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Impact of Manufacturing Errors on Tooth Surface Forces in RV Reducer Cycloid Pinwheels

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Authors' contributions

This work was carried out in collaboration among all authors. All authors read and approved the final manuscript.

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ABSTRACT

The mechanical calculation model of rotary-vertical (RV) reducer is established on the basis of considering displacement and stiffness. Without considering the manufacturing error, the parameters of the RV reducer are input into the model and solved by MATLAB calculation software. Comparing the solution results with the analysis results based on Hertz contact theory, the error is within a reasonable range, which verifies the reliability of the mechanical calculation model. The manufacturing errors of the RV reducer based on the high-precision measuring instrument were imported into the mechanical calculation model for solving, the results of the force analysis of the tooth surface of the cycloidal pinwheel considering the manufacturing errors were obtained. The maximum value of the load was 1713N, and the number of meshing teeth was 17 pairs. Compared with the case without manufacturing error, the maximum value of load increased by 89% and the number of meshed teeth decreased by two pairs. 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.

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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 [1]. 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 [2]. 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.

RV reducer cycloid pinwheel tooth surface force situation is affected by a series of factors such as manufacturing error, processing technology, lubrication effect. Among them, the manufacturing error directly affects the contact effect of the cycloid pinwheel tooth surface, so it has the greatest impact on the force on the cycloid pinwheel tooth surface [3]. The existing mechanical calculation model on the RV reducer cycloid pinwheel tooth surface force analysis have not considered the influence of manufacturing error, so the analysis results have a large gap with the actual situation [4]. The mechanical calculation model of RV reducer based on displacement and stiffness established in this paper introduces the manufacturing error into the analysis of the tooth surface force on the cycloidal pinwheel of RV reducer. The analysis results obtained are closer to the actual situation. The research results provide a theoretical basis for the design optimization and strength analysis of the RV reducer cycloid pinwheel. Of course, this paper only considers the impact of manufacturing errors on the cycloid pinwheel tooth surface force, did not take into account like the lubrication effect, processing technology and other secondary factors, the results of the analysis and the actual situation is still there is a certain gap to continue to study and improve [5].

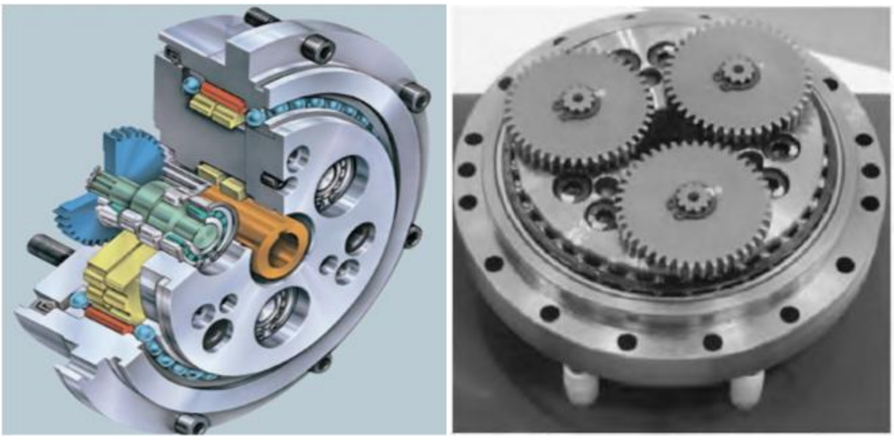

Fig. 1. RV reducer

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 [6].

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

3.1 Mechanical Modeling of RV Reducer

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 cannot meet the requirements, the need to establish the consideration of displacement, stiffness of the mechanical model, the RV reducer mechanical model is shown in Fig. 2 [7].

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 wheel $\mathrm{k_i}$, the bearing support stiffness between the crankshaft and the hole of the balance wheel shaft $\mathrm{k_{ji}}$, the bearing support stiffness between the crankshaft and the hole of the planetary carrier shaft $\mathrm{k_{bi}}$, the bearing support stiffness between the planetary carrier and the pinwheel $\mathrm{k_{ca}}$, and the meshing stiffness between the balance wheel and the pinion $\mathrm{k_{jk}}$ and the Central gear $\mathrm{k_s}$. The above stiffnesses are determined as follows: $\mathrm{k_i}$ is determined by using the bending deformation of the wheel teeth, and $\mathrm{k_s}$ is obtained by calculating the bending deformation of the Central gear shaft, while $\mathrm{k_s}$ is determined by using the bending deformation of the wheel teeth $\mathrm{k_{jk}}$ and the support stiffness of the Central gear$\mathrm{k_s}$. The above stiffnesses are determined as follows: $\mathrm{k_i}$ is determined by using the bending deformation of the wheel teeth, $\mathrm{k_s}$ is obtained by calculating the bending deformation of the Central gear shaft, and $\mathrm{k_{ji}}$, $\mathrm{k_{bi}}$, $\mathrm{k_{ca}}$, $\mathrm{k_{jk}}$ are determined by using the Palmgren formula.

