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The Analysis of The Effect of System Parameters on the RV Reducer Dynamic Characteristics Based on ADAMS

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ABSTRACT: In order to ensure the motion accuracy, transmission efficiency and load carrying capacity of the robot with high precision RV reducer, under the condition of certain parameters, this paper analyzes the contact deformation relationship of the cycloidal gears in theory, the engaging force of the needle teeth, and then obtain the number of teeth when the cycloidal wheel and the needle wheel match wtih each other at the same time. The model of RV-80E reducer was established by using SolidWorks software, and then use the ADAMS to do the dynamics simulation . In this situation, The effect of the meshing force between cycloidal and needle teeth in RV reducer is explored when changing a single parameter such as short range coefficient、the radius of needle tooth's center circle、the radius of needle tooth、the teeth's number of cycloidal gear. Then find the best range of parameters to ensure the force between cycloidal and needle teeth .It provides useful conclusions for improving the performance of the overall transmission stability and carrying capacity of the gear unit. It also provides a reference for research methods on dynamics problems which use the virtual prototyping ,and have great significance for the production and design of RV reducer in the future.

Keywords: Gear reducer; Industrial robot; Meshing teeth; RV reducer ;Virtual prototype;

I. INTRODUCTION

The high precision of RV transmission which used in robots is different from the traditional transmission. It has the transmission ratio, high bearing capacity, bigger stiffness , as well as the higher precision of movement and smaller backlash, so it is widely used in the robot. Cycloid gear is the core part of the RV reducer. The shape of the tooth should ensure the ratio of Instantaneous transmission should be constant, ,the ransmission accuracy should be high .It also need to make sure the gap which between the cycloidal gear and the needle meet the requirement hysteresis gap[1] .At the same time, It sholud ensure that cycloidal teeth and needle teeth can be multi-tooth meshing one time.So,It can make the gear meshing stiffness increases, and the transmission can be more stable [2].

In this paper, the theoretical calculation is used to analyze the meshing teeth of the cycloidal and needle teeth and the force of the gear teeth after the modification. It provides a theoretical basis for the new method to improve the meshing stiffness and the stability of the transmission. Then, we use the virtual prototype to simulate and verify the influence of single factor on the meshing force of the teeth, and find out the optimal parameter range to increase the meshing force between teeth.

II. THE TECHNICAL PARAMETERS OF RV REDUCER

1.The technical parameters of RV reducer

The basic transmission mode of RV reducer is the planetary gear train, but it is different from the general reducer. It use the cycloid wheel drive, and the tooth profile is the external cycloidal profile. The picture below shows the structure of the RV reducer .The basic parameters of RV reducer showns in the table[3].

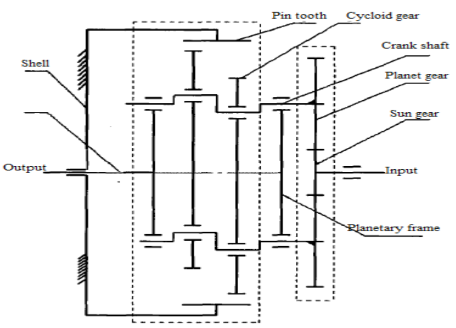
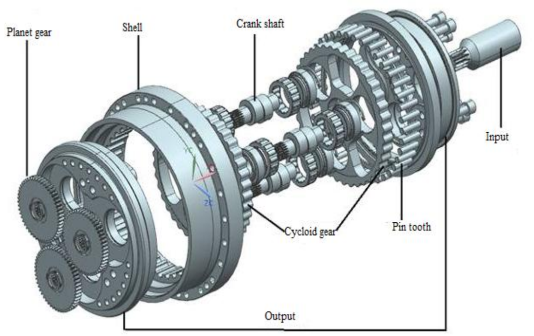


Diagram for RV-type cycloid-pin drive.



Actual picture for RV-type cycloid-pin drive.

RV reducer composition chart

Table.1 RV-80E reducer basic technical parameters

parameter name	Code	unit	data
Needle wheel center circle diameter	dp	mm	128
Needle diameter	Drp	mm	6
Eccentricity	a	mm	1.3
Number of cycloidal gears	Ze		40
Shift from the amount of repair	Δrp	mm	0.008
Isometric amount	Δrrp	mm	-0.002
Number of center teeth	Z1		21
Planetary gears	Z2		42
Modulus	M	Mm	2
pressure angle	α	(º)	20
Motor output power	P	kW	0.4
Motor output speed	n	r/min	525

II. TO EXPLORE THE NUMBER OF SIMULTANEOUS TEETH BY THEORY

2.1 The Selection of the Modification of Cycloid Wheel

In this study, the cycloidal gears were used to select the model from which choose the positive distance from the distance isometric way.The displacement of the distance was $\Delta r_p = 0.008$mm .The isometric amount is $\Delta r_{rp} = -0.002$mm,and no angle modification was used.The parametric equation of the geometric curve of the cycloidal wheel theory as fellow[4]:

$$x = r_p\left(\sin\varphi - \frac{K_1}{z_p}\sin Z_P\varphi\right) + r_{rp}\frac{K_1\sin z_p\varphi - \sin\varphi}{\sqrt{1 + K_1^2 - 2K_1\cos Z_c\varphi}}$$

$$y = r_p\left(\cos\varphi - \frac{K_1}{z_p}\cos Z_P\varphi\right) - r_{rp}\frac{-K_1\cos z_p\varphi + \cos\varphi}{\sqrt{1 + K_1^2 - 2K_1\cos Z_c\varphi}}$$

Where: K_1 is the short coefficient, $K_1 = aZ_P / r_p$ (a is the eccentricity); Z_P is the number of teeth of the needle wheel; Z_C is the number of teeth of the cycloidal wheel; r_p is the center circle radius of the pin; r_{rp} is the needle radius, Are the pin pin diameter.

