한국기계가공학회지, 제19권, 제7호, pp.52～59(2020.07) ISSN 1598-6721(Print)
Journal of the Korean Society of Manufacturing Process Engineers, Vol. 19, No. 7, pp.52~59(2020.07) ISSN 2288-0771(Online)

https://doi.org/10.14775/ksmpe.2020.19.07.052

A study on the Design of Drum Type Automatic Tool Changer

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드럼형 자동공구교환장치의 설계에 관한 연구

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(Received 17 June 2020; received in revised form 19 June 2020; accepted 05 July 2020)

ABSTRACT

Automatic tool changers (ATCs) can be divided into drum and chain types. Drum-type ATCs contain a magazine, where the tools are mounted, and a cam gearbox, which swaps the tools via roller gear and grooved plate cams. Drum-type ATCs are advantageous in that the operating time for the tool magazine is more rapid than that of chain-type ATCs and the length of the unit is shorter. Thus, drum-type ATCs can be fabricated into various shapes and forms depending on the number of tools and the magazine size in accordance with machining center requirements and consumer demand. In particular, the price competitiveness of a machining center with a drum-type ATC is higher, while drum-type ATCs are more rigid with fewer parts, possibly reducing the need for regular servicing. This study aims to verify the structural stability and design validity of the magazine base, which is the main structure of a drum-type ATC, using finite element analysis. This study kinematically verifies the specifications of the selected drive motor and reducer and assessed the design of the cam gearbox. It also conducts a structural analysis of the roller camp, which is the core component of the cam gearbox, based on the results of the kinetodynamic analysis, thus validating the structural design.

Keywords : Automatic Tool Changer(자동공구교환장치), Finite Element Analysis(유한요소해석), Magazine Parts(매거진부), Cam Gear Box Parts (캠기어박스부), Roller Cam(롤러 캠)

1. Introduction

Automatic tool changers (ATCs) can be classified into two types based on the shape of the magazine: drum-type and chain-type. The former can be divided into the magazine, onto which the tools are mounted, and a cam gearbox, which uses roller gear and grooved plate cams to change the tools. Drum-type ATCs have a faster operating time and a shorter unit length than chain-type ATCs, allowing them to be more flexible in terms of design and size to meet the requirements of the machining center and consumers in terms of the number of tools and the magazine size. Drum-type ATCs are also more cost-effective, especially in terms of the potential costs of servicing because drum-type ATCs are more rigid with fewer parts and are thus easier

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to service. Using finite element analysis (FEA), this study aims to verify the structural stability and design of the magazine base of a drum-type ATC. In particular, the specifications of the drive motor and reducer selected and the design of the cam gearbox are kinematically verified. Structural analysis of the roller camp, a key component of the cam gearbox, is also conducted based on this kinetodynamic analysis.

2. Structure of an Automatic Tool Changer

An ATC is composed of a tool magazine and a cam gearbox (Figure 1). The drive arm in the cam gearbox is rotated to mount a tool stored in the tool magazine along the principal axis and then returned to the magazine when the tools are changed. The tool magazine assembly consists of a tool port and linking components, onto which 60 tools are mounted, and the magazine base and drum magazine, which are rotated to change tools. The cam gearbox assembly automatically attaches and detaches the tools by combining vertical linear motion and the forward and reverse rotation of the arm. This forward and reverse movement is controlled by the roller gear cam, while the vertical linear movement is the product of the grooved plate cam.

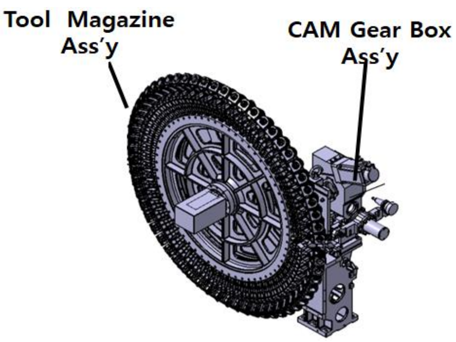

Fig. 1 Structure of drum type automatic tool changer

3. Structural Analysis of the Magazine Base

The magazine base is the main structural component of the ATC. It mounts the cam gearbox and supports 60 tools. This study aims to verify the stability and validate the design of this structure in the design phase using FEA. The optimal design of the structure can be determined by generating a finite element model and improving the design based on the results of this analysis. This process of improving the design and validating the improvement can be iteratively repeated. To analyze the structural characteristics using FEA, a mesh that divides the three-dimensional (3D) model into finite elements is employed. The larger the number of elements, the higher the accuracy. However, this also increases the analysis time and can lead to memory shortages. Thus, the finite element model was created by setting the number of elements to within 1% of the total system size.