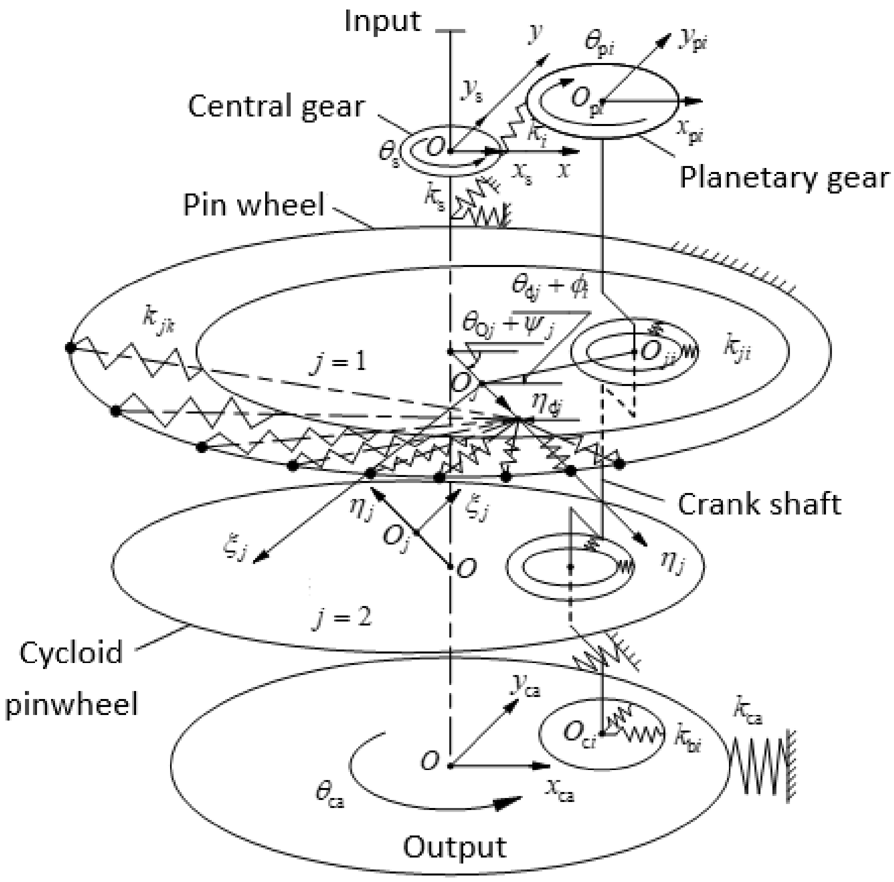

Fig. 2. Mechanical model of RV reducer

The next step is to determine the relevant static and dynamic coordinate systems. Taking the center O 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, O_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 (η_j, ξ_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, O_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:

