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Research Article

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Research on transmission stability of rotation roller screw for rail transit vehicles door

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ABSTRACT

Taking transmission system of rail transit vehicles door as research object, virtual simulation and transmission stability of rotation roller screw are carried out. Firstly, modal analysis is made to get the natural frequency and vibration mode. Then with mnf files imported, rigid-flexible mixed virtual prototype of door system is established. Based on the vibration analysis of flexible screw, transmission stabilities of rail transit vehicles door are obtained. Finally, velocity profiles are optimized to favor smooth start-up and braking. The presented research can provide a design reference for rotation roller screw for rail transit vehicles door.

Keywords: transmission stability, rail transit vehicles, door system, modal analysis, speed profiles

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INTRODUCTION

Transmission screw, as common transmission mechanism, is widely used in machinery tools, automotive, aerospace, weapons and so on [1,2]. Scroll screw can be classified into ball screw and roller screw. Ball screw transmission was invented in 1874. In China, studies on ball screw began from late 1950s [3]. So far, researches on ball screw transmission from home and abroad have been extended and now is widely used in CNC and other application fields. Roller screw transmission gas been developed in recent years with prior usage and manufacturing performance. Roller screw transmission can be divided into rotation and planetary roller. Roller screw transmission was studied relatively late. Planetary roller screw was firstly invented in 1942 followed by a series of patented roller screw vice, including differential, ring-bearings and rolling-ball roller screw.

Study of roller screw transmission in China began from the 1990s. The earliest research was from Huazhong University of Science & Technology. Ji Qianzhong et al studied the basic theory of roller screw transmission as well as ball screw parameter selection method, which can lay good foundations for roller screw design, lubrication analysis & design and life time analysis. Studied on static rigidity of roller screw were followed in-depth [4]. Yang Baozhe analyzed the running stability, modal analysis and load rating of roller screw for the following optimal structure design and production [5]. Several structural parameters of planetary roller screw were optimized by the multiplication-division and simulated annealing algorithm [6]. A kinematic model to predict the axial migration of the rollers relative to the nut in the planetary roller screw mechanism (PRSM) was developed [7]. A stiffness model for the PRSM was developed as well as the dynamics of the PRSM, including an effective inertia of the mechanism, the constraint force on the spur/ring gear pair, the steady-state angular velocities, the screw/roller slip velocity, and efficiency of the mechanism [8]. The direct stiffness method was used to construct a stiffness model of the roller screw mechanism, which models the entire roller screw mechanism as a large spring system composed of individual springs representing the various compliances [9].

At present, researches on rotation roller screw are seldom reported. The presented research seeks to research on the transmission stability of rotation roller screw in order to lay good foundation for engineering practice and promote the use of rotation roller screw.

EXPERIMENTAL SECTION

Modal analysis of crew structure

Modal analysis is used to analyze the vibration characteristics of the structure to obtain the natural frequencies and vibration mode, which is the basis of kinetic analysis. According to the results of modal analysis, design engineers for structure design can make natural frequency avoid the external excitation frequency. Therefore, before analyzing the transmission stability of door system, it is of great need to carry out modal analysis of screw. The supporting way has great importance to modal analysis. The supporting of screw is as shown in Figure 1.

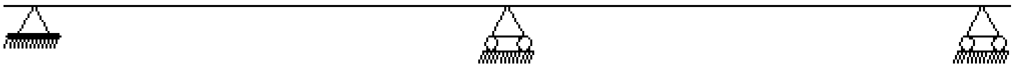
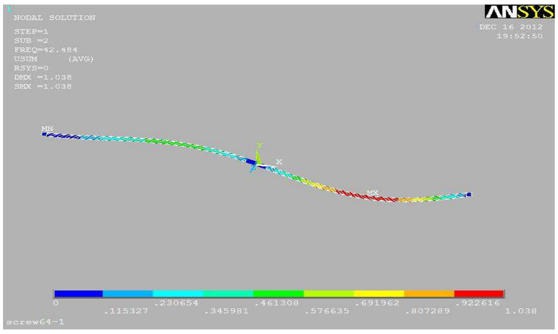
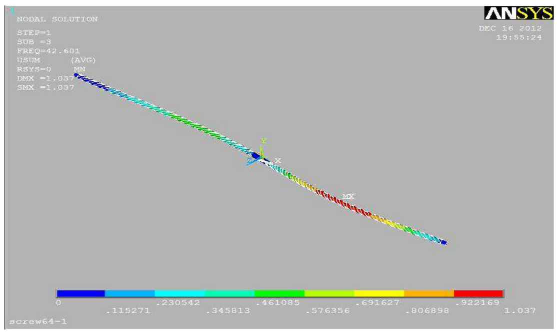