Table.2 RV-80E cycloidal wheel basic technical parameters

Needle wheel center circle radius r_p/mm	Eccentriy a/mm	Pin radius r_{rp}/mm	Number of teeth
64	1.30	3	39

Table 2 is the cycloidal gear's tooth equation which after be modified when take the cycloidal gear parameters into the cycloidal gear formula modified when take the cycloidal gear parameters into the cycloidal gear formula,as follows:

$$x = 63.992\sin(0.025) - 1.3\sin\varphi + \frac{2.43943\sin\varphi - 3.002\sin(0.02\varphi)}{\sqrt{1.66032 - 1.62520\cos(0.975\varphi)}}$$

$$y = 63.992\cos(0.02\varphi) - 1.3\cos\varphi - \frac{\mathbf{-2.43943\cos\varphi + 3.002\cos(0.02\varphi)}}{\sqrt{\mathbf{1.66032 - 1.62520\cos(0.975\varphi)}}}$$

The Tooth shape of which be modified and the Standard tooth profile will be draw by MATLAB.Then a tooth of the tooth top part be taken out ,and be contrast with each other after be enlarged.The figure 2 below is the comparison as you can see.

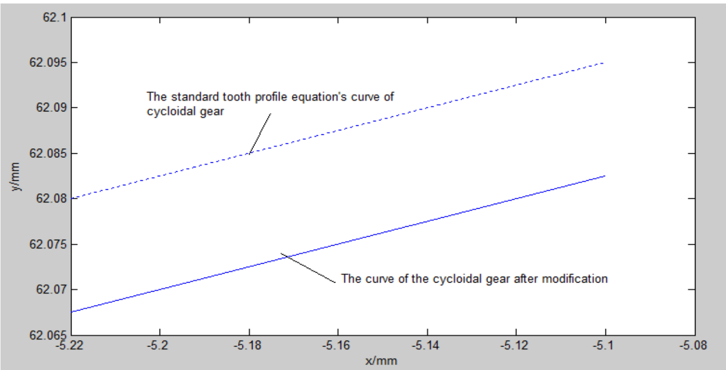


Fig.1 Comparison of standard tooth profile and trimming

In the figure above, the curve curve of the standard cycloidal gear is obviously larger than that of the modified cycloidal curve. So that there will be an intervening gap when the modification gear is engaged with the standard needle. The existence of the gap is conducive to compensation for manufacturing errors, to ensure lubrication that between the teeth, and reduce noise, which making the operation more stable.And it also change the number of teeth when the cycloidal gear math with needles at the same time.

2.2 Calculate the initial gap

The distribution of the initial gap of the cycloidal gear is as follows: The number of teeth can reach about half of the number of cycloidal gear teeth in theory at the same time when the standard cycloid or only the modified cycloid through the corner is meshed with the standard needle.The multi-tooth meshing situation is no longer exist and becomes a tooth and needle's contact without consider the elastic deformation of the role of parts after the equidistant distance repair or shift from the repair.The rest place of the cycloidal gear and the needle exists the the different size of initial gap. First use the theoretical calculation method. The cycloidal gear is fixed by the equidistant formations and the distance. The initial gap $\Delta\varphi_i$ which in the normal direction can be calculated as follows[6]:

$$\Delta\boldsymbol{\varphi}_{\mathbf{i}} = \frac{\Delta\mathbf{r}_{\mathbf{p}}(\mathbf{1} - \mathbf{K}_{\mathbf{1}}\mathbf{cos}\boldsymbol{\varphi}_{\mathbf{1}} - \sqrt{\mathbf{1} - \mathbf{K}_{\mathbf{1}}^{\mathbf{2}}}\mathbf{sin}\boldsymbol{\varphi}_{\mathbf{i}})}{\sqrt{\mathbf{1} + \mathbf{K}_{\mathbf{1}}^{\mathbf{2}} - \mathbf{2}\mathbf{K}_{\mathbf{1}}\mathbf{cos}\boldsymbol{\varphi}_{\mathbf{i}}}} + \Delta\mathbf{r}_{\mathbf{rp}}(\mathbf{1} - \frac{\mathbf{sin}\boldsymbol{\varphi}_{\mathbf{i}}}{\sqrt{\mathbf{1} + \mathbf{K}_{\mathbf{1}}^{\mathbf{2}} - \mathbf{2}\mathbf{K}_{\mathbf{1}}\mathbf{cos}\boldsymbol{\varphi}_{\mathbf{i}}}})$$

Where φ_i is the corner of the i-th needle relative to the boom,

When $\Delta\varphi_i = 0$, $\varphi_i = \varphi_0 = \cos^{-1} K_1$

where φ_0 is the angle at which the initial gap is zero, and only one pair of (or closest) $\varphi_0 = \cos^{-1}K_1$ Tooth meshing.