$$
\begin{aligned}
& m_s\ddot{x}_s + F_{sx} + \textstyle\sum_{i=1}^{3}(F_i\cos A_i) = 0 \\
& m_s\ddot{y}_s + F_{sy} + \textstyle\sum_{i=1}^{3}(F_i\sin A_i) = 0 \\
& J_{os}\ddot{\theta}_{si} + \textstyle\sum_{i=1}^{3}(F_i R_{bs}) = \frac{T_{out}}{i_z} \\
& m_{sp}\left[\ddot{x}_{pi} - R_{dc}\omega_c^2\cos(\theta_c+\phi_i) - R_{dc}\ddot{\theta}_{ca}\sin(\theta_c+\phi_i) - 2\omega_c\dot{y}_{pi}\right] \\
& -F_i\cos A_i + \textstyle\sum_{j=1}^{2}F_{jix} + F_{cix} = 0 \\
& m_{sp}\left[\ddot{y}_{pi} - R_{dc}\omega_c^2\sin(\theta_c+\phi_i) + R_{dc}\ddot{\theta}_{ca}\cos(\theta_c+\phi_i) + 2\omega_c\dot{x}_{pi}\right] \\
& -F_i\sin A_i + \textstyle\sum_{j=1}^{2}F_{jj} + F_{cij} = 0 \\
& J_{Op}\ddot{\theta}_{pi} - F_iR_{bp} - e\textstyle\sum_{j=1}^{2}\left[F_{jxx}\sin(\theta_p+\psi_j) + F_{jy}\cos(\theta_p+\psi_j)\right] = 0 \\
& (i = 1,2,3) \\
& m_{br}\left[\ddot{\eta}_{dj}\cos(\theta_p+\psi_j) - e\omega_p^2\cos(\theta_p+\psi_j) - e\ddot{\theta}_{Qj}\sin(\theta_p+\psi_j)\right. \\
& \left.-2\omega_p\dot{\eta}_{dj}\sin(\theta_p+\psi_j)\right] - \textstyle\sum_{i=1}^{3}F_{jkx} + \textstyle\sum_{k=1}^{\frac{z_k}{2}}\left[F_{jk}\cos(\alpha_{jk}+\theta_p+\psi_j)\right] = 0 \\
& m_{br}\left[\ddot{\eta}_{dj}\sin(\theta_p+\psi_j) - e\omega_p^2\sin(\theta_p+\psi_j) + e\ddot{\theta}_{oj}\cos(\theta_p+\psi_j)\right. \\
& \left.+2\omega_p\dot{\eta}_{dj}\cos(\theta_p+\psi_j)\right] + \textstyle\sum_{i=1}^{3}F_{jij} + \textstyle\sum_{k=1}^{\frac{z_r}{2}}\left[F_{jk}\sin(\alpha_{jk}+\theta_p+\psi_j)\right] = 0 \\
& J_{0j}\ddot{\theta}_{dj} - \textstyle\sum_{k=1}^{\frac{z_r}{2}}\left[F_{jk}R_d\sin\alpha_{jk}\right] - R_{dc}\textstyle\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_{ca}\ddot{x}_{ca} - \textstyle\sum_{i=1}^{3}F_{cix} + F_{car} = 0 \\
& m_{ca}\ddot{y}_{ca} - \textstyle\sum_{i=1}^{3}F_{ciy} + F_{cay} = 0 \\
& J_0\ddot{\theta}_{ca} + R_{dc}\textstyle\sum_{i=1}^{3}\left[F_{ccx}\sin(\theta_c+\phi_i)\right] - R_{dc}\textstyle\sum_{i=1}^{3}\left[F_{ccy}\cos(\theta_c+\phi_i)\right] + T_{out} = 0
\end{aligned}
$$

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.

3.2 Model Reliability Analysis

RV-E is a 2-stage reduction gear; RV-110E is a particular RV-E model having some performance characteristics, The RV-110E type reducer is used as an example for analysis without considering manufacturing errors. Input its parameters into the established mechanical model and use MATLAB computing software to solve the model. Taking one week of output shaft rotation as a cycle, the force on the tooth surface of the cycloid pinwheel is shown in Fig. 3. It can be seen that the force on the tooth surface of the cycloid pinwheel is in a stable cycle during the operation of the RV reducer.

Taking one of the cycles of the force situation and the calculation results based on the Hertz contact theory, as shown in Fig. 4, the two calculation results are basically the same, which verifies the reliability of the mathematical model [8].

