which verifies the reliability of the mathematical model [4].

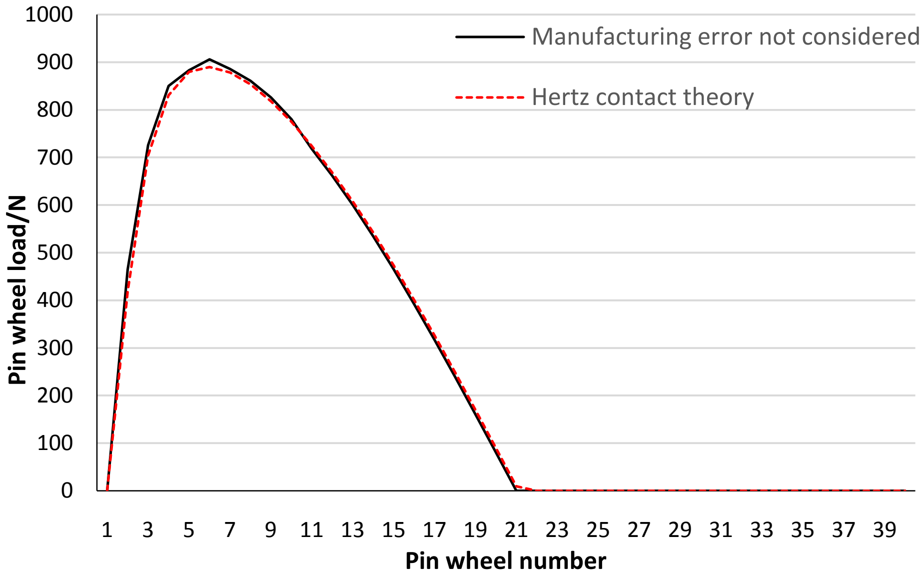

Fig. 3 Comparison of cycloid pinwheel tooth loads without considering manufacturing errors and based on Hertz contact theory

3.ANALYSIS OF FORCES ON THE TOOTH SURFACE OF A CYCLOID PINWHEEL CONSIDERING MANUFACTURING ERRORS

Due to the inevitable existence of errors in the processing and manufacturing process, so that the actual working load of the RV reducer cycloid pinwheel and the ideal working load there is a gap, based on the VMZ-R4540 type high-precision automatic image measuring instrument, the RV110E type reducer of the various manufacturing errors measured, the results of the measurements are shown in the following tables [5].

Table 1 Planetary gear base circle eccentricity error

	1		2		3	
	/μm	/(°)	/μm	/(°)	/μm	/(°)
	33.2	227.3	44.4	225.9	46.0	259.1

Table 2 Eccentricity errors of the crankshaft bore of the Cycloid pinwheel

	1		2		3	
	/μm	/(°)	/μm	/(°)	/μm	/(°)
1	6.4	270.0	14.9	270.0	4.5	90.0
2	4.2	270.0	0.5	90.0	4.8	270.0

Table 3 Eccentricity error of crankshaft eccentric cams

	1	2	3

	/μm	/(°)	/μm	/(°)	/μm	/(°)
1	19.2	180.0	18.9	0.0	14.2	0.0
2	19.2	180.0	38.3	295.6	14.2	0.0

Table 4 Eccentricity errors of planetary carrier crankshaft bores

1		2		3	
/μm	/(°)	/μm	/(°)	/μm	/(°)
0.4	0.0	18.0	80.6	7.4	112.9

The measured manufacturing errors are entered into the established mechanical model and solved by MATLAB computing software to obtain the cycloidal pinwheel tooth surface loads considering manufacturing errors. The maximum value of the tooth surface load of the cycloid pinwheel considering manufacturing errors is 1713N, and the number of meshing pairs is 17 pairs. Comparing the obtained results with the results without considering the manufacturing error, as shown in Fig. 4, it can be seen that the maximum contact load increases by 89%, and the number of meshing pairs decreases by 2 pairs, which is a large difference between the two results. Based on the existing experimental data, it can be seen that the force on the tooth surface of the cycloid pinwheel after considering the manufacturing error is closer to the actual situation [6]