Take 1 ° for a step, make 0 ~ 180 °for a part (that is, the left half of the cycloid, as shown) to study. As a result of 40 needle tooth in total, so every 9 ° there is a needle. The initial meshing gap of each angle can be obtained from the formula of 2, but not every 1 ° can correspond to a needle.But this don't affect the next study. Calculate the initial meshing gap and then draw the image through MATLAB, as shown.

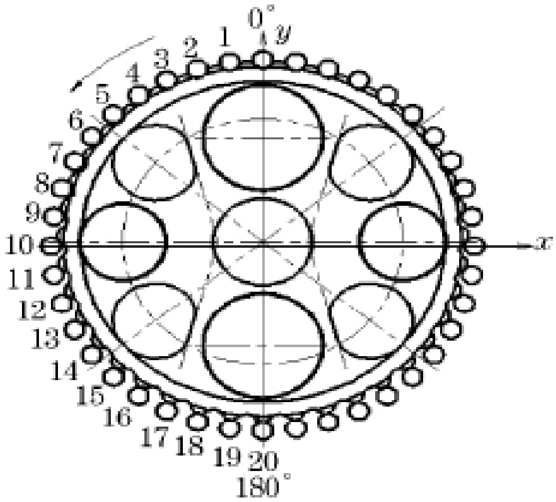


Fig.2 Needle and teeth cbart(1-20 for the pin tooth number)

2.3 Determine the number of teeth while the cycloidal gear and pin wheel match at the same time For the elastic deformation is already existed when the needle wheel engage with the cycloidal gear ,a total deformation in the direction of the common law of each engagement point of the cycloidal gear is existed or a certain displacement along the normal direction of the meshing points which to be engaged is existed.

When the total deformation or displacement is greater than the initial gap corresponding of the position, the teeth in this range will be engaged, on the other hand it will not enter the engagement. So, The number of engaging teeth can be calculated.

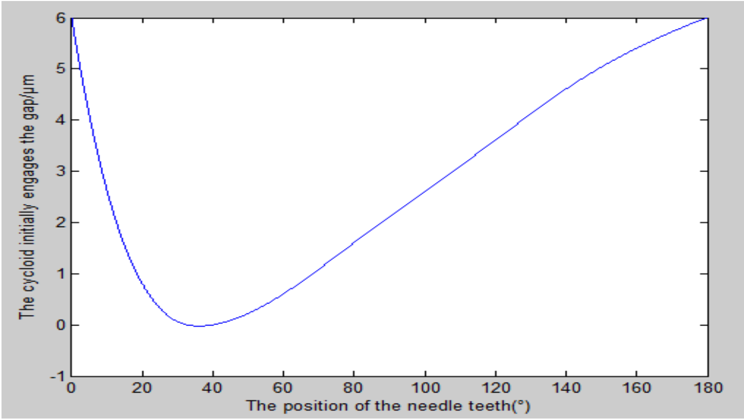


Fig.3 Cycloid initial teeth clearances curve

The torque of each piece of cycloid is T when set the transmission load,and cycloid wheel will appear contact deformation, needle pin will be bent deformation.The cycloidal gear tilts an angle of β.The displacement of the common law direction of the cycloidal gear's meshing point or the normal direction of the meshing point to be meshed is[7]: $\delta_i = L_i\beta$ ($i = 1,2, ..., z_p / 2$)

Where: β is the rotation angle of the cycloidal wheel due to the deformation of the part, rad; L_i is the distance between the normal line of the i-th needle contact point or the normal line of the meshing point to the center of the cycloidal gear O_c,mm. The distance between the contact deformation of the pair of cycloidal gear teeth and the pin tooth and the bending deformation of the needle pin is δ_{max}. And the distance between the common point of the meshing point and the center line O_c of the cycloidal gear is l_{max}.we can get the formula:

$$\boldsymbol{\delta_i} = l_i\beta = l\frac{\delta_{max}}{l_{max}} = \frac{l_1}{r_c}\delta_{max} = \frac{\sin\varphi_i}{\sqrt{1+K_1^2-2K_1\cos\varphi_i}}\delta_{max}$$

Combined with the various datas and the formula, you can calculate δi the Sum of the Needle gear'contact deformation and the Bending deformation of needle pin,and use the tool of MATLAB to draw it's figure as what is show below.In the figure, the sum of the contact deformation of the cycloidal of pinch gears and the bending deformation of needle pin is bigest where close to $\varphi_0 = \cos^{-1} K_1$, which means that the force is the largest, and the deformation and displacement are the largest,while the force on both sides of the meshing point gradually decreases. If the initial gap $\Delta\varphi_i$ in the normal direction of the modification i-th teeth that to be engaged and the total deformation of the cycloidal direction in the normal direction of the meshing point or the displacement δ_i in the normal direction of the meshing point are being draw in one picture,you can get the picture above.