Figure.1. Supporting way of screw

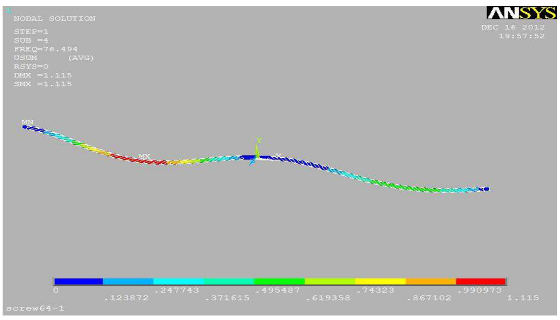
Constraints Definitions are as follows:
1) Left side: Fix movement freedom of X, Y, Z and rotational freedom of Y, Z;
2) The middle and right side: Fixed direction movement freedom of Y, Z;
Modal evaluation mainly depends on lower modes. The below are the first four modal analysis results of screw below based on ANSYS.



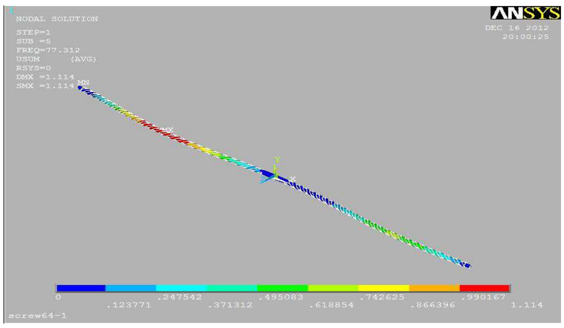
a) 1-order modal analysis



b) 2-order modal analysis



c) 3-order modal analysis



d) 4-order modal analysis

Figure.2. The first four modal analysis results

Table.1. Natural frequency of the first four modal analysis

Modal analysis	1-order	2-order	3-order	4-order
Natural frequency	42.48	42.60	76.49	77.31

As can be seen from the above results of modal analysis, 1-order frequency of screw is 42.48HZ. The vibration mode of the first four modal analysis is bending plus torsion in XY and XZ plane. It is obvious that smaller natural frequency will result in larger vibration during operations.

Transmission stability analysis of rotation roller screw
Screw belongs to slender with small natural frequency, which is very important to smooth motion of door system. Rigid-flexible coupling analysis of ADAMS software [10] is utilized to study the transmission stability of door system as shown in Figure 3.

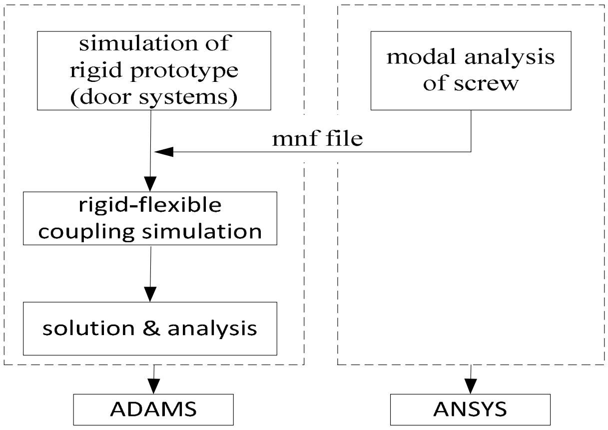
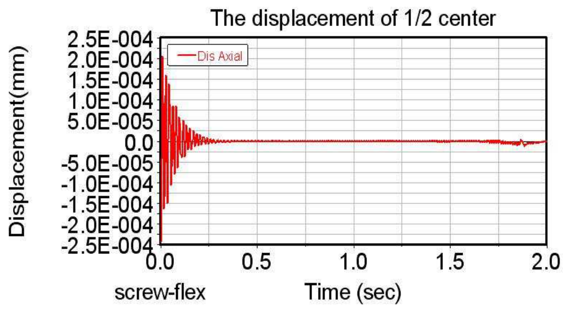


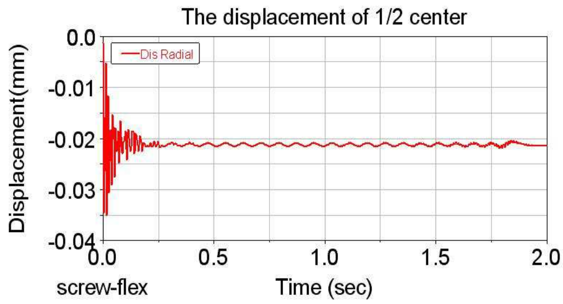
Figure.3. Analysis procedure of transmission stability

The key component with the greatest impacts of transmission stability on door system is the screw. So during rigid-flexible coupling simulation[11], the screw is considered as flexible, and the rest as rigid. By accessing the motion of screw node, the analysis results are as shown in Figure 4 and 5.