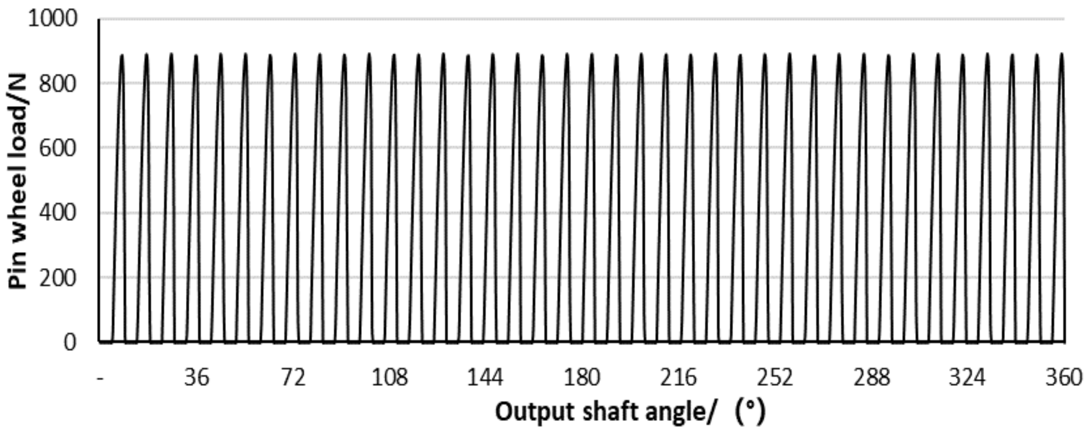

Fig. 3. Cycloid pinwheel loads for one week of output shaft operation without considering manufacturing errors

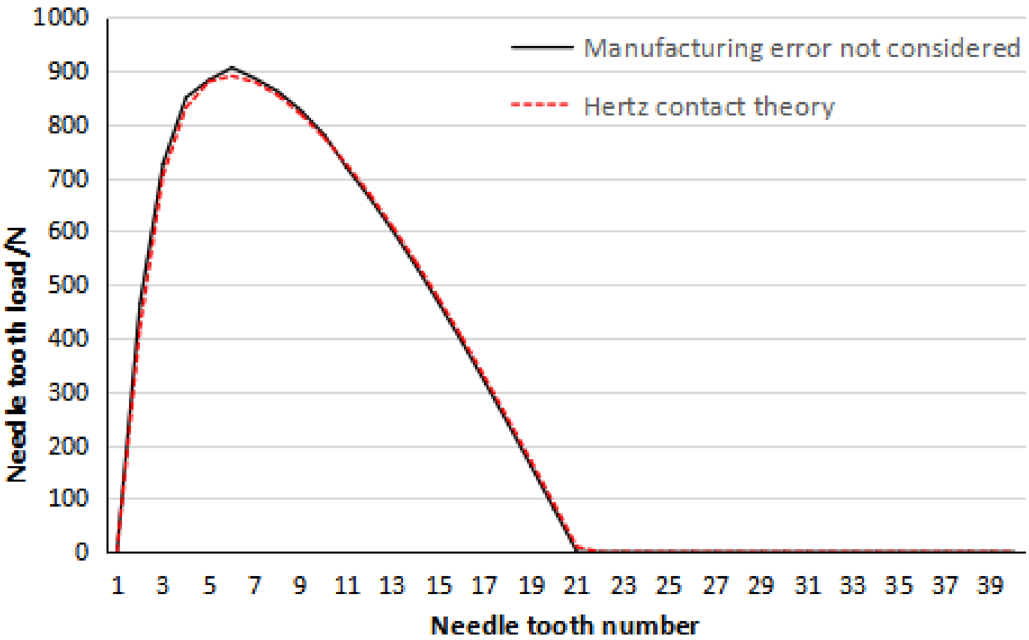

Fig. 4. Comparison of cycloid pinwheel tooth loads without considering manufacturing errors and based on Hertz contact theory

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

4.1 Error Analysis and Measurement

For RV reducer, the common manufacturing errors are tooth groove error, tooth pitch error, runout error, assembly tolerance and so on. The existence of manufacturing errors makes the actual working load of the cycloid pinwheel of RV reducer differ from the ideal working load [9]. Among the errors that have a greater impact on the load of the cycloid pinwheel are the eccentricity error of the crankshaft hole of the cycloid pinwheel, the eccentricity error of the planetary wheel base circle, the eccentricity error of the crankshaft eccentric cam and the eccentricity error of the planetary carrier crankshaft hole [10]. Based on the VMZ-R4540 type high-precision automatic image measuring instrument, the RV-110E type reducer of the various manufacturing errors measured, the results of the measurements are shown in the following tables.

4.2 Analysis of Results

The measured manufacturing errors are input into the established mechanical model and solved using MATLAB computing software. One cycle is taken as one week of rotation of the output shaft, in which the force state of the tooth surface of the cycloid pinwheel is in a stable cycle, as shown in Fig. 5. Analyzing the data of one of the cycles, it can be seen that the maximum value of the tooth surface load of the cycloid pinwheel considering the manufacturing error 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. 6, 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 [11-18].