In the above figure, two curves have two intersections at 18 ° and 150.02 °, respectively. According to the principle of the cycloidal wheel and the gear wheel, the cycloidal gear between 3.18 ° and 150.02 ° is engaged with the pin tooth. Since the pin wheel has 40 teeth, there is one wheel every 9 ° teeth, so there are 16 teeth in the range of teeth, so 16 cycloid teeth and needle teeth in engagement with the state at the same time.

III. ANALYSIS OF ENGAGEMENT FORCE OF GEAR

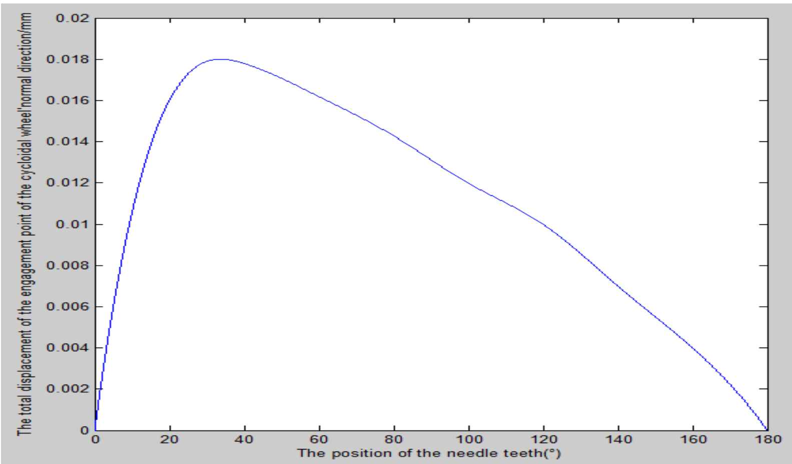


Fig.4 The total displacement curve of the cycloidal wheel following the normal direction of the meshing point

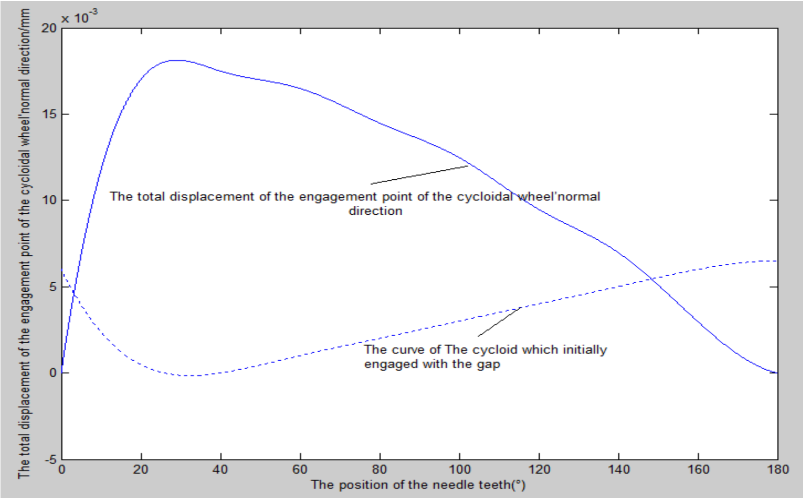


Fig.5 The total displacement of the initial meshing gap and the load direction along the normal direction of the meshing point

Force analysis of the engagement of Meshing Cycloid and Needle :At the time of idling, the modified cycloid is in contact with (or closest) to $\varphi_0 = \cos^{-1} K_1$, and the remaining teeth and pin teeth have an initial gap of different size in the normal direction of the meshing point . This difference is particularly pronounced when the amount of deformation is large. Therefore, it can not be assumed that the force of Fi is linearly proportional to the total deformation δi = liβ. It can only be assumed that F_i is linearly proportional to δ_i-$\Delta\varphi_i$. This assumption takes into account the initial role of the initial gap 、the elastic deformation and bending deformation. So it used for the force analysis to be accurate enough.

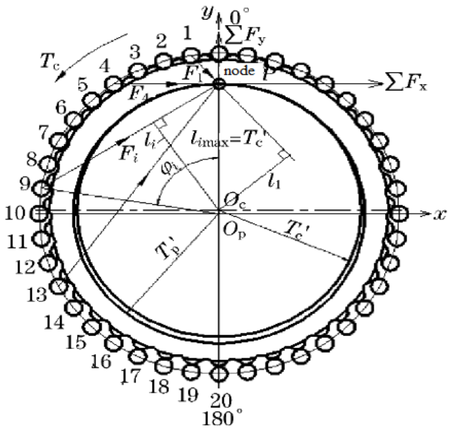


Fig.6 The force diagram of the cycloidal wheel

Fig.6 1 is the force diagram of the cycloid .Due to the simultaneous engagement of the teeth are so many , so only marked a few of the engaging force to indicate. In the respective teeth of which the force is simultaneously engaged, the i-th tooth force F_i is expressed by the following equation: $F_i = (\delta_i - \Delta\varphi_i) F_{max} / \delta_{max}$[7]. Since the teeth at (or closest) $\varphi_0 = \cos_{-1} K_1$ are in contact with the force first, it is clear that the teeth are the most stressed in the teeth at the same time, so that the tooth is forced by F_{max}.