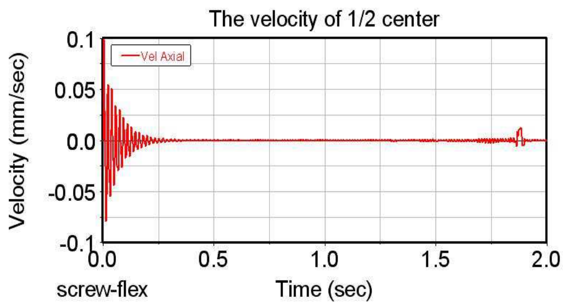
1) Motion of 1/2 central node



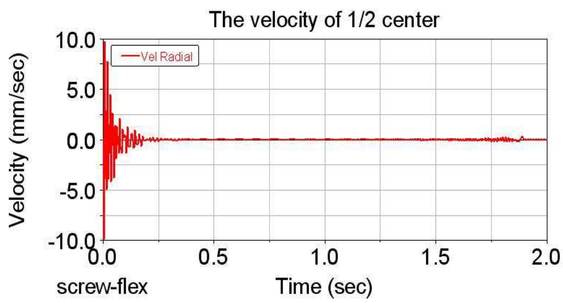
a) vibration of axial displacement



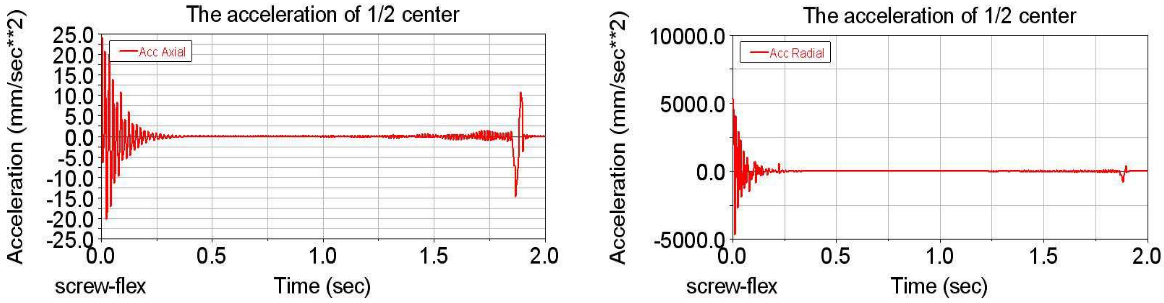
b) vibration of radial displacement



c) vibration of axial speed



d) vibration of radial speed



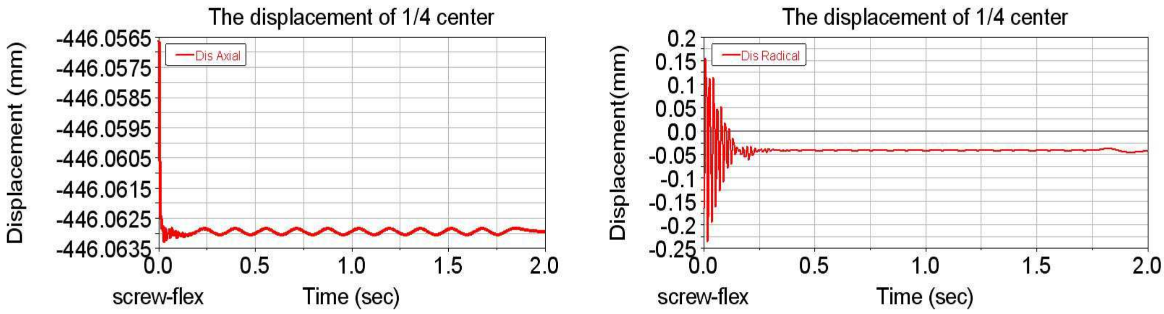
e) vibration of axial acceleration **f) vibration of radial acceleration**