The material of the magazine base was selected to be GC300. Its modulus of elasticity, tensile strength, density, and Poisson's ratio were set at 200,000 MPa, 241 MPa, 7.15E-09 ton/㎣, and 0.2, respectively. In addition, considering the weight of the magazine, cam gearbox, and tools, the loading conditions were established, and the boundary conditions were set so that the displacement (UX, UY, UZ) was restrained to the node fixed at the lower support of the magazine base and the displacement load was placed on the body. The primary analysis results indicated that deflection occurred with a maximum deformation of 0.432 mm and a maximum stress of 33.4 MPa, which occurred on the lower side of the magazine base. The safety factor was 7.2, which was considered structurally acceptable based on the yield stress of the material (241 MPa). However, secondary analysis was conducted by installing a rib at the rear of the

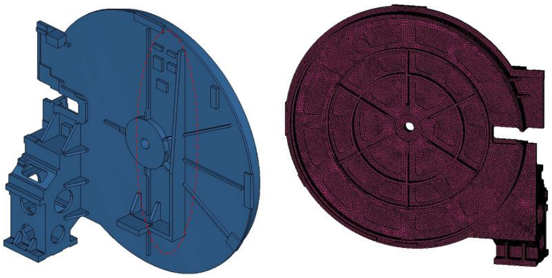

Fig. 2 Reinforced design and modified model of rear rib

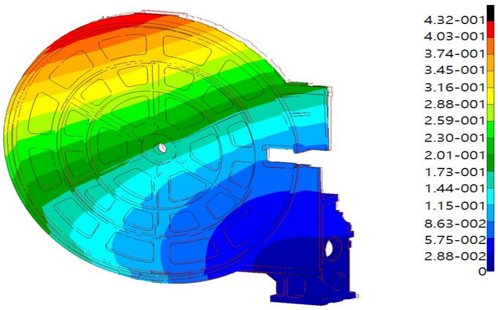

Fig. 3 Analysis result before complementary design of structure

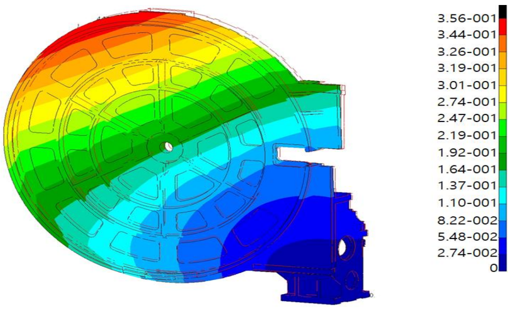

Fig. 4 Analysis result after complement design of rib

magazine base (Figure 2) to improve the high-rigidity structural design. The second analysis produced a maximum stress of 34.6 MPa on the lower side of the magazine base, which was the same location as in the first analysis. The maximum deformation of 0.356 mm was reduced by 0.076 mm (17%) compared to that produced before adding the rib. Figure 3 presents the analysis results prior to the improvements to the structural design, and Figure 4 displays the analysis results for the rib-reinforced design.

4. Design of the Cam Gearbox

The cam gearbox consists of an arm that rotates to change tools, a gear that rotates the arm, a roller cam, turret, camfollower, lever, and drive motor. Figure 5 shows the structure of the cam gearbox. It has rotation movements of the arm through one drive motor and roller cam and vertical movements of the arm to avoid the interference. The roller cam is the cam mechanism that provides vertical movements for the ATC. The camfollower is inserted into a groove on the roller cam to deliver the driving force to the link arm, which is designed to deliver the drive force for vertical movements along the tool change axis. The tools are changed when the holder in the spindle shaft axis vertically moves along the axis through the link when the camfollower roller moves along the groove of the roller cam and the holder is connected to the tool change axis, thus changing the tools via the vertical and rotational movements produced by the roller cam. Six camfollower rollers are arranged on the turret at an angle of 60°. In addition, assembly is simplified by installing three eccentric caps, and noise is minimized by adjusting the eccentric caps when noise is generated. The gear backlash is adjusted by designing an eccentric gear housing, and the distance between the camfollower and cam can be adjusted by installing an eccentric cap in the

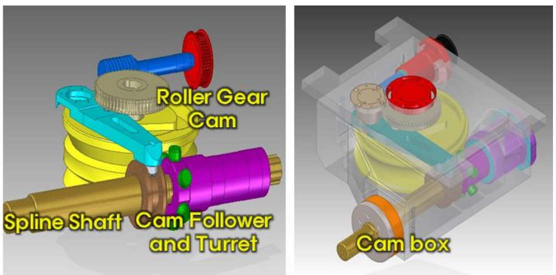

Fig. 5 Structure of cam gearbox

portion supporting the whole roller cam in the body and cover.