Using the method of computer iteration to calculate the size of each tooth's force, then draw the Fig.7 below with MATLAB.In the figure below, the force at the point close to $\varphi_0 = \cos_{-1}K_1$ is the largest, and the operator is well with the theory.

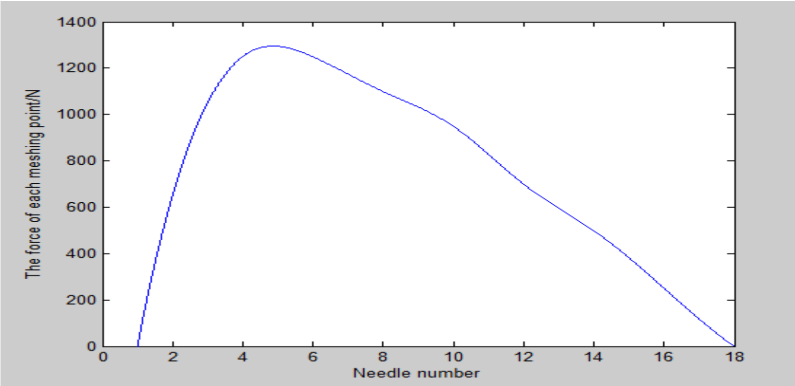


Fig.7 Tooth forces graph on cycloid

IV. THE ESTABLISHMENT OF VIRTUAL PROTOTYPE OF CYCLOID

4.1 The establishment and simplification of the solid model

Build the 3D Solid Modeling with SolidWorks.According to the equation of the cycloid wheel to establish the cycloidal gear,and then build the other three-dimensional solid modelings of the reducer such as the needle wheel, the input shaft, the flange, the planetary frame and the shell .At last,assemble them into assemblies.In order to save storage space, improve the speed of operation, the model was simplified.For example: using the cylindrical instead of involute gear and crankshaft as a whole;Flanges are integrated with planets as a whole;Remove the retaining ring, bearings, bolts and other unrelated parts;Remove all chamfers that do not affect the drive.After the model is established, it is necessary to carry out part interference check, including static interference detection and dynamic interference detection.This simulation has been adjusted to no any interference.

4.2The model's import and definition

The method of simulation analysis:put the SolidWorks simplified assembly model saved as X-T format,and then through the ADAMS Import command to import them to ADAMS software for motion simulation and analysis.It has been assembled components in SolidWorks ,so the location of the various components have been adjusted already. It only need to adjust the color and material properties of each component in ADAMS . The material settings are shown in Table 3.

Table .3 RV reducer parts of the material properties

name	material	Elastic Modulus /(N · m^{-2})	Poisson's ratio	Mass density/ ($K_g.m^{-3}$)
Cycloid gear	20CrMnMo	2.07×10^{11}	0.254	7.87×10^{3}
Input shaft	20CrMoMo	2.07×10^{11}	0.254	7.87×10^{3}
Crankshaft	20CrMOMo	2.07×10^{11}	0.254	7.87×10^{3}
Needle roller	GCrl5	2.08×10^{11}	0.300	7.80×10^{3}
shell	QT500-7	1.68×10^{11}	0.240	7.25×10^{3}
Planetary frame	ZG65Mn	1.98×10^{11}	0.230	7.85×10^{3}
The Flange	ZG65Mn	1.98×10^{11}	0.230	7.85×10^{3}

4.3 Add the Constraints

The type of constraint and the contact type are shown in the following two tables, and two gear pairs are set for the input shaft and two spur gears[8].

Table .4 The type of constraint for RV reducer simulation

Constraint type and number	part 1	part 2
Fixed vice（40）	Needle roller	ground
Rotateing pair（1）	Input shaft	ground
Rotateing pair（1）	Planetary frame	ground
Fixed vice（1）	shell	ground
Rotateing pair（1）	Input shaft	Planetary frame
Rotateing pair（2）	Two crankshafts	Planetary frame
Rotateing pair（2）	Two crankshafts	Left cycloid
Rotateing pair（2）	Two crankshafts	Right cycloid
Contact vice（40）	Needle roller	Left cycloid
Contact vice（40）	Needle roller	Right cycloid
Contact vice（2）	Left and right cycloid	Planetary frame

RV reducer simulation of the contact setting type

This time import forty-seven rigid bodies as a total, fifty bounds , eighty-two contacts, one rotation drive, two gear pairs , and be consistent with the model self-test results. We Set the load torque to 572N · m, set the rotational drive input speed of 525r / min, define the input speed function for the slow growth curve within 1s, and to within 1s constant and F (time) = 3150d * time * 0,0.1,1), define the load torque for step (time, 1.0,0,1.5,57200). The simulation time is 5s, and 500 steps in one simulation step.