Figure.4. Screw vibration of 1/2center

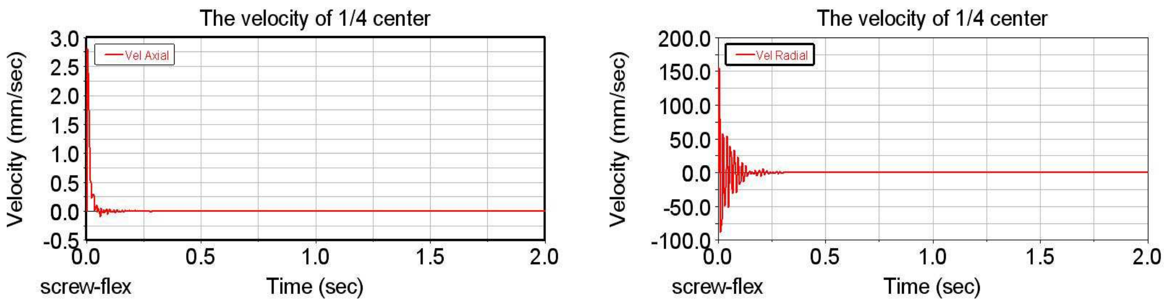
It can be concluded from Figure 4 that centroid vibration of screw during operations is small. Axial displacement amplitude is about 10^{-4}mm, velocity amplitude of 0.1mm/s or so, and acceleration amplitude of about 50 mm/s^2. Radial displacement amplitude is about 0.05mm, velocity amplitude 20mm/s or so, acceleration amplitude is about 10^4 mm/s^2. It is obvious that displacement of 1/2 center is smaller due to the support in the centroid of screw.

2) Motion of 1/4 central node

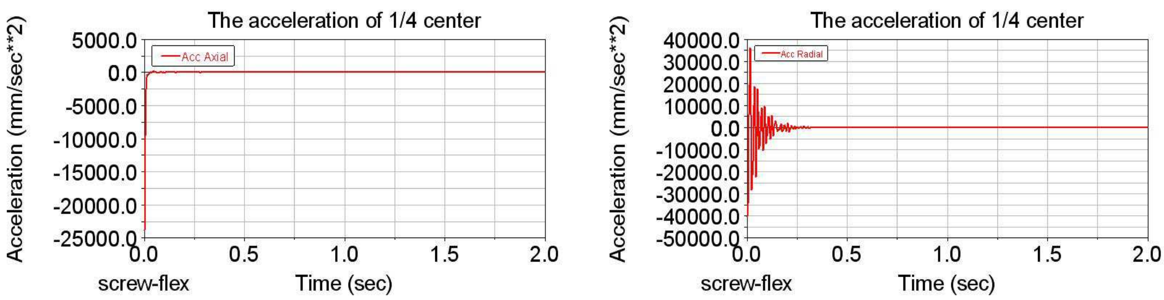
During operations, the biggest vibrations occur in the 1/4 and 3/4 position of screw. The vibration of 1/4 screw are measured as shown in Figure 5.



a) vibration of axial displacement **b) vibration of radial displacement**



c) vibration of axial speed **d) vibration of radial speed**



e) vibration of axial acceleration **f) vibration of radial acceleration**

Figure.5. Screw vibration of 1/4 center

It can be concluded from Figure 4 that vibration of 1/4 screw during operations is big. Axial displacement amplitude is about 0.07mm, velocity amplitude of 3mm/s or so, and acceleration amplitude of about 2.5×10^4 mm/s^2. Radial displacement amplitude is about 0.4 mm, velocity amplitude 250 mm/s or so, acceleration amplitude is about 6.5×10^4 mm/s^2.

It is obvious that the vibration of screw during operations is relatively big due to the smaller due to the slender structure. The length of transmission structure needs to shorten for transmission stability improvement.

RESULTS AND DISCUSSION

The durations of accelerated switch and decelerated braking of door system are very short with big acceleration and deceleration. It is of great importance for door system whether to switch and brake smoothly. Therefore, it is significant to select a reasonable velocity profile. During operations, rotary drive is applied, the input member of which is the screw. So, angular velocity and angular acceleration are considered as profile optimization objectives.

1) Analysis and simulation of constant acceleration start-up

The velocity profile of constant acceleration start-up[12,13] is illustrated in Figure 6.