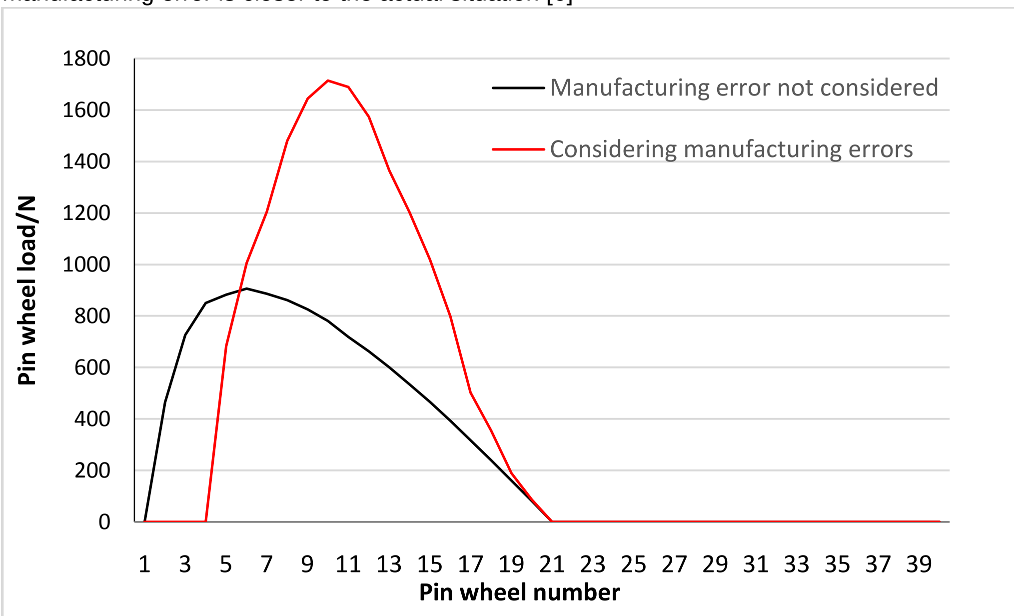

Fig. 4 Comparison of cycloid pinwheel tooth load considering manufacturing error and without manufacturing error

4. CONCLUSIONS

Compared with the traditional calculation method, this paper introduces the manufacturing error of RV reducer into the force analysis of the cycloid pinwheel by establishing the mechanical model of RV reducer considering displacement and stiffness. And based on the MATLAB computing software to solve the model, obtained the consideration of the RV reducer manufacturing error of the cycloid pinwheel tooth surface force situation, based on the existing experimental data can be seen, compared with the case of non-consideration of the manufacturing error of the case of the manufacturing error is closer to the actual force situation. The research results provide a necessary reference for future research on the force analysis and strength calculation of key components of RV reducer.

REFERENCES

1. Zhu Liangbin. Research on contact fatigue of Cycloid pinwheel based on dynamic analysis of multi-tooth meshing[D]. Kunming University of Science and Technology,2021.DOI:10.27200/d.cnki.gkmlu.2021.000286.
2. HAN Linshan,WENQuan.Force analysis and stress calculation of rotary arm bearing of RV gearbox[J]. Automobile Practical Technology,2020,45(23):123-125+135.DOI:10.16638/j.cnki.1671-7988.2020.23.039.
3. Zhao Yakun,ShangguanLinjian.RV reducer cycloid key error software production technology and data processing[J]. Automobile Practical Technology,2021,46(01):73-77.DOI:10.16638/j.cnki.1671-7988.2021.01.023.

4. Jiao W.-H. Research on transmission error of 2K-V reducer and its cycloid life [D]. North China University of Water Resources and Hydropower,2020.DOI:10.27144/d.cnki.ghbsc.2020.000036.
5. LinsanHan,ChaoKe,Caijuan Qi. Reliability analysis of fatigue strength of 2K-V reducer Cycloid pinwheel based on MATLAB[J]. Mining Machinery,2013,41(04):131-134.DOI:10.16816/j.cnki.ksjx.2013.04.035.
6. WEN Quan. Calculation technology of bearing life at the Cycloid pinwheel of 2K-V reducer[D]. North China University of Water Resources and Hydropower,2021.DOI:10.27144/d.cnki.ghbsc.2021.000046.
7. W.D. He,L.H.Li,J. Li. Analysis of the force on the Cycloid pinwheel in the RV drive for robots[J]. Journal of Dalian Railway Institute,1999(02):49-53.
8. HAN LINSHAN,WU YANGYANG. Error-based force analysis of typical parts of 2K-V reducer[J]. Hunan Agricultural Machinery,2014,41(03):103-106+130.
9. Zhenxu Zhu, Navy Dong, Linshan Han et al. Sensitivity analysis of error combination mode on transmission accuracy of RV-type reducer[J]. Mechanical Design,2008(10):69-73.DOI:10.13841/j.cnki.jxsj.2008.10.015.
10. HAN Linshan, SHEN Yunwen, DONG HUAI et al. Theoretical study on the dynamic transmission accuracy of 2K-V transmission[J]. Journal of Mechanical Engineering,2007(06):81-86.KuangHongjin. Problems and optimization strategies of core components of industrial robots in China[J]. Paper Equipment and Materials,2022,51(01):10-12.
11. ZihangZ,Qing Z. Study of contact load on Cycloid pinwheel of RV reducer based on MATLAB[J]. Journal of Physics: Conference Series,2022,2383(1).
12. RonghuaZ,JianxingZ,Zheng W. Study on transmission error and torsional stiffness of RV reducer under wear[J]. Journal of Mechanical Science and Technology,2022,36(8).
13. Qiao X,ZhangL,Chen C, et al. Study on transient contact performance of meshing transmission of cycloid gear and needle wheel in RV reducer[J]. The Journal of Engineering,2020,2020(14).
14. Li H,HouG,He W, et al. Simulation and analysis of contact state between cycloid gear and pin wheels in RV reducer[J]. Journal of Physics: Conference Series,2018,1074(1).
15. Zhang H Y,Wang Y X,He D W, et al. The Precise Measurement Technology for Manufacturing Error of Cycloidal Gear Used in RV Reducer[J]. Applied Mechanics and Materials,2017,4529.