V. INFLUENCE OF SYSTEM PARAMETERS ON DYNAMIC CHARACTERISTICS OF RV REDUCER

After understanding some of the dynamic response of RV reducer, it is necessary to study the effect of RV reducer system parameters on the dynamic characteristics of RV reducer. Based on the model of dynamic , the system parameters of RV reducer are analyzed. Here RV-80E as the object for study, in which RV-80E system parameters as: R (needle tooth center circle radius) = 70mm. P(Pitch ratio) = 3 mm, Z_b (number of needle teeth) = 40, Zg (cycloidal gear number) = 39, e (eccentricity) = 1.5 mm for short coefficient, φ_{max} is the angle for cycloidal gear Symmetrical axis and cycloidal profile of the starting line .In order to investigate the influence of a certain parameter on the dynamic characteristics of the system, to explore the influence of the dynamic characteristics when one of parameters is changed without changing the other parameters. According to the analysis of the characteristics of the two-stage drive, the output characteristics of the RV reducer are mainly determined by the transmission characteristics of the second-stage drive. Therefore, the meshing force of the cycloidal gear and the needle in the second stage is the research goal, and make a comprehensive study of RV reducer system parameters on the dynamic characteristics of the system. System parameter analysis includes: short coefficient analysis, needle center distribution circle analysis, needle radius analysis, cycloidal gear tooth number analysis.

5.1 The analysis of short coefficient

The short coefficient is an important parameter of the RV drive in the RV reducer, which is related to the quality of the cycloidal gear transmission quality.And it matters whether it can avoid the "root cutting" and other issues. The short coefficient have large effects on the RV reducer dynamic transmission performance .The calculated of The short coefficient as: $K = \frac{eZ_b}{R_Z(Z_b - Z_g)}$

When R_Z = 70mm, r_z = 3mm, Z_b = 40, Z_g = 39 be stable, the size of the short coefficient depends on the size of the eccentricity, so that e is equal to 0.6-1.6mm.It can take a value of 0.1mm ,so the corresponding short coefficient K equal 0.34,0.4,0.46,0.51,0.57,0.63,0.68,0.74,0.8,0.86,0.9 respectively. The kinetic model of the RV reducer is used to simulate and get the average meshing force of the cycloidal and needle teeth with different short coefficient. Using the spline command of MATLAB to draw the points of the average meshing force, and get curve of the the engaging force of the needle and the change of the shortness coefficient (eccentricity),as is how in Fig.8

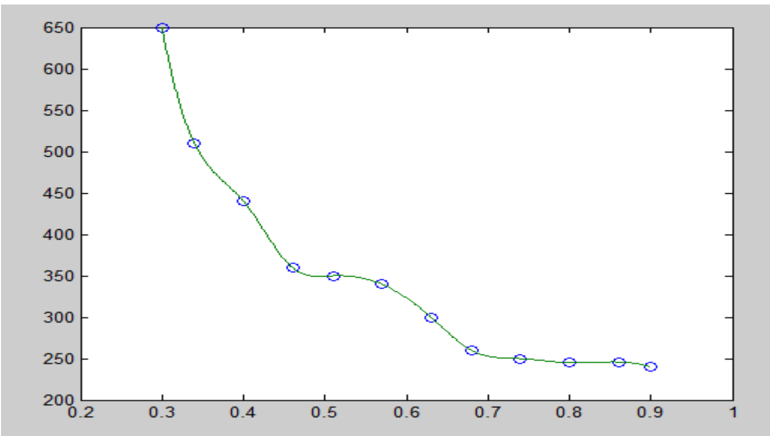


Fig.8 Curve of the meshing force between the cycloid and the pin and the segment coefficient (eccentricity)

The meshing force between the cycloidal and the needle teeth decreases with the increase of the short coefficient. The maximum engaging force is 650N, which is 2.6 times of the minimum engaging force. This is because with the increase of the short coefficient, the deviation also increases, so that the radius of the roll increases. As the transmission torque is constant, the meshing force between the cycloidal wheel and the needle will be reduced. According to the Hertz theory, the maximum contact of the contact area should be：

$$\sigma_{H_{max}} = \sqrt{\frac{1}{2\,\Pi\,(1-v^2)}\frac{FE}{b\rho}}$$

Where F is the normal engaging force of cycloidal and needle teeth, P is the radius of curvature, V is the Poisson's ratio, E is the integrated elastic modulus, and b is the contact width. It can be seen from the figure that when the short coefficient K is small, the meshing force F between the cycloidal and the needle teeth is large, and the contact stress when the two are engaged is also large, so that the engaging rigidity is reduced, and it's easy to cause cycloid and needle tooth's damage by wear and fatigue .

However, when K> 0.7 , the reduction of the meshing force between the cycloid and the needle be decreased. And when K > 0.8, the convex portion of the cycloidal wheel is much larger than its concave portion, the radius of curvature will be greatly reduced according to the formula of cycloidal gear tooth profile . When the K> 0.8 , the meshing force F decreases at a smaller magnitude, and the magnitude of the integrated curvature radius P decreases. According to the Hertz's formula, the contact stress between the cycloidal gear and the pin tooth is increased .It also cause meshing damage between the cycloid and pin tooth .In addition, the short coefficient will lead to be rotation of the cycloid.It also hard for the RV drive, so a reasonable short coefficient range should be K = 0.6-0.75.