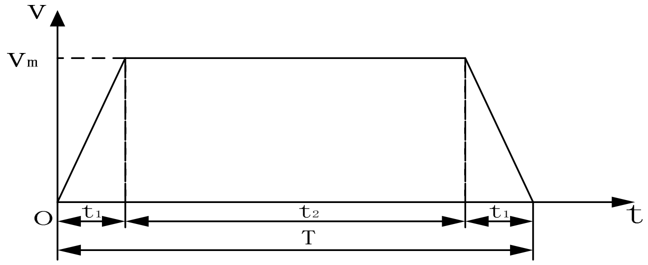


Figure.6. Velocity profile of uniform acceleration start-up

Let the acceleration denoted by *a* and the hodometer by *H*, then:

$$H = at_1^2 + at_1t_2$$
$$T = 2t_1 + t_2$$

Motion settings are as: Type-Velocity; Function-IF(time-0.25 : -10400d*time , -2600d , IF(time-1.75 : -2600d , -2600d , IF(time-2 : 10400d*time-20800d , 0d , 0d))). The simulation result is shown in Figure 7.

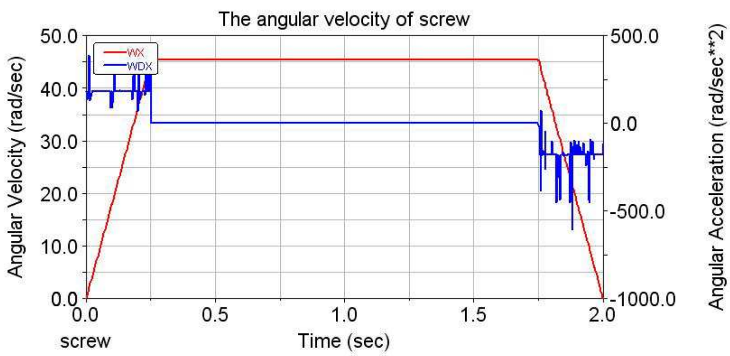


Figure.7. Simulation of velocity profile with constant acceleration start-up

From Figure 7, it can be seen that uniform acceleration start-up arises slight fluctuations of discontinuous velocity and significant fluctuations of acceleration, which will result in bad impacts on system stability.

2) Analysis and Simulation of uniformly-varied acceleration start-up

The velocity profile of uniformly-varied acceleration start-up is illustrated in Figure 8.

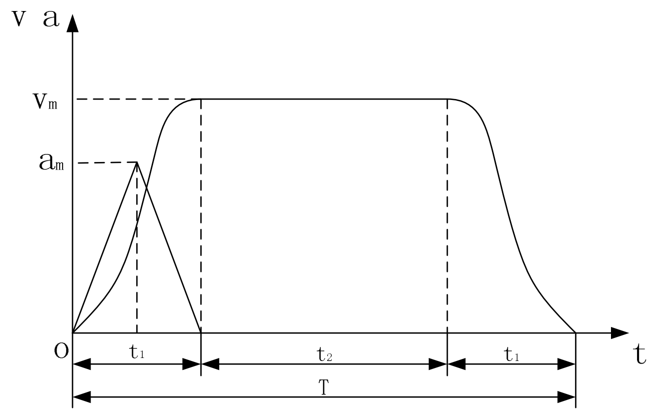


Figure.8. Velocity profile of uniformly-varied acceleration start-up

Let change curvature of acceleration denoted by ρ and the hodometer by *H*, then:

When $t = t_1$, $v_m = \frac{1}{2}\rho t_1^2$

$$H = 2\int_0^{t_1} v(t)dt + v_m t_2 = \frac{\rho t_1^3}{4} + \frac{1}{2}\rho t_1^2 t_2$$

$$T = 2t_1 + t_2$$

Motion settings are as: Type：Acceleration; Function：IF(time-0.125:-160000d*time, -20000d, IF(time-0.25: 160000d*time-40000d,0d,IF(time-1.75:0d,0d,IF(time-1.875:160000d*time-280000d,20000d,IF(time-2:-160000d*time+320000d,0d,0d))))). The simulation result is shown in Figure 9.