Table 1. Planetary gear base circle eccentricity error

	Planetary gear 1	Planetary gear 2	Planetary gear 3
Error value /(μm)	33.2	44.4	46.0
Error angle /(°)	227.3	225.9	259.1

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

	Crankshaft bore 1		Crankshaft bore 2		Crankshaft bore 3	
	Error value /(μm)	Error angle /(°)	Error value /(μm)	Error angle /(°)	Error value /(μm)	Error angle /(°)
Cycloid pinwheel 1	6.4	270.0	14.9	270.0	4.5	90.0
Cycloid pinwheel 2	4.2	270.0	0.5	90.0	4.8	270.0

Table 3. Eccentricity error of crankshaft eccentric cams

	Crankshaft 1		Crankshaft 2		Crankshaft 3	
	Error value /(μm)	Error angle /(°)	Error value /(μm)	Error angle /(°)	Error value /(μm)	Error angle /(°)
Cycloid pinwheel 1	19.2	180.0	18.9	0.0	14.2	0.0
Cycloid pinwheel 2	19.2	180.0	38.3	295.6	14.2	0.0

Table 4. Eccentricity errors of planetary carrier crankshaft bores

	Crankshaft 1	Crankshaft 2	Crankshaft 3
Error value /(μm)	0.4	18.0	7.4
Error angle /(°)	0.0	80.6	112.9

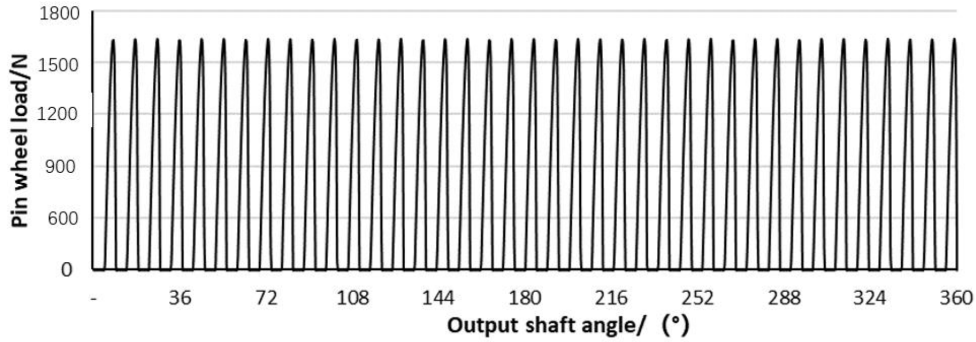

Fig. 5. Cycloid pinwheel loads for one week of output shaft operation considering manufacturing errors

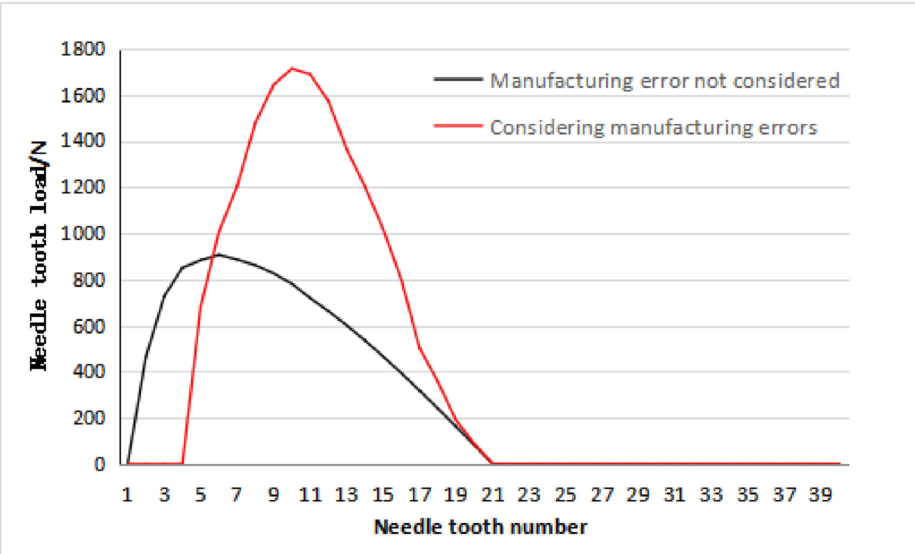

Fig. 6. Comparison of cycloid pinwheel tooth load considering manufacturing error and without manufacturing error

5. 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.

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COMPETING INTERESTS

Authors have declared that no competing interests exist.

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