5. Kinetodynamic Analysis of the Cam Gearbox

5.1 Purpose and Scope of the Kinetodynamic Analysis

Kinetodynamic analysis of the cam gearbox drive was conducted. Through this, the drive torque required for the rotational movement of the arm to change tools was calculated to kinetically verify the specifications of the initially selected motor and reducer, thus validating the design. In addition, the maximum contact force of the roller cam was derived in the analysis, which was then assigned to the roller cam. Based on this, structural analysis was conducted to validate the structural design of the cam. Figure 6 presents a layout model of the dynamic analysis.

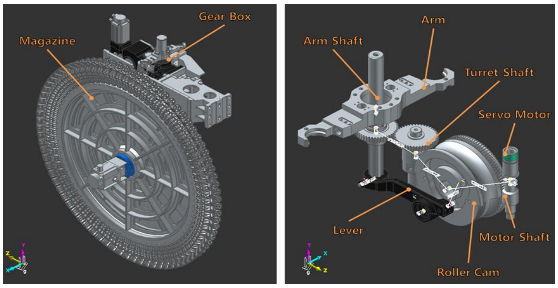

Fig. 6 3D layout model of simulation analysis

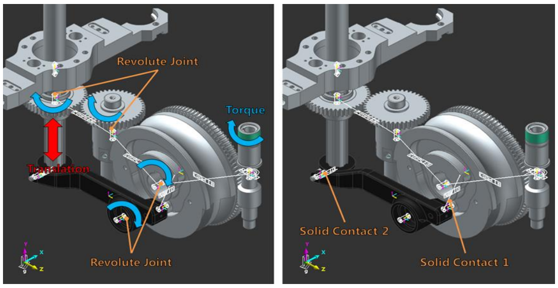

Fig. 7 Constraints in the analytical model

Table 1 Gear ratio of drive gear

Distribution	Gear ratio
Motor Shaft : Roller Cam	5 : 1
Roller Cam : Turret Shaft	2 : 1
Turret Shaft : Arm Shaft	1 : 1

5.2 Constraints and Drive Conditions

The constraints and drive conditions for the analysis model are presented n Figure 7. A revolute joint was assigned to the drive motor, each shaft, and lever, and the static friction coefficient of each rotational axis and dynamic friction coefficient were set to 0.1 and 0.08. Table 1 presents the gear ratio for each gear. Because the lever moves along the orbit of the roller cam, solid contact was assumed, and the damping coefficient was set to 1/10,000 of the spring coefficient. Here, the spring coefficient K was calculated using Eq. (1). R_1, R_2 in Eq. (1) refer to the radius of contact surfaces 1 and 2, respectively, E_1, E_2 refer to the modulus of elasticity of contact surfaces 1 and 2, respectively, and ν_1, ν_2 refer to the Poisson's ratio of contact surfaces 1 and 2, respectively.

$$K = \sqrt{\frac{16RE^2}{9}},$$

$$R = \frac{R_1 R_2}{R_1 + R_2},$$

$$E = \frac{E_1 E_2}{E_2(1-\nu_1^2) + E_1(1-\nu_2^2)} \qquad (1)$$

5.3 Analysis Results

The analysis results are shown in Figures 8 and 9. The maximum rotational angular velocity of the motor shaft during one rotation of the arm by rotating 10 revolutions (1,800 degrees) of the motor shaft for 1.5 sec was 31.4 rad/sec. The arm was finally rotated 180 degrees, as shown in Figure 10. The maximum rotational angular velocity of the arm was 3.14 rad/sec. The arm shaft was moved up to