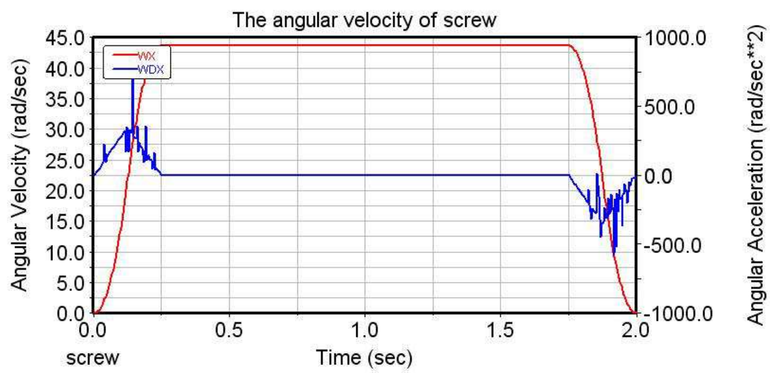


Figure.9. Simulation of velocity profile with uniformly-varied acceleration start-up

According to the results shown in Figure 9, it can be drawn that uniformly-varied acceleration start-up arises significant fluctuations a significant acceleration, which will result in bad impacts on system stability.

2) Analysis and Simulation of uniformly-varied acceleration start-up

The velocity profile of uniformly-varied acceleration start-up is illustrated in Figure 8.

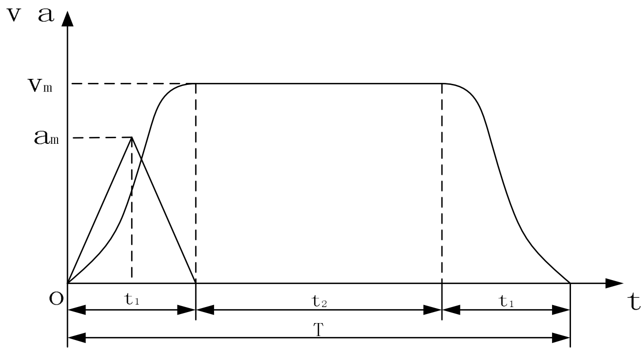


Figure.8. Velocity profile of uniformly-varied acceleration start-up

Let change curvature of acceleration denoted by$\rho$and the hodometer by H, then:

When $t = t_1$, $v_m = \frac{1}{2}\rho t_1^2$

$$H = 2\int_0^{t_1} v(t)dt + v_m t_2 = \frac{\rho t_1^3}{4} + \frac{1}{2}\rho t_1^2 t_2$$

$$T = 2t_1 + t_2$$

Motion settings are as: Type : Acceleration; Function : IF(time-0.125:-160000d*time,-20000d, IF(time-0.25:160000d*time-40000d,0d,IF(time-1.75:0d,0d,IF(time-1.875:160000d*time-280000d,20000d,IF(time-2: -160000d*time+320000d,0d,0d))))). The simulation result is shown in Figure 9.

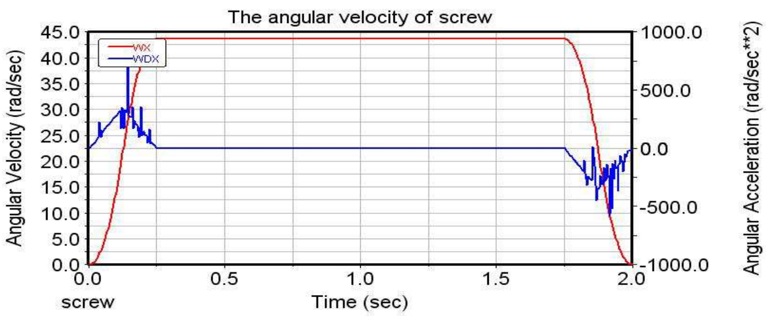


Figure.9. Simulation of velocity profile with uniformly-varied acceleration start-up

According to the results shown in Figure 9, it can be drawn that uniformly-varied acceleration start-up arises significant fluctuations a significant acceleration, which will result in bad impacts on system stability.

3) Analysis and Simulation of acceleration start-up with sine functions

The velocity profile of acceleration start-up with sine function is illustrated in Figure 10.