5.2 Analysis the Circle Radius of the Center of Needle

RV reducer size depends mainly on the needle tooth is also distributed in the radius of the radius RZ, so keep the other parameters unchanged, separate study of the needle tooth is also distributed circle radius of the impact of RV transmission. That is, keep Zb = 40, Zg = 39, e = 1.5mm, rz = 3mm, so that the center circle of the needle tooth radius is equal to 66-90mm, which take a value every 2mm. Using the dynamic model of the RV reducer to simulate, the results obtained as shown in Figure 2, cycloidal and needle tooth meshing force with the needle tooth is also distributed in the radius of the relationship between the changes..

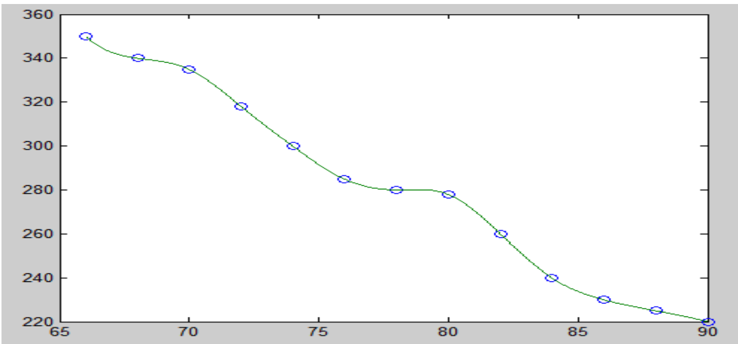


Fig.9 Curve of the radius of the cycloidal gear and the meshing force of the needle and the center of the It can be seen from Fig. 9 that the meshing force between the cycloidal and needle teeth decreases while the increase of R_Z, and the maximum engaging force is about 1.6 times of the minimum engaging force.When the RZ increases, the radius of the cycloid will increase, the distance from the point of engagement to the node increase, that is, the tooth profile of cycloidal gear will thicker, the engagement stiffness of the cycloidal wheel and the pin increases , And the maximum stress in the meshing process is educed .

However, with the further increase of R_Z, as shown in the figure when R_Z> 85 or so, the cycloidal force between the pulley and the needle is reduced slowly.It can be seen, when do the design of RV reducer, it should be based on the size of the load, and the characteristics of the material that it used. The needle tooth center distribution radius should be designed as large as possible , so that the engaging stress in the process between the cycloidal and needle will be smaller. The teeth in the process of will be smaller. This design is conducive to the RV reducer drive. However, when the radius of the center of the pin tooth is too large, the overall shape of the RV reducer becomes large .It will be the waste of material, and reduce the transmission efficiency of RV reducer. As a result ,it lost the advantage of small size and high transmission.

5.3 The analysis of needle radius

In order to explore the influence of the needle radius on the RV reducer transmission, set some parameters be stable such as: R_Z = 70mm, Z_b = 40, Z_g = 39, and e = 1.5mm.The size of the needle radius rz value in the range of 1.6 to 4.0mm, take a value every 0.2mm. And then use the ADAMS dynamic model to do simulation. The results are shown in Figure 3 which draw the relationship's curve of the engagement force between cycloidal gear and the needle tooth when change the needle tooth's radius .

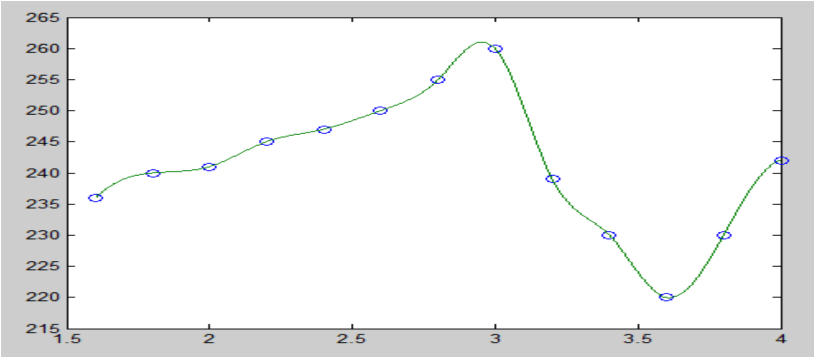


Fig.10 Curve of the meshing force between the cycloid and the needle and the radius of the needle

It can be seen from Fig. 10 that there is a serious non-linear relationship of the meshing force between the cycloidal gear and needle tooth and the needle radius rz. The engaging force between the cycloid and the needle is beginning to increase with the increase of rz; but when rz> = 3mm, the engaging force starts to decrease abruptly; and when rz> = 3.6mm,the engaging force began to increase rapidly. On the whole, the amplitude of the meshing force is not changed greatly, and the variation range is only 220N to 260N. The variation range is 18.2%. As the rz increases, the combined radius of curvature increases in the transmission process,but the contact stress decreased. And the meshing stiffness of the cycloidal and pinch teeth increases, which is favorable for the transmission of the RV reducer .From the overall point of view, to increase the needle radius will improve the performance of the RV drive.However, in the process to design RV reducer, the maximum needle radius can not exceed the minimum radius of curvature of the cycloid, otherwise it will cause "root cutting" phenomenon.In addition, the needle is inserted on the needle plate evenly. Too large needle radius will cause the gap between the needles too small when installa it on the cycloidal gear.To cause the stiffnes of the needle plate be reduced ,and affect the stability of the RV drive . So in the guarantee is not "root cutting" and the needle tooth installation gap is large enough ,to maximize the needle radius is conducive to the RV reducer drive.