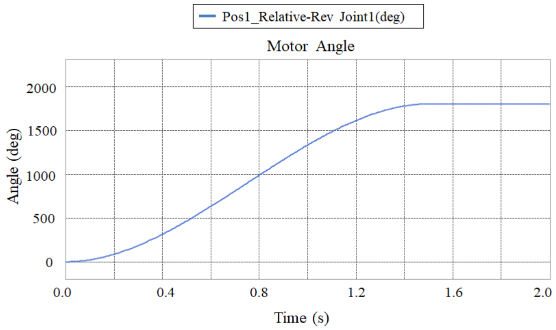

Fig. 8 Motor rotation angle

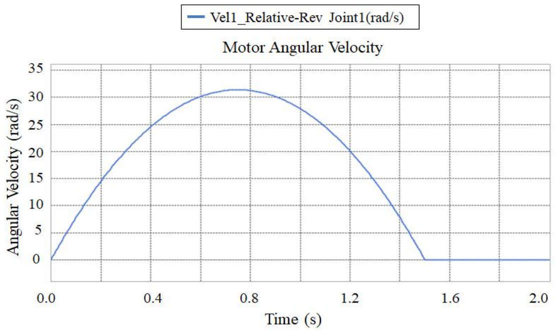

Fig. 9 Angular velocity of motor shaft

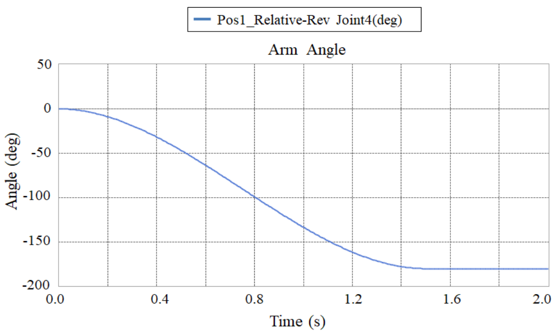

Fig. 10 Arm rotation angle

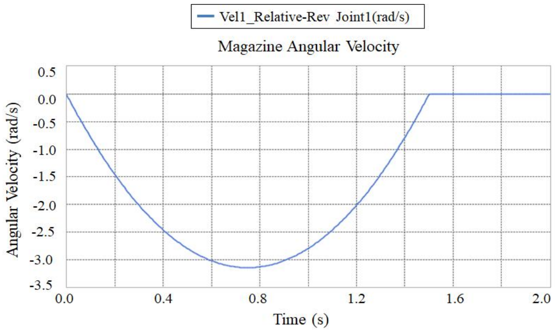

Fig. 11 Angular velocity of magazine

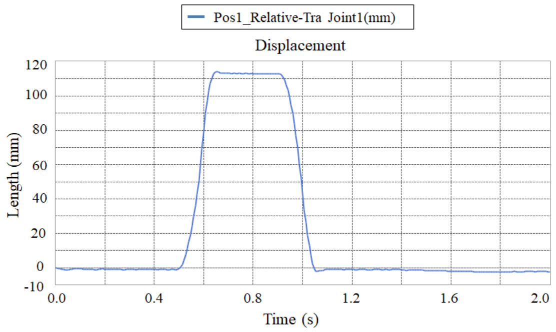

Fig. 12 Height of arm shaft

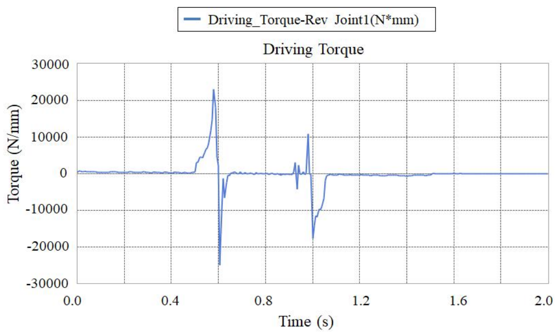

Fig. 13 Motor driving torque

115 mm by the cam's orbit and then returned to the original position. Figures 11 and 12 present the angular velocity of the magazine and the height of the arm shaft. The maximum torque required for one revolution of the arm was 218,238 N•mm and the average torque required was 15,285 N•mm (Figure 13). The maximum torque required to rotate the arm by 180 degrees for 1.5 sec in the gearbox drive of the ATC was 218.23 N•m, and the rated torque was 15.28 N•m or larger. Given the reducer's transmission efficiency of 70% and a reduction gear ratio of 1:47, the maximum and rated torques required for the selected motor were 6.63 N•m and 0.46 N•m, respectively, which satisfied the specifications

Table 2 Specification of motor selected in design

Items	Specifications				
Rated output	kWalt	0.8	1.5	2.0	3.0
Rated torque	kgf·cm	25.98	73.08	97.44	146.16
	N·m	2.55	7.16	9.54	14.32
Continuous Max. torque	kgf·cm	77.95	219.24	292.32	438.48
	N·m	7.64	21.49	28.66	42.49
Rated speed	RPM	2,000			
Max. speed	RPM	3,000			
Power rate	kW/S	23.64	13.28	42.29	90.94

of the selected motor (Table 2). In addition, the maximum rotational angular velocity of the output axis was around 300 rpm, which satisfied the reducer specifications.