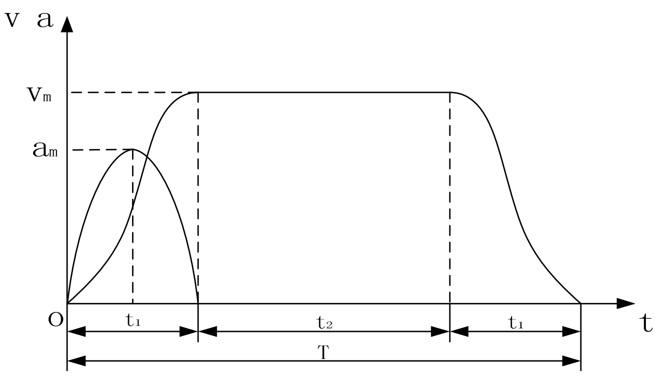


Figure.10. Velocity profile of acceleration start-up with sine functions

Let change curvature of acceleration denoted by$\rho$and the hodometer by H, then:

$$a = A\omega\sin(\omega t),\ \text{where } A\omega = a_m,\quad \omega = \frac{\pi}{t_1}$$

$$v = A(1-\cos(\omega t)),\ v_{t=\frac{t_1}{2}} = A$$

$$H_1 = A\left(t_1 - \frac{\sin(\omega t_1)}{\omega}\right)$$

$$v_m = A(1-\cos(\omega t_1)) = 2A$$

$$H_2 = v_m t_2 = A t_2(1-\cos(\omega t_1))$$

$$H = A\left(2t_1 - \frac{2\sin(\omega t_1)}{\omega} + t_2 - t_2\cos(\omega t_1)\right)$$

Motion settings are as: Type: Acceleration; Function: IF(time-0.25:-16000d*SIN(PI/0.25*time),0d,IF(time-1.75:0d,0d,IF(time-2:16000d*SIN(PI/0.25*(time-1.75)),0d,0d))). The simulation result is shown in Figure 11.

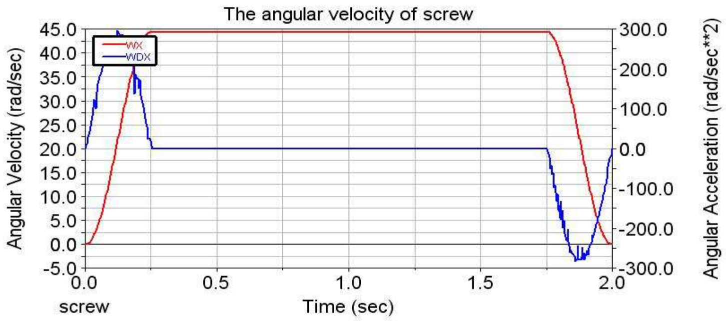


Figure.11. Simulation of velocity profile with uniformly-varied acceleration start-up

According to the results shown in Figure 11, it can be drawn that uniformly-varied acceleration start-up get smooth velocity with small acceleration fluctuation. However, the acceleration duration is long with low efficient.

Based on the above analysis, an optimized "sine+constant" acceleration start-up is proposed, which inherits the advantages of acceleration start-up with sine functions with improved efficiency. The velocity profile of acceleration start-up with sine function is illustrated in Figure 12.

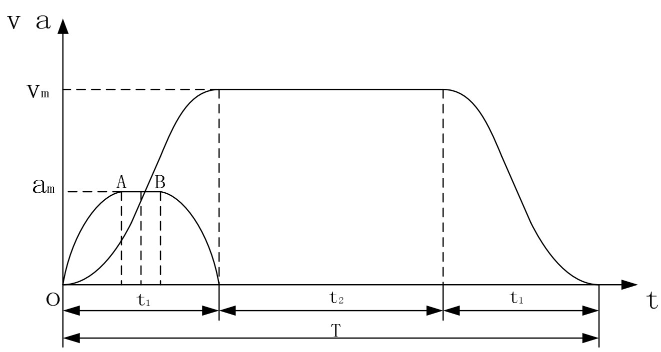


Figure.12. Velocity profile of "sine+constant" acceleration start-up

OA motion:

$$a = A\omega\sin(\omega t),\quad \text{where } A\omega = a_m,\quad \omega = \frac{\pi}{2t_A}$$

$v = A(1-\cos(\omega t))$, when $t = t_A$, $v_A = A$

$H = A\left(t - \frac{\sin(\omega t)}{\omega}\right)$, when $t = t_A$, $H_A = A\left(t_A - \frac{\sin(\omega t_A)}{\omega}\right)$

AB motion:

$v = A + A\omega(t - t_A)$, when $t = t_B$, $v_B = v_A + A\omega(t_B - t_A)$

$H = H_A + v_A(t - t_A) + \frac{1}{2}A\omega(t - t_A)^2$,

$when\ t = t_B$, $H_B = A\left(t_A - \frac{\sin(\omega t_A)}{\omega}\right) + At_{AB} + \frac{1}{2}A\omega t_{AB}^2$, where $t_{AB} = t_B - t_A$