5.4 Analysis of Gear Number of Cycloid

The number of cycloidal gears Z_g is one of the important parameters when design the RV reducer, which is related to the transmission ratio of RV reducer and the meshing performance of cycloidal gear and pin wheel.And it directly affects the transmission performance of RV reducer.Here,we focus on the effects of the dynamic performance of RV reducer when we change the number of cycloid gears .Set some parameters be stable,such as:R_Z=70mm，r_z=3mm，e=1.5mm,and then make Zg equal to 19-39 for every two teeth to take a value.As a result of this study is a differential trochoidal pinch wheel meshing, so the number of the needle in the corresponding cycloid is the number of the cycloidal gear plus one.The dynamic model of the RV reducer is used to simulate the relationship between the cycloidal force of the cycloidal gear and needle tooth and the number's change of cycloidal gear , as it showns in Fig. 11

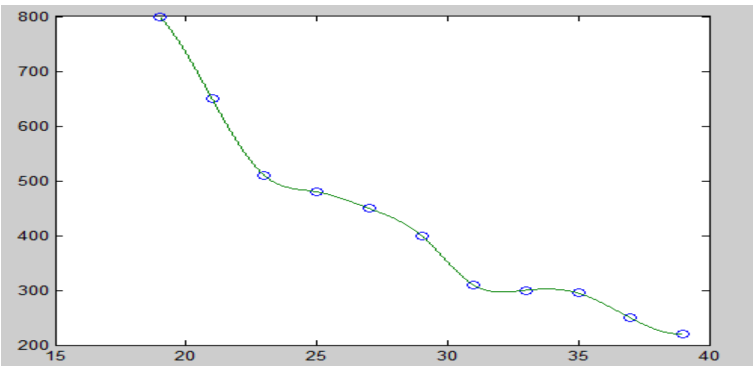


Fig.11 Curve of the engagement force between cycloidal and needle teeth and the number of teeth of cycloidal gear

It can be seen from Fig. 11 that the meshing force between the cycloidal gear and the needle is more sensitive to the change of the number of teeth of the cycloidal wheel. The maximum engaging force is 800N and the minimum engaging force is 220N. The size of the meshing force between the cycloid and the needle wheel decreases as the number of teeth of the cycloidal gear increases.This is mainly due to the increase of the number of teeth of the cycloid make the number of engagement participating teeth increased.In the case of the constant transmission torque, the average engagement force will be reduced.From the engaging arm (the distance from the engaging node to the center) r1 = eZ_g, with the increase of Z_g, the engaging arm will increase, and the contact stress between the cycloidal gear and the needle tooth will be reduced.Therefore, in the case of other parameters of the system unchanged, increase the number of teeth of the cycloid gear will be good for RV drive .On the other hand, the excessive number of cycloid tooth will cause "root cutting" phenomenon, and too many cycloid tooth will produce too much needles,and it may lead to situation that the excessive needle tooth can not be installed on the needle plate.

And when $Z_g >= 30$, the amplitude of the meshing force between the cycloidal gear and the needle teeth decreases.According to the characteristics of cycloidal profile curve, with the increase of the number of cycloidal gear teeth, the convex part of the cycloidal tooth profile will gradually increase, the concave part gradually decreases,the equivalent radius of curvature of the cycloidal gear will reduced.According to the formula, the contact stress during the meshing process will increase,and will cause the damage of the cycloidal gear and the needle tooth.It is not conducive to the RV reducer transmision. Therefore, the number of teeth of the cycloidal gear should be appropriately increased after considering many previous factors [10].

Conclusion

1.The RV reducer which used in the robot requires multi-tooth meshing, but also has the reasonable meshing clearance to compensate for the error and to meet the lubrication conditions. Therefore, it is very important to do the research on the number of teeth which are working at the same time.

2.In this paper, the meshing force and the number of meshing teeth which drive in cycloidal gear are calculated and analyzed at the view of the theoretical calculation

3.In this paper, the virtual prototype is established by ADAMS, and the influence of these parameters on the transmission characteristics of RV reducer is discussed such as short range coefficient、 the radius of needle tooth’s center circle、 the radius of needle tooth、 the teeth’s number of cycloidal gear,by using the dynamic model of RV reducer. Some important conclusions be founded. The influence of the transmission characteristics of the RV reducer is as follows: the teeth’s number of cycloidal gear、short range coefficient、 the radius of needle tooth’s center circle and the least is the radius of needle tooth. In order to increase the engaging force, the teeth’s number of cycloidal gear should be appropriate increased under the condition of avoided the "root cutting". According to the size of the transmission torque, the radius of needle tooth’s center circle should be appropriate increased to improve it. The reasonable short coefficient range should be 0.6 to0.75.

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