6. Structural Analysis of the Roller Cam

Static structural analysis was conducted based on finite elements by applying the maximum contract force derived from the kinetodynamic analysis to the roller cam. To produce the finite elements, HyperMesh was utilized, with the length of one side set to 5 mm by default. Finite elements such as hexes, wedges, and tets were used to complete the mesh. The boundary conditions were assigned by matching the node in the portion assembled by the bolt and key groove. The boundary conditions were restrained in three regions in the central axis with three degrees of freedom. The material used was SCM415, the modulus of elasticity was set to 220,000 MPa, the tensile strength was 350 MPa, the density was 7.83E-09 ton/㎣, and Poisson's ratio was 0.27. Figure 14 displays the analysis constraints and boundary conditions, while Table 3 presents the maximum contact load of the roller cam derived from the dynamic analysis. The static stiffness analysis results revealed that the stress and deformation were concentrated in the contact area of the camfollower, with a maximum displacement and deformation of 0.016 mm and 207 MPa, respectively, which did not exceed the yield stress. Thus, it was determined that the static stiffness of

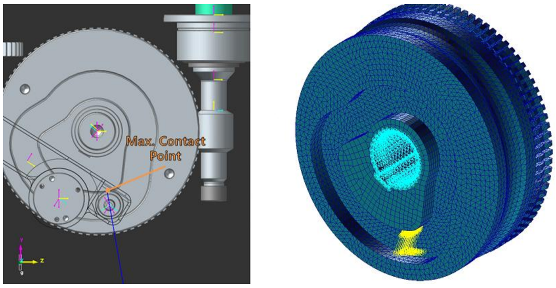

Fig. 14 Analysis and boundary conditions

Table 3 Maximum contact force of roller cam

Contact Force(N)	Fx	Fy	Fz
Roll CAM	-4	27,647	11,047

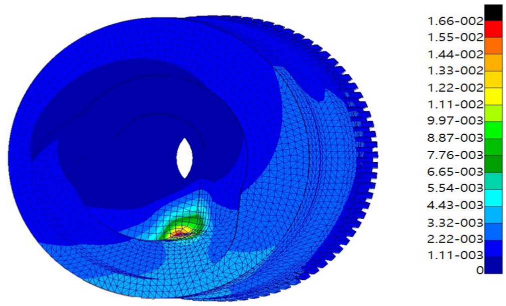

Fig. 15 Analysis result of dynamic characteristics of roller cam

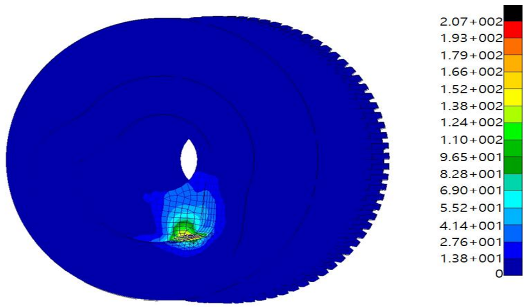

Fig. 16 Results of the static rigidity analysis of the roller cam

the cam in the roller cam drive box of the ATC was acceptable. Figures 15 and 16 present the structural analysis results for the roller cam.

7. Conclusions

A drum-type ATC consists of a magazine, on which the tools are mounted, and a cam gearbox, which is employed to change the tools via roller gear and grooved plate cams. In this study, the stability of the design was verified based on FEA of the magazine base, thus validating the selection of the drive motor, reducer, and cam gearbox design. This study also conducted structural analysis of the roller cam. The following conclusions were derived:

1. Primary structural analysis of the magazine base revealed that deflection occurred with a maximum deformation of 0.432 mm and a maximum stress of 33.4 MPa. To minimize the deformation, reinforcement with a rib at the rear of the magazine base was employed. As a result, the deflection was reduced by 0.076 mm, which was 17% lower than the deflection before adding the rib. The rib was thus applied to the final structural analysis.
2. The design of the cam gearbox was assessed using kinetodynamic analysis. The maximum and rated torques required for the selected motor based on the reducer's transmission efficiency and the reduction gear ratio were 6.63 N•m and 0.46 N•m, which satisfied the motor specifications. In addition, the maximum rotational angular velocity of the output axis was around 300 rpm, which satisfied the reducer specifications.
4. Static structural analysis of the cam was conducted by applying the maximum contract force derived from the kinetodynamic analysis to the roller cam. The resulting maximum displacement was 0.016 mm and the deformation was 207 MPa, which did not exceed the target displacement and yield stress of the design. Thus, the static stiffness of the roller cam was found to be acceptable.
5. The lifetime of the roller cam will be determined using driving simulations of the roller cam to validate the final design. A prototype will then be manufactured and experiments carried out to verify the analysis results.

Acknowledgments

This study was conducted using funding from the Technical Development Project (Task No. 20001305) in the Advanced Technology Center supported by the Ministry of Trade, Industry and Energy and the Korea Evaluation Institute of Industrial Technology. We appreciate their support.

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