BC motion:

$v = A + A\omega t_{AB} - A\cos\omega(t - t_{AB})$, when $t = t_1$, $v_1 = A + A\omega t_{AB} - A\cos\omega(t_1 - t_{AB})$

$H = H_B + A(t - t_B) + A\omega t_{AB}(t - t_B) + \frac{A}{\omega}(1 - \sin\omega(t - t_{AB}))$

when $t = t_1$, $H_1 = H_B + A(t_1 - t_B) + A\omega t_{AB}(t_1 - t_B) + \frac{A}{\omega}(1 - \sin\omega(t_1 - t_{AB}))$

Motion with constant velocity:

$$H_2 = v_1 t_2 = At_2 + A\omega t_{AB} t_2 - A\cos\omega(t_1 - t_{AB})t_2$$

Total hodometer : $H = 2H_1 + H_2$

Motion settings are as: Type：Acceleration; Function: IF(time-0.25: -16000d *SIN (PI /0.25 *time),0d , IF(time-1.75:0d,0d,IF(time-2 : 16000d*SIN(PI/0.25*(time-1.75)),0d ,0d))). The simulation result is shown in Figure 13.

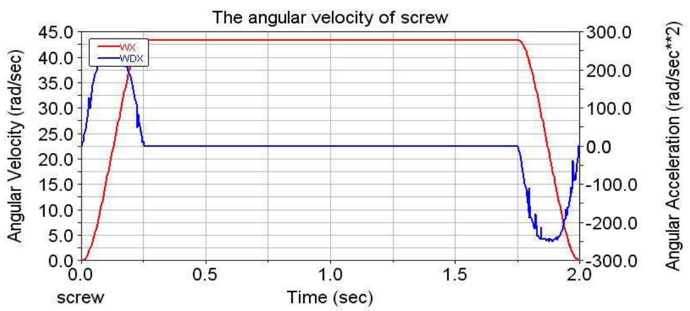


Figure.13. Simulation of velocity profile with “sine+constant” acceleration start-up

From Figure 13, it can be seen that “sine+constant” acceleration start-up get smooth velocity with very small acceleration fluctuation. Moreover, the maximum acceleration is much smaller with high efficient.

CONCLUSION

In this paper, taking transmission system of rail transit vehicles door as research object, virtual simulation and transmission stability of rotation roller screw are carried out. Then, velocity profiles are optimized for better transmission stability.In summary, optimal “sine+constant” acceleration start-up applied into screw results in improved performances in start-up, braking and efficiency.

Acknowledgements
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REFERENCES

[1] Yongwei Z; Jian S; Hongzhen C; *Journal of Chemical and Pharmaceutical Research*, **2013**, 5(9): 381-387.
[2] Xue Y; Hongbo W; Qiang W;et al. *Journal of Chemical & Pharmaceutical Research*, **2014**, 6(1)：266-270.
[3] Zhan Xiaoming; Dynamic structure analysis and simulation of ball screw. Southeast university, Nanjing, **2010**.
[4] Ji Qianzhong; Yang Jiajun; *Mechanical Science and Technology*, 199, 18(2):230-232.
[5] Yang Baozhe; The Analysis and Research of Load Sharing and Stationarity for Planetary roller screw. Huazhong University of Science & Technology, Wuhan, **2012**.
[6] Wei Zhenxing;Yang Jiajun; Zhu Jisheng; et al. *Journal of Mechanical Transmission*,**2011**,35(6):44-48.
[7] Jones M H, Velinsky S A. *Journal of Mechanical Design*, **2012**, 134(6): 061006.
[8] Jones M H; Mechanics Based Design of the Planetary Roller Screw Mechanism. University of California, Davis, **2013**.
[9] Jones M H; Velinsky S A; *Mechanics Based Design of Structures and Machines*, **2014**, 42(1): 17-34.
[10] Yang Z; Wang R; Wu M; et al; *Journal of Chemical and Pharmaceutical Research*, **2014**, 6 (1): 434-438.
[11] Kang-xing D; Min-zheng J; *Journal of Chemical & Pharmaceutical Research*, **2013**, 5(9)：182-187
[12] Chen S H;Jehng W D; Chen Y S; *Journal of Chemical & Pharmaceutical Research*, **2013**, 5(9)： 314-323
[13] Manli D; Gang W;Chun S;et al. *Journal of Chemical & Pharmaceutical Research*, **2014**, 6(2).